AIR CONDITIONING

AND ELEMENTS OF REFRIGERATION

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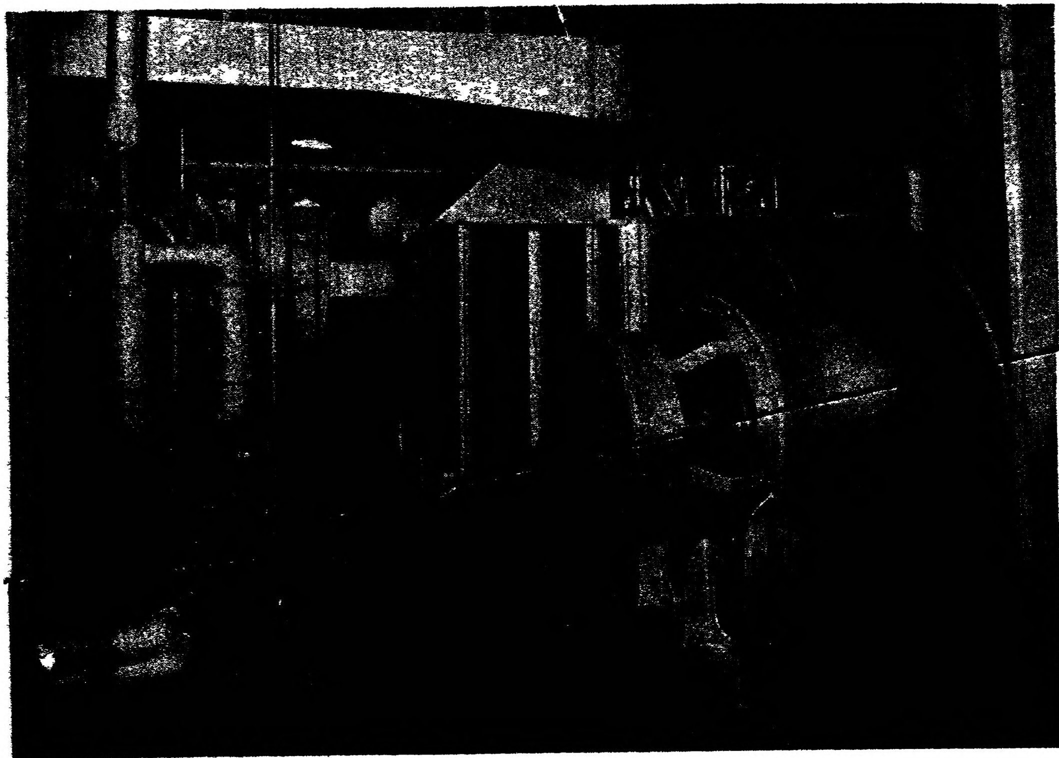

Frontispiece

GENERAL VIEW OF A TYPICAL AIR CONDITIONING EQUIPMENT ROOM. *GENERAL ELECTRIC CO.*

AIR CONDITIONING

and *ELEMENTS OF REFRIGERATION*

by SAMUEL P. BROWN., B.S., P.E.

Staff Engineer, Coverdale and Colpitts, Consulting Engineers, New York
Member Board of Directors, former Chief Instructor Vocational Division, Delehanty Institute
Registered Professional Engineer, New Jersey and Ohio

with the editorial assistance of
JOHN D. CONSTANCE
Registered Professional Engineer, New York

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AIR CONDITIONING

AND ELEMENTS OF REFRIGERATION

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To My Wife, Helen Marie

PREFACE

This book is based upon a textbook originally prepared for a resident course in air conditioning conducted by the Delehanty Institute of New York for use in connection with classroom training—training of vocational students of all ages having a background of high-school education or its practical equivalent. College students, and of course college graduates, will therefore find much of the explanation unduly detailed, particularly as to the phases of basic science explained at considerable length wherever necessary as a preliminary to understanding some phase of air conditioning. Yet the practical material in Chap. VII and thereafter has been used as the basis for lectures to college students and to graduate engineers and architects.

The objective of this book is to provide a complete, practical, working textbook and handbook suitable for study by any interested person and especially desirable for those who are working or have worked in design, installation, or operation of heating, ventilating, and allied mechanical equipment of buildings. The actual design of air-conditioning equipment, complete units and components thereof, has become so technical as to require not only graduate engineers, but engineers with considerable specialized experience. The design of the bulk of the installations, however, including surveys of premises and selection of equipment and job layout, which combine to produce satisfactory results, is the chief concern of this book. Plumbing and heating contractors who wish to add air conditioning to their line and building contractors who buy air-conditioning installations will both find the material of great value.

Anyone engaged in operating, purchasing, or installing air-conditioning equipment must be able to talk intelligently with equipment sales engineers and be able to understand catalogue information and other data published by manufacturers—in short, be able to match equipment with the needs of the job on a basis of the facts, regardless of the eloquence of a too persuasive equipment purveyor. To this end, the author has emphasized the understanding of fundamentals: how much cooling or heating does a certain place need; how much cooled and dried air is needed to do a certain summer air-conditioning job; how much equipment is needed to cool and dry the air; what general arrangement of automatic controls is needed. Throughout the text, illustrative practical problems are presented; typical manufacturers' rating data for equipment are reproduced, and their use is explained. The latter, supplemented by the reproduction of necessary data from the American Society of Heating and Ventilating Engineers' "Guide," makes this book a complete manual for the design of all but the most complex multistory systems.

It should be remembered that equipment manufacturers change models from time to time; therefore the local sales office of two or more competitive manufacturers should always be consulted and asked for quotations on price and technical data. Basically different methods of cooling often should be compared, and different layouts of equipment and ducts should always be compared, to assure the most economical scheme.

It is impossible to express here appreciation due to all the equipment manufacturers who cooperated in furnishing illustrations and technical data; instead, an effort has been made to show individual credit where material is used. Special thanks are conveyed to the American Society of Refrigeration Engineers and to the American Society of Heating and Ventilating Engineers for permission to reprint technical data from the handbooks of these two societies. The author also wishes to thank E. W. Feller, associate editor of *Power*, for his advice and editorial assistance.

Particular thanks and appreciation are due John D. Constance, registered professional engineer, New York, whose unstinting work and wide engineering experience have been of immeasurable value in the preparation of the manuscript for publication. For over 6 months Mr. Constance devoted much time to editorial review, contribution of additional practical problems, obtaining the latest manufacturers' photographs and data, and making real creative contribution to the final draft of the manuscript.

Finally, the author wishes to express his gratitude to M. J. Delehanty and the members of the Delehanty Institute for their splendid cooperation and foresight in making this work possible.

SAMUEL P. BROWN

SHORT HILLS, N. J.,
March, 1947.

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INTRODUCTION

AIR CONDITIONING IN THE WORLD AROUND YOU

First air-conditioned theaters and then air-conditioned restaurants have become well known to the general public. Today when the members of a family are discussing what motion-picture theater to go to, they are likely to say, "Let's go to the Ajax; it is air-conditioned."

Theaters were among the earlier ventures in the air conditioning of public buildings, and it is unfortunately true that the effect is not always pleasant. There are many air-conditioned theaters in which the temperature is held too low or in which the humidity is permitted to reach almost 100 per cent. Such conditions are definitely unhealthy.

Upon leaving an overcooled building, the hot street air strikes one like a blast from an oven. The mere discomfort of this is not the end of the story. There is a nervous shock involved, which many doctors feel is harmful.

Most buildings in which substandard air conditioning was installed made these installations 8 years or more ago. Therefore, this early equipment is wearing out, and better designed modern equipment is being installed. Recently, two theaters near New York discarded their original air-conditioning plants in favor of up-to-date modern equipment.

Restaurants did not accept air conditioning widely until after it had become well known in the theater field. Hence, there are very few restaurant air-conditioning projects that have proved unfit. If you encounter a restaurant where the air-conditioning equipment does not provide a properly pleasing indoor atmosphere, nine chances out of ten the blame is not with the air-conditioning installation or design. Instead, it is probably caused by improper operation in one way or another.

Larger Comfort Air-conditioning Projects. It is a fact that at the present time there is no building too large to be air conditioned. Just before the war, a large insurance company had its office building, over 20 stories high, covering an entire city block, 100 per cent air conditioned all the year round. Another large group of buildings has thousands of tons of refrigeration for its air-conditioning load. The equipment involved handles the air-conditioning demands of the tenants in those buildings.

In the construction of new, higher type city buildings, it is rare indeed that some air conditioning is not included. Ordinarily, the entire building is winter air-conditioned when it is built, while sufficient summer air conditioning is installed to handle one-third to one-half of the building.

Then as the need arises (some New York building owners have found it arising more rapidly than they could handle it) additional summer air conditioning is provided.

In year-round air conditioning of city office buildings and apartments, the air-conditioning industry owes thanks to the banks and insurance companies for their help.

Many of New York's downtown banks have air-conditioning plants of 50 to several hundred tons capacity. Insurance companies, likewise, have been quick to realize the increased employee efficiency, the improved health, and the better morale resulting from providing completely air-conditioned offices and recreation rooms.

Air Conditioning in Residences. In the apartment-house field the surface has hardly been scratched. Prior to the war, a few new apartment houses were completely air-conditioned, but throughout the East these constituted only a small percentage of the total apartment-house construction Some builders recognizing the advantage of having a building completely air-conditioned include this work at the time of constructing the building, so that the air-conditioning equipment can be properly designed as an integral part of the building.

In private homes, both large and small, winter air-conditioning has become accepted. In new homes costing anywhere from $4,900 to $15,000, winter air-conditioning plants are usually available at a cost not above that of an ordinary heating system. As a matter of fact, in many large developments, progressive builders are utilizing winter air conditioning as a *standard* feature and charging an additional cost if a purchaser requires older heating equipment.

However, in spite of its head start over apartments with winter air conditioning, the air conditioning of private homes is not yet at the point where summer air conditioning is common. This field promises to be one of the most productive in the future. The reason is that most winter air-conditioned homes have complete and well-designed duct systems with provision for adding summer cooling at a later date. This means that, as soon as the cost of summer cooling for a residence is reduced somewhat, many homeowners will want to add this feature to their present winter air-conditioning and ventilating systems.

Industrial Air Conditioning. Few people, even those in the air-conditioning industry itself, realize the extent of industrial air conditioning. A small percentage of those engaged in the air-conditioning field today realize that in the early 1920's the air conditioning of bakeries was becoming a common thing. This does not mean that bakeries were maintained at the same conditions that we strive for in theaters, offices, restaurants, etc. On the contrary, the conditions that are desirable in industrial processes—in the tobacco industry, the textile industry, the baking industry, the electrical industry, and many others—are frequently conditions that would be extremely uncomfortable for other purposes.

The increase in industrial air conditioning from 1920 to 1935 was brought home rather sharply to the author several years ago. At that time he was discussing the subject with a mechanical engineer who had had about 8 years' experience in both industrial and comfort air conditioning. This man had compiled numerous figures on the subject, and the conclusion he drew was that up to 1925 industrial air conditioning had in every year accounted for 85 to 90 per cent of the total air-conditioning business. Since industrial air conditioning is frequently mechanically connected and integrally designed with a comfort air-conditioning installation, it is difficult to get exact facts today. However, from the number of firms that handle industrial air conditioning exclusively, it is probably true that the industrial work accounts for at least half of the total air-conditioning business today.

It does not seem likely that industrial air conditioning will ever be eclipsed by comfort air conditioning. Each month—almost each week—new industrial applications for air conditioning are being discovered. In the manufacturing of delicate instruments, electrical and radio parts, etc., it has recently been discovered that several processes can operate much more advantageously in carefully air-conditioned spaces. From the standpoint of one within the air-conditioning industry, the advantage in this is that these industrial applications of air conditioning frequently pay for the cost of the installation in a surprisingly short time. This may be contrasted with the case of a restaurant, shop, or theater owner, who must plan on getting back his investment over a period of quite a few years.

From the viewpoint of engineers and contractors in air-conditioning work, industrial projects have one feature that makes them more attractive than comfort installations. In the first place, the industrial purchaser ordinarily has engineers on his staff who are somewhat familiar with the problem involved in air conditioning. Furthermore, these engineers ordinarily know exactly what air conditions they desire for their process. In the case of a comfort installation, frequently the air-conditioning man is called upon to recommend the desired conditions in the conditioned space.

Without any knowledge of air conditioning, the difference in these two situations is easily understood. In the industrial instance, the purchaser specifies his conditions, and instruments are set up to determine if these specifications are being met. In the case of the comfort installation, on the other hand, instruments may prove that the specified conditions are being met; but if the purchaser is not pleased with these conditions, he will make a tardy demand that the specified conditions be changed. Fortunately, however, the public and air-conditioning purchasers are now becoming educated regarding the proper standards of air conditioning for human comfort.

Engineering in Air Conditioning. It is true that today there are numerous so-called "package" or "self-contained" units available in a wide variety of sizes and types. The use of these is feasible from a cost stand point only when the amount of space to be conditioned is relatively small. Even with these package units, however, since there are many makes, models, and designs, it is quite an engineering problem to select the proper unit or units to handle a certain air-conditioning load.

The bulk of the air-conditioning tonnage, both industrial and comfort, consists of central plant installations, *i.e.*, installations that are designed and installed tailor-made for the particular application This means that, except for minor mechanical repair details, personnel in the air-conditioning field must have some training in basic scientific subjects, plus specialized training in the engineering problems involved in the air-conditioning work

In the field of central heating, 15 or 20 years were required to make it possible for ordinary plumbing and heating men to select and install heating equipment without engineering assistance. Even today, large buildings, offices, hospitals, schools, factories, apartments, etc , ordinarily hav their heating plants designed by engineers and architects rather than by the heating contractor. Simple heating installations involve only the control of the *temperature* of the air within the building. When we come to air conditioning, the problem becomes much more complex. According to the American Society of Heating and Ventilating Engineers, air conditioning has been defined as the "supplying and maintaining, in a room or other enclosure, of an atmosphere having a *composition*, *temperature*, *humidity*, and *motion* which will produce desired effects upon the occupants of the room or upon materials stored or handled in it." The "composition" of the air involves our controlling its *cleanliness* and the portion of *outdoor air* that is delivered to the air-conditioned space. Hence, you see that, in *air conditioning*, temperature is only one of *five factors* that must be controlled Perhaps it would not be accurate to say that year-round air conditioning involves five times as much engineering as an ordinary heating installation. However, it is certainly correct to say that it involves a great deal more engineering skill.

Everyone should be interested in the engineering problems of air conditioning and ventilating work from either one or two standpoints. (1) Both men and women in the metropolitan area spend a good many hours in air-conditioned buildings. They are interested in having conditions such that they are always comfortable. They demand air at the proper temperature and humidity, air that is clean and dust-free, inside air with a sufficient proportion of outdoor air; last but not least, they demand enough air motion without any drafts. (2) Men with mechanical minds and aptitudes are looking more and more toward air conditioning as a field for employment. For the specialized trade mechanics such as electricians, steam fitters, sheet-metal men, and mechanics' laborers, this work offers no special problems, it is just another job, and since they do not understand e job as a whole they do their own work well, just as they would if they were helping to build a new superservice garage. However, for men who have the natural aptitude, the mechanical inclination, plus the specialized training in the engineering phases of air conditioning, there is an opportunity to work in the planning, designing, installation, supervision, and installation-testing field of air-conditioning projects.

We shall therefore present a summary of the work involved on the part of those who complete the design and operation of air-conditioning systems.

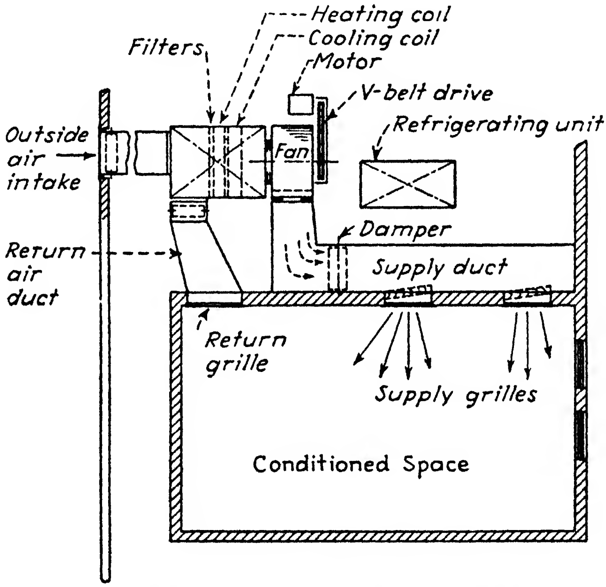


FIG. 1.—Layout of a simple air-conditioning system.

Few people realize the complexity of this problem and the diversity of the knowledge required in not one but numerous related engineering fields. To have modern air-conditioning men appreciated by the public as they should be, it is necessary that the public become educated, in a nontechnical manner, concerning the nature of modern air conditioning. See Fig. 1 for a simple air-conditioning system.

We have outlined briefly the historical growth of air conditioning within the past 15 or 20 years. Actually, of course, isolated cases of air-conditioning installations can be found dating from much earlier than this. Comfort installations were made in New York City as early as 1889. However, the development of engineered air conditioning, as we know it today, has taken place mainly during the past 20 years.

We shall now outline the equipment and numerous devices that are involved in a modern air-conditioning job, so that you will better appreciate the broadness of the field and the scope of the specialized training that is required for responsible designing and supervisory work in this field. In order to do this, let us trace the path that would be taken by outdoor air traveling through an air-conditioning system. Outdoor air would enter the system through some type of rainproof opening equipped with waterproof louvers or other means to keep out rain and snow. Since outdoor air is surprisingly dust-laden, it would then pass through some type of filter or air cleaner. This must be carefully selected as to size, type, and method of installation. If the building owner has no mechanical maintenance personnel, he could use a throw-away type of filter having a paper frame. These are easy to replace but involve a nominal replacement cost every 3 to 5 months. If the building owner has maintenance men on the premises, he could save money by buying more costly filters of a type that could be cleaned and reused The designer of the air-conditioning system must carefully select the proper filters to meet the needs of each installation.

After leaving the filter, the air that we are following would pass over two heating coils. These heating coils resemble automobile radiators that are filled with steam in order to heat the air that flows through them. The first heating coil operates only when the outdoor air is colder than 35°F. The other heating coil operates so that the air after leaving it will be at a predetermined temperature from 75 to 100°. These coils must be properly selected with regard to the amount of air they must heat and the type of steam that will be supplied to them. Furthermore, proper *automatic* controls must be provided to turn the steam on and off as required.

The duct through which our outdoor air is traveling at this point is called the "outdoor-air intake duct." As we follow the air along this duct, we should shortly find it joined by another duct that would be bringing return, or recirculated, air back from the air-conditioned rooms. In certain cases this return-air duct would join the fresh-air intake duct between the two steam-heating coils described above. This is one of the decisions that the designer of the system would have to make. Our mixture of outdoor air and recirculated air would then enter one of the major units of the system, the best general name for which is "conditioner." Conditioners are of many different types, sizes, and designs, and the selection of the proper conditioner is an important task that is governed by many factors. The majority of these factors will be pointed out as we proceed.

Conditioners may range from the small factory-assembled type, which are shipped in one piece ready to operate, up to large conditioners as big as the average two-car garage. The large conditioners, of course, are built up right on the job. The interior of the conditioners may consist of finned coils resembling huge automobile radiators through which chilled water

(50°F), hot water, or steam may be circulated, depending upon the season of the year and the operating conditions. Other types of conditioners called "air washers" consist of a mass of spray nozzles or other water-atomizing devices such that the entire chamber is filled with a fog of tiny water particles.

During the winter season, these conditioners add humidity and may or may not add heat to the air. During the summer season, they remove humidity from the air and cool it. Conditioners must be selected carefully so that they will be able to add or remove the proper amount of humidity and heat. They also must be sized according to the amount of air that is to be handled.

The air is moved about through its duct system by means of fans of a type called "centrifugal fans" or "blowers." In the case of the smaller factory-assembled conditioners, which are now available in sizes sufficiently large to condition restaurants having a seating capacity of 150 to 200 or larger, the fans are frequently built in the conditioner itself. Where the conditioner is the type assembled on the job, as is the case in theaters, large building installations, etc., the fan is generally a separate unit connected to the outlet of the conditioner by means of a duct.

There are numerous reliable fan manufacturers who publish complete technical data regarding their products. Men in the air-conditioning field must be completely familiar with all aspects of fan data, because the fan is a vital part of the conditioning system. Fans all cause a certain amount of noise and must be selected properly so that this noise will not be objectionable. Some installations can stand fans having a noise level several times as great as others. Since fans do the work of moving the air around the system, they must be of a capacity sufficient to overcome the resistance that acts against the air circulation.

As has been stated, the air is conveyed in ducts, the ducts ordinarily being constructed of sheet metal. The size and length of these ducts, the number and sharpness of the turns, the amount of air that is to be moved, and the characteristics of the conditioner itself determine the resistance that will be offered against the circulation of air. These must all be carefully planned and calculated before the proper fans can be chosen. On the other hand, if a fan is already available, the entire duct system including the conditioner must be planned so that it will not offer more resistance than the fan can overcome.

The duct delivering the completely conditioned air to the conditioned rooms is usually called the "supply duct." From this duct the air is delivered through various types of grilles, or outlets. A grille is available that will throw air 80 or 90 ft down a path as narrow as 20 ft in width. Other grilles will throw air for distances as short as 20 ft covering a path 40 or 50 ft in width. Also, ceiling-type grilles and grilles combined with lighting fixtures are available. The cost of grilles depends upon size and design and may be as little as $5 to $100 or more. Hence, it can be seen that the proper selection of a detail as simple as the air-outlet grilles really presents a problem that requires considerable specialized knowledge.

Thus far we have spoken of heating and cooling without any mention of their *sources*. Well water at a temperature of 55°F or colder is an ideal and inexpensive source of cooling. However, rarely are building owners so fortunate as to have such a source available. Throughout New York City, we find well water outside Manhattan quite plentiful, but usually at temperatures that run from 58 to 60° in June and rise above 65° as the season advances.

Therefore, refrigeration machinery is ordinarily used. This refrigeration machinery must be properly selected as to the size, type, and number of machines. The use of the refrigerant in this system is strictly regulated, and the so-called "safety" gas Freon is almost universally employed. These large refrigeration machines have a part similar to the "ice-cube maker" that we find in kitchen refrigerators. In some installations this part may be made up in the shape of a cooling coil, which may be placed directly in the path of the air, inside the conditioner. Frequently, in larger jobs where more than one floor is to be conditioned, this "chiller" portion of the refrigeration machine must be located within a tank so that it will chill water. This chilled water is then piped to the conditioner from which it flows back to be rechilled. Everything in connection with the refrigeration machine must be automatically controlled both from the standpoint of safety controls and from the standpoint of maintaining the proper chilling temperature.

Frequently, the heating plants in existing buildings that are being air conditioned are obsolete. In the case of new buildings, the heating plant must actually be considered a part of the air-conditioning system. Therefore, men in the air-conditioning field today must have a sufficient knowledge of heating to select the proper steam or hot-water boiler. There are available a wide variety of boilers and a wide variety of automatic coal-burning, gas-burning, and oil-burning equipment. Here again proper automatic controls must be provided both from the standpoint of safety and from the standpoint of maintaining the desired heating temperature.

We have now completed the tracing of the path of air through the supply duct system. The supply of conditioned air is distributed carefully throughout the conditioned space. The recirculated air is withdrawn from the conditioned space, connections to the recirculation duct being considerably less in number than the supply outlets. The returned-air outlets must be so located that air does not travel in a "short-cut" path from a supply outlet to a return grille.

So much for the system required to handle and condition air. There are three phases remaining, which are, from the engineering standpoint, really the heart of the problem of designing an air-conditioning system. (1) We have the problem of how much heating and cooling are to be required for a certain building. (2) After calculating the amount of heating or cooling required, how much air shall be moved about, and what must be done to this air in order to have it handle the desired heating or cooling? In addition to the amount of ordinary heating or cooling, the amount of humidity to be added or removed must be calculated; then this amount of humidity must be removed or added to the proper quantity of air. (3) A coordinated automatic control system must be provided in order to make the system complete.

Consider briefly the problem of calculating how much heating or cooling is required by a certain building. As you know, a flimsily constructed summer cottage would be hard to heat because the heat and warm air easily leak out. Similarly, in the summertime such a building easily becomes as hot as or hotter than outdoors. It is very difficult to cool such a building in the summer because warm air and heat from the sun leak inward easily.

You have probably heard a great deal in the last few years about insulation. Insulating material when built into the walls and roof of a building makes it difficult for heat to leak through inward or outward. Published information is available that shows illustrations of all ordinary types of outside walls, inside partitions, floor, ceilings, and roof construction. Allowance is made for different types of inside finishes on walls, floors, and ceilings. Counting all varieties, several hundred different types of construction are included, and for each complete information is given as to the heat leakage that will occur through such a wall.

Although you may never have thought of this, air can leak through a solid brick wall. Data are also available on the quantity of actual air leakage in addition to the heat leakage, through ordinary types of walls. From this information one can calculate the heat that will leak out of a building at 70° when the outside temperature is 0°. Similarly, one can calculate the amount of heat that will leak into a building in the summertime when the temperature is 95° outside and 79 or 81° inside the building. You can see that this is an extremely important procedure since it must be done correctly in order to arrive at the correct size of the cooling and heating plant.

In deciding the size of the cooling plant for summer requirements, two other factors must be considered. One is heat from the sun. The other is heat generated inside the building by people, electric lights, steam tables, electric motors, and other sources of heat. In restaurants the heat from steam tables, coffee urns, toasters, and grills sometimes accounts for as much as 20 or 25 per cent of the cooling equipment required for the entire installation. The sources of heat inside buildings, if sufficiently large, can sometimes be used to reduce slightly the size of the heating plant required. They invariably must be taken into consideration in choosing the size of the cooling plant. Heat from the sun need not be considered in the wintertime. The reason is that a building must be heated properly when it is zero outdoors and the sun is obscured by heavy clouds. In the summer, however, solar heat, or heat from the sun, is extremely important. In some buildings with large exposed glass window surfaces having no awnings or Venetian blinds, the sun heat may be as much as 50 per cent of the total heat that leaks into the rooms we are cooling. When sun heat strikes windows, it appears as heat inside immediately. Sun heat on heavy brick walls does not reach the inside for several hours. Thus it can be seen that air-conditioning engineers face a real problem, which must be carefully solved, in the calculation of what we call the "cooling load" and "heating load" of a building.

Earlier in this discussion we mentioned the adding of heat and humidity or the removing of heat and humidity of air in units called "conditioners." The problem of heating or cooling that affects temperature only is relatively simple. In figuring the heating or cooling load, it is simply necessary to figure the proper amount of air and the proper temperature at which to supply this air from the conditioner. However, the problem of humidity control, which is so important in summer and winter, is another story. In the winter, the outdoor air is very dry. You have all had the experience of getting an electric shock after scuffing the feet on a rug in the wintertime. This is caused by the excessive dryness of the air. The problem of the air-conditioning engineer is to add humidity to the air that is supplied to the air-conditioned space. It is important, however, that just the correct amount of humidity be supplied. You have all seen the windows of a kitchen either become wet or, if it is sufficiently cold outdoors, become covered with frost. This is because the cooking of foods has added too much humidity to the air in the kitchen. With storm windows or some of the new double-glass windows, more humidity can be added than is permissible with ordinary single-glass windows. Thus it is evident that the problem of winter humidifying is one that must be handled carefully and accurately.

In the summertime the outdoor air is very humid; in addition, human beings add humidity because of the evaporation of perspiration. This human humidity load is greater if persons are active than if they are sitting quietly. The problem of the air-conditioning engineer in the control of humidity in the summertime is therefore twofold. (1) The excess humidity must be removed from the outdoor air. (2) The humidity that has been added by human beings and by any steam appliances must be removed from the recirculated air. The final effect must be such that a properly low humidity is maintained in the air-conditioned space. For a building such as a theater or a large restaurant designed to accommodate many persons, conditions are different depending on whether the building is partly or fully occupied. The supply of conditioned air must be dried or else larger in quantity when the building is fully occupied. Of course, both in the summer and in the winter control of humidity, instruments must be provided that make the control reasonably automatic.

The study of air and the humidity in it is called "psychrometry." To master properly the subject of psychrometry and to be able to solve practical problems of this nature require probably more study than any other phase of the theoretical study in the whole subject of air conditioning. Literally thousands of individual problems exist, almost no two being exactly alike; nor is theoretical knowledge of this subject alone sufficient. One must have practical experience with commercial equipment in order to learn how closely actual results follow the theoretical design.

At several specific points we have mentioned that automatic control must be provided. Automatic control is in itself a subject for an entire book. Controls are usually actuated either electrically or by compressed air, the fundamental principle of operation being the same. Fans, compressors, and pumps are usually driven electrically and hence must be electrically controlled even though the basic temperature and humidity control system is of the compressed-air type. If the automatic controls in an air-conditioning system fail to function, the entire system breaks down and fails to produce the desired conditions in the conditioned space.

CHAPTER I

BASIC SCIENCE

Since modern air-conditioning work involves an approach from the engineering viewpoint, we must learn to think along scientific lines before attacking specific air-conditioning problems. Furthermore, the engineering problems in air-conditioning work involve a knowledge of certain fundamental scientific principles. Therefore, this chapter may be considered as a foundation upon which our structure of air-conditioning knowledge is to be built.

Chapters II to V are a more detailed study of the factors indicated by their titles. It would be impossible to go on to a detailed study of this subject without a thorough understanding of the material in this chapter.

Composition of Matter. For many years, scientists have been investigating the composition of the ordinary materials we find in the world around us. For a long time there were conflicting theories propounded by various physicists and chemists regarding the composition of matter. Today there is still disagreement among scientific men, but these disagreements are confined almost entirely to some of the modern, complicated, advanced material. The elemental concepts of chemists and physicists are now almost unanimously agreed upon. We shall have occasion to study only these well-known clearly defined principles.

All matter, whether stone, metal, wood, water, oil, or gas, is composed of small particles known as "molecules." We may define a molecule, in layman's language, as the smallest *stable* bit of matter having the identifying properties or nature of the particular substance that it composes. This means that a molecule is the smallest bit of matter which, if we take an iron molecule as an example, would bear all the identifying marks of iron and would at the same time be *stable.* In this case by "to be stable" we mean "satisfied to remain as it is."

Molecules are made up of still smaller bits of matter known as "atoms." An atom may be defined as the smallest bit of matter bearing the identity of the substance that it composes. *Note:* In this definition the word "stable" does not appear. Essentially, this is the only difference between atoms and molecules. Let us illustrate this difference a little further: One nitrogen molecule contains two nitrogen atoms. Now such a nitrogen molecule is extremely stable; not only is it contented to stay a simple nitrogen molecule, but, in addition, it is extremely difficult to break up this molecule and cause the two nitrogen atoms to do anything different from remaining united in the original molecule. On the other hand, if we have two free unconnected nitrogen atoms, they are not stable; they refuse to stay isolated nitrogen atoms. Each of two such isolated atoms immediately insists upon joining with some other atom or with several other atoms. If no other atoms were available, the two nitrogen atoms would immediately tightly join one another, forming one stable nitrogen molecule.

Both molecules and atoms are extremely small. As a matter of fact, if we try to imagine a pile of 1,000 molecules, the pile would be too small to be viewed under a microscope. The extent of scientific knowledge concerning these particles is remarkable when we stop to think that they are so small that we cannot see them at all.

When millions and millions of molecules are joined to form some substance, the behavior of the molecules depends upon the type of substance. It is known, for instance, that these molecules do not lie absolutely still as do grains of salt or sugar. Instead, the molecules are always moving more or less. Energy of motion, whether the energy of motion of a molecule, of a man running, or of a moving automobile, is the same type of energy. We define this type of energy as "kinetic energy." "Kinetic" simply means "movement" or "motion." Hence kinetic energy simply means energy of motion.

State. All known substances fall into three different states. By the state of a substance we mean *solid, liquid,* or *gas.* In a *solid* substance, the molecules are vibrating back and forth at a relatively low speed and through relatively short distances. The molecules are bound together rather closely, and their freedom of motion is restricted. This explains why we cannot pour a stone or a block of wood through a funnel as we can pour a liquid; the solid will not easily change its shape.

When a substance is in the *liquid* state, the molecules are somewhat less rigidly bound together than when in the solid state. The molecules are then able to vibrate more rapidly and possibly through somewhat greater distances than those in a solid.

Imagine two automobiles, one moving at 10 mph and one moving at 40 mph. Suppose that the automobiles collide head on with a heavy brick wall. We know that the automobile traveling 40 mph will be much more badly damaged. The reason for this is that its kinetic energy is greater than that of the more slowly moving automobile.

In the same way it is true that the molecules of a substance in a liquid state have a greater kinetic energy than the molecules of that substance in the solid state. How do we increase the kinetic energy when we change a substance from a solid state to a liquid state? If we reverse this question, it answers itself. How do we melt a block of wax or a cake of ice? The answer is, of course, by the application of heat. In other words, in order to change the state of a substance from solid to liquid, we must put heat energy into the substance, and this heat energy increases the kinetic energy of the molecules.

When substances are in a *gaseous* state, the molecules are flying at high speed. The path through which these molecules fly has a greater length than the path of molecules in the liquid state. In fact, a molecule in a gas stops moving in one direction only when it hits another molecule, when the two molecules act exactly like two moving billiard balls striking one another on a billiard table. Also, when a molecule in a gas hits a wall, such as a wall of the tank containing the gas, the molecule will bounce back just as a billiard ball bounces on hitting the cushion around the table.

As you may already have realized, it is necessary to supply heat to a substance to change it from a liquid to a gas just as in changing from a solid state to a liquid state. This heat energy is required to increase the kinetic energy of the molecules. For the molecules in the same substance, the kinetic energy is much greater in the gaseous state than in the liquid state. Also, note that the molecules are no longer bound together, not even loosely.

One important phenomenon that we shall meet again many times should be pointed out. In fact, the performance of all refrigeration machinery is dependent on this phenomenon. If we increase the pressure on a substance, we *lower its freezing point* and *raise its boiling point*. This is of special importance with regard to the boiling point of various substances.

Kinetic Theory of Gases. The change of state of a substance from a liquid to a gas is so important that it must be studied further. Scientists have agreed upon a theory, which has not yet been proved wrong, that satisfactorily explains the changing of a liquid to a gas. This theory also explains the change of state of a gas back to a liquid and the actions and properties of a substance while in the gaseous state. This theory, known as the kinetic theory of gases, is discussed below. Consider first a pure gas, not complicated by the presence of any liquid. Millions of molecules will be flying around at random, striking one another and striking the walls of the tank that contains the gas. Let us imagine that this tank is made of an elastic material or is of telescopic construction so that we can increase or decrease the size of the tank at will. As we know, air in an automobile tire is really a gas within a tank. If a pressure gauge is put on the automobile tire of a car, it will read somewhere between 25 to 35 lb, depending upon the type of tire and the amount of air forced in. Actually, this gauge does not read 25 "pounds," but "25 pounds per square inch." This means that on each square inch of the interior of the tire there is an outward force of 25 lb tending to burst the tire.

This pressure, which may be great or small, exists whenever a gas is confined within a tank or enclosure. The kinetic theory of gases states that this pressure is caused by molecules, millions of them every second striking the walls of the tank. This may be hard to imagine since, as has already been stated, molecules are very small. However, think of this comparison: A machine-gun bullet is extremely light, weighing perhaps an ounce. Certainly this weight is nothing compared with the weight of a man, and a man can easily carry several hundred machine-gun bullets. But suppose that you were given a shield so strong that machine-gun bullets could not penetrate it. Suppose that you were to hold this shield in your two hands and permit the operator of the machine gun to discharge his stream of bullets against the shield. You would, of course, be knocked down because of the pressure of this stream of small bullets hitting the shield.

To carry this comparison a little further, if the operator of the machine gun were to double the rate of firing, thus firing twice as many bullets per minute, the force tending to knock you down would be greater. Similarly, if he were to fire bullets at a reduced rate, the effective pressure on the shield would be reduced. Also, if the bullets were fired at a constant rate but were made to leave the machine gun at a higher speed, the effective pressure on the shield would be increased.

In the same way, more molecules striking upon 1 sq in. of the wall of a tank will cause an increase in pressure. If we take our imaginary tank and make it smaller, say half the original size, twice as many molecules will strike each square inch of surface and the pressure will increase. If we take this original tank and make it larger, the gas will immediately fill the increased space, but fewer molecules will bombard each square inch of the wall; therefore, the pressure will be lower. The kinetic theory of gases tells us that a gas will expand indefinitely. In other words, the compressed air (which is compressed gas) in the small tank at a gasoline station would expand to fill one of the huge tanks in which a gas company stores gas. This experiment, of course, could be carried out only provided that the large gas tank were *absolutely empty* when the compressed air was let in.

We will recall that when we wish to change a liquid to a gas we add heat to it in order to increase the kinetic energy of the molecules. Suppose we already have some gas in a tank and we add heat to this gas. This heat energy appears in the gas as increased kinetic energy of the molecules. In other words, the molecules are flying about more rapidly than before the application of the heat. If these molecules are flying about more rapidly, they necessarily have more kinetic energy. Therefore, when they strike the walls of the tank, they will strike harder than before. The result is that the pressure in the tank would increase.

To consider one more aspect of the kinetic theory of gases, let us again imagine our telescopic tank of gas. Now let us suppose that we *suddenly* collapse this tank to one-half or one-quarter its original size. As we have already pointed out, this would cause an increase in the pressure because of more molecules striking every square inch of the tank walls after the tank was made smaller. In addition, this sudden compression of the gas increases the kinetic energy of the molecules. From the previous discussion we have seen that it takes heat to give molecules an increased kinetic energy. A sudden compression of our tank of gas, which would increase the kinetic energy of the molecules, would also put additional heat into the gas. We have probably observed that an automobile-tire pump becomes heated when it is being used rapidly to pump up an automobile tire. This everyday occurrence is explained by our kinetic theory of gases. This is important to remember for further reference, because most refrigeration machinery contains a compressor that compresses gas in just this fashion.

We shall now examine the nature of *boiling* in one light of the kinetic theory of gases. We have stated that we add heat to a substance to change its state from a liquid to a gas. Let us imagine a kettle of water that is at the boiling temperature and that is ready to begin boiling. The molecules are bound together, but only loosely, and they are vibrating back and forth through limited paths. In order to cause the molecules to leave the liquid and become one of the molecules of a gas (steam), we must greatly increase the kinetic energy of the molecules. Let us consider what these molecules must do to leave the liquid and join the gas. First they must break the bond by which they are held to the other molecules. Then they must gain sufficient speed to leave the surface of the liquid and be thrown off as freely flying gas molecules having greatly increased kinetic energy. From this we can see that once we heat water to boiling temperature we must add a great deal of heat to the water in order to "boil it away," or change its state to that familiar gas, steam.

Now let us imagine the tea kettle with the water not quite at boiling temperature. In other words, the tea kettle would contain some very hot water, but no active boiling would be taking place. Under these conditions, we shall find that the inside of the tea-kettle lid will collect many droplets of water and will become completely wet. From this it would seem that some of the molecules of the water must have left the liquid state and traveled as a gas from the surface of the water to the inside of the lid. This is exactly what has happened. The kinetic theory of gases tells us that whenever we have a liquid in a partly or tightly closed container there is a small quantity of this substance in a gaseous state above the surface of the liquid. This means that a few molecules have left the surface of the liquid and are flying around space above the surface of the liquid. Why does this not continue indefinitely until a great deal of the liquid is changed to a gas? The reason is that the molecules of the gaseous portion are flying around striking the top and side walls of the container and also the surface of the liquid. If we put a liquid in a closed container in which there is absolutely nothing (perfect vacuum) at the start, some of the molecules in the liquid would immediately leave the surface and fly about in a gaseous state above the surface of the liquid. This would happen almost instantly. After this a few molecules would continue to leave the surface of the liquid, but remember that the molecules in the gas now flying about are now and then striking the surface of the liquid. A good many of these gaseous molecules "dive" right back into the liquid when they strike the surface (see Fig. 1).

Hence the quantity of gas upon the surface of the liquid would increase for a few seconds. It would increase until the number of molecules leaving the surface of the liquid was exactly equal to the number of molecules striking the surface of the liquid and changing back to liquid state. This condition is called "equilibrium" between a liquid and its vapor.

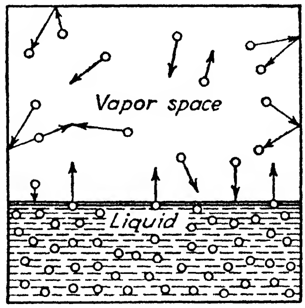


FIG. 1. At equilibrium the gas molecules return to the liquid phase as rapidly as the liquid molecules escape.

This is perhaps a new idea. Ordinarily we simply think that some of the liquid becomes a gas and then stays a gas above the liquid with no new gas being formed. As we have seen from the preceding discussion, some molecules of the liquid continue to leave and become molecules of the gas, but at the same time an exactly equal number of molecules of the gas reenter the liquid and change back to the liquid state.

Importance of Gas Laws and Change of State. You may have begun to wonder why we are interested in these fundamental scientific materials. Many chapters will be encountered later on that will require an understanding of these basic principles. For example, steam heating is an everyday occurrence, familiar to all. It is a vital factor in a good deal of winter air conditioning. If you stop to think, you will realize that, in steam heating, water is changed from a liquid to a gaseous state in the steam boiler down in the cellar. The attempt is then made to pipe it to radiators or heating coils, in the hope that very little of it will change back to a liquid state in the pipe lines. In the radiator or heating coil, steam is changed back to a liquid. This involves the theoretical material just discussed. As most of you know, the humidity in the air is one of the major concerns in both summer and winter air conditioning, and it is really water (or steam). In order to handle the control of heating and the control of humidity, it is absolutely necessary to know how steam and humidity are going to behave.

As you will learn later, in refrigeration machinery the refrigerant is a substance that changes from a liquid to a gas and back to a liquid once or twice each time that it travels around through the system. Also, the pressure and temperature of this refrigerant change several times for each trip around the system.

Finally, air is a gas or, more specifically, a mixture of several gases. In air-conditioning work not only do we try to control all factors regarding the air in rooms and buildings, but also we must move a great deal of air around through ducts, fans, and pieces of conditioning equipment. We are continually changing the temperature of air, often by a great deal. We must change its pressure somewhat, also. Hence it is essential that we understand what happens to a gas when we change it in this manner.

Physical Mixtures. We have just said that air is a mixture of several gases. In scientific work we have to deal with *mixtures* of two substances and also with *chemical combinations* of two substances. These are entirely different, and it is vital that we understand these differences. Let us define a physical mixture: "When two or more substances mix, the individual molecules of each substance remaining the same and not losing their identity, we have a physical mixture." If we mix salt with brown sugar, no matter how thoroughly we may stir them together it would still be possible to pick out grains that would be *pure salt* and grains that would be *pure unchanged sugar.* When we mix two gases, as oxygen and nitrogen are mixed in air, we cannot of course pluck out single nitrogen or oxygen molecules. This is only because the molecules are too small. However, the individual *unchanged* oxygen and nitrogen molecules are present in the mixture just as the grains of salt and brown sugar remain unchanged when these substances are mixed.

Another illustration of a familiar mixture is the ordinary alcohol used as an antifreeze for automobile cooling systems in winter. This is possible because the water and alcohol mix completely. Oil could not be used instead of alcohol because it does not mix with water. However, when water and alcohol are mixed, they mix thoroughly, and no new substance is formed. By putting a mixture of water and alcohol through the proper apparatus, it is possible to recover both the water and the alcohol in pure form. This shows that the water and alcohol *mix thoroughly, i.e.,* both the water and the alcohol have retained their original natures and have not changed chemically.

Chemical Combinations. Chemical combinations differ greatly from physical mixtures. A chemical combination may be defined as follows: "When two or more substances mix, the individual molecules of each substance combining with one another *to form new molecules* of *a new substance,* losing their original identity, we have a chemical combination." For example, consider air and gas burning in an ordinary kitchen gas stove or any other type of gas burner. You are probably familiar with the slotted openings through which air is drawn to mix with the illuminating gas. At this point and through the pipe to the point where flame begins there is a physical mixture of air and illuminating gas. However, at the point where the flame begins the oxygen in the air combines chemically with the illuminating gas to form a new chemical substance. Beyond the flame it is impossible to identify any of the original oxygen or any of the original illuminating gas. (The inert nitrogen in the air passes through the flame unchanged.) For the moment the nature of the new chemical substance that is formed by the combination of the oxygen and the illuminating gas will not be considered.

It is possible to mix hydrogen and oxygen gas in the proper proportions and cause them to combine, forming ordinary water. In this case we start with gas molecules of hydrogen and oxygen; when they combine, they form water in a gaseous state (steam). This steam then changes its state and becomes liquid water. The molecules of the liquid water are, however, the same as the molecules of the steam. These new molecules of water are composed of one hydrogen molecule combined with half of an oxygen molecule. As we know, the water molecules bear absolutely no resemblance to the original hydrogen and oxygen molecules. This is another illustration of a chemical combination. Note that it is also possible to mix hydrogen and oxygen gas only physically. Special arrangements must be made in order to cause them to combine chemically.

Gas Laws. All the various gas laws that we shall consider in Chaps. III and IV depend upon our having either a single gas or two or more gases combined as a physical mixture. The preceding two sections should have made clear the reason for this. For instance, in the making of water by a chemical combination of hydrogen and oxygen, several cubic feet of the original gases may combine to form only a portion of a teaspoonful of liquid water. Thus we see why a chemical combination in this case disturbs any laws we might set up regarding gas volume.

To repeat, our gas laws apply in dealing with one gas only. If two or more gases are involved, gas laws will apply if we have a physical mixture of the several gases. *A gas law will not apply if we have a chemical combination of the several gases.*

In all the gas laws, we speak of a nonexistent gas that we call a "perfect gas." This imaginary concept of a perfect gas is necessary because all the actual gases we know fail to conform with our gas laws. On the other hand, ordinary gases—steam, air, and many other refrigerants—at the pressures and temperatures we deal with are nearly perfect gases. For all practical purposes we may consider that they do conform to the gas laws.

In Chaps. III and IV we shall study five gas laws. These are Avogadro's law, the pressure-volume law, the temperature-pressure law, the temperature-volume law, and Dalton's law.

Chemical Symbols. We shall deal with only a few of the many substances that have been classified by chemists. For convenience we find it desirable to use the chemist's language in dealing with these substances. For this purpose chemists have invented what might be called a special "shorthand." The following are a few chemical symbols that we shall encounter from time to time:

H	Hydrogen	Hg	Mercury
O	Oxygen	Fe	Iron
N	Nitrogen	Cu	Copper

From this list we see that in some instances the chemists have simply taken the first letter of the name of the element and called it the symbol for that element. In other cases, such as the last three symbols, they have taken the first two letters of the Latin word for the element. Some chemical symbols apparently have been developed according to no system whatever, but these we must accept inasmuch as they are accepted by all chemists.

When several of these simple elements, such as those listed above, are chemically combined, we can write a symbol for the newly formed substance by a proper combination of chemical symbols. However, before doing this let us note the proper way to write the chemical symbol for *one molecule* of some substances we shall deal with:

H_2	One molecule of hydrogen
N_2	One molecule of nitrogen
O_2	One molecule of oxygen
C	One molecule of carbon
S	One molecule of sulphur
Hg	One molecule of mercury

You should memorize this list, for these chemical elements will frequently be encountered. The subscript 2 used in connection with hydrogen, oxygen, and nitrogen means that *one molecule* of each of these substances *contains two atoms.* In the case of carbon, sulphur, and mercury *one molecule* of the substance contains only *one atom.*

The formulas and names of three chemical combinations, *i.e.*, substances formed by two elements that have combined chemically, that we shall encounter frequently are given below:

CO	Carbon monoxide
CO_2	Carbon dioxide
H_2O	Water

It will not be necessary to memorize these at this time as this material will be dealt with further in this chapter.

Molecular Weights. At this point we should recall the definition of the word molecule given in the first section of this chapter. This definition is: "The smallest *stable* bit of matter having the identifying properties or nature of the particular substance that it composes." Again we should note the word "stable" in this definition.

Pure basic substances that cannot be broken down into two or more still simpler substances are called "elements." The chemical substances listed on page 20, are all chemical elements. When we have a combination of two or more chemical elements, the resultant new substance is called a "chemical compound." Carbon monoxide, carbon dioxide, and water are illustrations of a chemical compound.

It will be recalled that we said that there are two atoms in an oxygen, a hydrogen, or a nitrogen molecule. Chemists know that a hydrogen atom is the lightest in weight of any known atom. Since there are only two hydrogen atoms in a hydrogen molecule, it is true that a hydrogen molecule is the lightest molecule that we know of. Of course, we cannot weigh a molecule

TABLE 1

Element	Symbol for one molecule	Molecular weight
Carbon	C	12
Hydrogen	H_2	2
Nitrogen	N_2	28
Oxygen	O_2	32
Sulphur	S	32
Mercury	Hg	200

of any substance on an ordinary scale. However, chemists have been able to measure the "relative weight" of molecules of different substances. By this we mean that they have been able to determine that an oxygen atom weighs sixteen times as much as a hydrogen atom. Hence an oxygen molecule weighs sixteen times as much as a hydrogen molecule (because one molecule of each gas contains two atoms). Similarly, it is known that one nitrogen atom is fourteen times as heavy as a hydrogen atom. One sulphur atom is thirty-two times as heavy as a hydrogen atom, and so on.

Because we cannot actually weigh atoms or molecules on a scale, the chemists have said, "Let us call the weight of one hydrogen atom 1.00, since it is the lightest atom we know. Now since one hydrogen molecule contains two atoms, its weight will be 2.00. We know that an oxygen molecule contains two atoms each sixteen times as heavy as a hydrogen atom, and so we shall call the weight of an oxygen molecule 32." These numbers that have been assigned by the chemists are called "*atomic* weights" in speaking of one *atom* and "*molecular* weights" in speaking of one *molecule*. Avogadro's law, which we have mentioned, also involves an understanding of molecular weights.

The molecular weights that we need to know are shown in Tables 1 and 2.

Let us figure out how the molecular weight of carbon dioxide is found to be 44. From Table 1, we see that the molecular weight of carbon is 12, while the molecular weight of oxygen is 32. From the formula for carbon dioxide we can see that one molecule of carbon dioxide contains one molecule of carbon and one molecule of oxygen. Hence we add 12 plus 32 and get 44, as the molecular weight of the combined form.

TABLE 2

Chemical compound	Symbol for one molecule	Molecular weight
Carbon monoxide	CO	28
Carbon dioxide	CO_2	44
Water	H_2O	18

Simple Chemical Equations. In this section we shall present some chemical equations that show in the chemist's language the chemical combination of two substances to form a new substance. For illustration we are taking simple equations, but nevertheless ones that we shall encounter in a practical way later.

$$\underset{\text{(One molecule)}}{C} + \underset{\text{(one molecule)}}{O_2} = \underset{\text{(one molecule)}}{CO_2} \tag{1}$$

Equation (1) represents the burning of carbon (or charcoal), which is actually a combination of carbon with oxygen. If there is enough oxygen so that each carbon molecule can obtain one whole oxygen molecule with which to unite, the result is the formation of carbon dioxide, as shown in the equation

$$\underset{\text{(One molecule)}}{H} + \underset{\text{(one-half molecule)}}{\tfrac{1}{2}O_2} = \underset{\text{(one molecule)}}{H_2O} \tag{2}$$

Equation (2) illustrates the formation of water by a combination of the proper quantities of hydrogen and oxygen gas. Since $\frac{1}{2}O_2$ is really *one atom* of oxygen, we can use O instead of $\frac{1}{2}O_2$, rewriting Eq. (2) thus:

$$H_2 + O = H_2O \tag{3}$$

In the event that carbon burns in the presence of insufficient oxygen, we do not get the result shown in Eq. (1); instead, carbon monoxide gas is formed, as indicated in Eq. (4).

$$C + O = CO \tag{4}$$

$$CO + O = CO_2 \tag{5}$$

Carbon monoxide gas, however, is a perfectly good fuel and may be burned with oxygen, forming carbon dioxide as indicated in Eq. (5).

Many fuels contain sulphur, the chemical symbol for which is S, as we have already stated. When sulphur burns with oxygen, the result is a gas known as sulphur dioxide, a gas having an extremely irritating effect when breathed by human beings. This process is represented by Eq. (6).

$$S + O_2 = SO_2 \tag{6}$$

$$SO_2 + H_2O = H_2SO_3 \tag{7}$$

When sulphur dioxide and water are brought together, the two compounds do not mix physically, *i.e.*, they do not mix the way water and alcohol mix. The sulphur dioxide and the water combine chemically to form a corrosive acid known as sulphurous acid. This process of forming sulphurous acid is shown in Eq. (7).

PROBLEMS

1. Of what is matter composed? When we say kinetic energy, what type of energy do we mean?
2. All matter exists in one of three different states. What do we call these three states? When a substance changes state, in what way is heat involved? What happens to the kinetic energy of the molecules?
3. Explain the kinetic theory of gases in your own words. Illustrate your explanation by tracing the change of state of water from ice to liquid water and finally to steam.
4. Distinguish between a physical mixture of two substances and a chemical combination of two substances.
5. Write the chemical symbol for one molecule of each of the following substances, showing also the molecular weight of these substances:

Elements	Chemical Compounds
Carbon	Carbon monoxide
Hydrogen	Carbon dioxide
Nitrogen	Water
Oxygen	
Sulphur	
Mercury	

6. Write the chemical equation illustrating the combination of one molecule of carbon with one molecule of oxygen. What is the name of the new substance formed as a result of this combination?
7. Write the chemical equation that illustrates the combination of two molecules of hydrogen and one molecule of oxygen. What is the name of the new substance formed, and how many molecules of this new substance are formed?

CHAPTER II

GRAPHS AS ENGINEERING TOOLS

One of the simplest illustrations of a graph, although it is not of an engineering nature, is the graph made by mothers of the weight of their children. The hospital charts kept by nurses showing the patient's

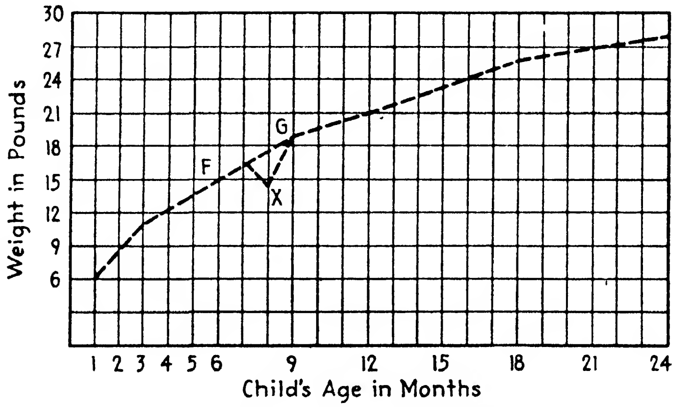


FIG. 1.—The child's weight from week to week is not as significant as his weight from month to month.

temperature and pulse rise during illness are also examples of commonly found nonengineering graphs. Graphs of this type are illustrated in Figs. 1 and 2.

The important thing to recognize about these graphs is that they are made by plotting a number of isolated points. Nothing is shown as to the value of the variable (the "variable" is the child's weight in Fig. 1 and the patient's temperature in Fig. 2) between the plotted points.

For instance, according to Fig. 1, we must assume that the child was not weighed during the period between the ages of six and nine months. *We cannot* assume that at the age of 7½ months his weight was 17 lb. Glancing at the dotted line in Fig. 1, it would seem as though the child's weight probably was about 17 lb when the child was 7½ months old. However, our known data are only at points *F* and *G*, at ages six and nine months. It would have been possible for this child to have been ill for 2 weeks commencing at the age of seven months. This might have caused his weight to drop, as shown by the broken line dropping to point x. So long as he recovered from this illness and regained weight normally, his weight could still be 19 lb at the age of nine months.

In a graph of the type shown in Fig. 1, we have data taken at a few isolated intervals with *no data* between the plotted points. In such a graph it is really inaccurate and unjustifiable to draw a line through the points that we plotted because we can read nothing with certainty from our line between the plotted points. When we do draw a line on such a graph, the line serves only to guide the reader's eye from point to point along the graph. Let us repeat once more that we are not justified in reading any information from lines between two plotted points.

Graphs of the type just discussed are called "discontinuous" graphs. In engineering work, most of the information shown on graphs is not of this

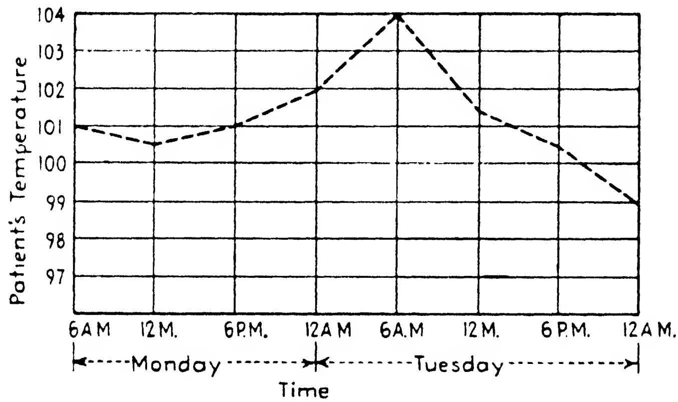


FIG. 2.—Patient's temperature chart.

discontinuous nature. Ordinarily even though a graph that may have been made from 10 or 15 plotted points, we can use the lines in between the plotted points with certainty that the values read will be accurate. Dependable graphs of this nature are called "continuous" graphs, or graphs of continuous functions.

The Fundamental Nature of Continuous Graphs. Except for the so-called "bar" graphs, which are used to show how many bushels of potatoes, pounds of butter, dozens of eggs, etc., are sold and for other economic and commodity market records, all graphs are based upon having curved or straight lines to tell a story or reveal information. Frequently both straight and curved lines and combinations of the two may appear on a single graph.

In the simplest sort of graph we must have quantities each of which is changing. For instance, you know that the faster you drive an automobile, the greater the gasoline consumption. Here are two quantities: (1) *speed* in miles per hour and (2) gas mileage in miles per gallon of gasoline. For such a graph we, of course, assume that the automobile is driven in high gear only and we shall consider speeds from 10 mph upward. Let us suppose that we had tested a certain make of automobile at speeds of 10, 15, 20 mph, etc., up to 70 mph. We can show the results of our test, as indicated in Table 1.

TABLE 1

Speed, mph	Miles per Gallon of Gasoline
10	13
15	15
20	16.5
25	18
30	19
35	19.7
40	19.5
45	19.0
50	18.5
55	17.5
60	16.0
65	14.5
70	12

Table 1 illustrates the results of our test on this automobile. However, it does not provide a quick and easy means of determining the gas mileage at any speed at all between 10 and 70 mph. Furthermore, it does not give

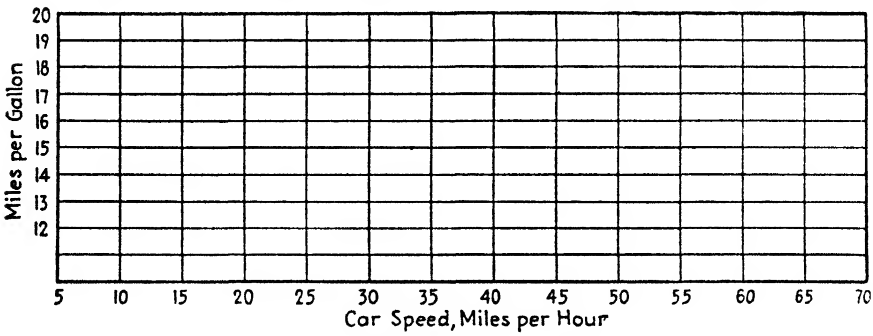


FIG. 3.—Simple coordinate chart.

the reader at a glance a complete picture of how the gas mileage changes as the speed of the car is increased. To get such a picture, the reader must study the table and make a graph in his own mind.

In starting to make a graph of the results of our test on the automobile, we first draw our two scales, or axes, along which we are going to plot (or mark off) the values given in Table 1. In speaking of a graph, each such scale is called an *axis* of the graph, and in speaking of several such scales the plural of axis is used, which is axes. In simple graphs such as this, the two axes are ordinarily two lines at a 90-deg angle, as shown in Fig. 3. Figure 3 shows just the two axes with the scales marked along them.

We now place Table 1 before us so that we can begin "plotting our curve."

The first value in our "mph" column is 10; therefore, on the bottom axis of the graph place a pencil opposite 10 mph then run vertically up the paper until you are opposite the value 13 on the miles per gallon axis. At this point put a heavy black dot or a small x. Now in a similar manner plot the

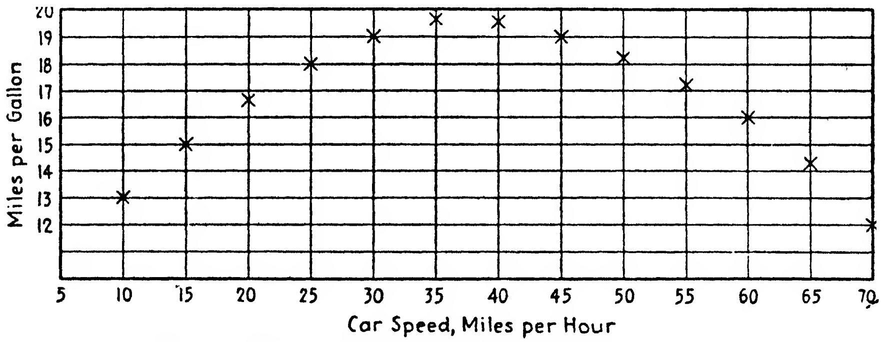


FIG. 4.—Best gasoline economy lies between 30 and 45 mph.

point for 15 mph, and so on, for every car speed up to 70 mph, which is the highest speed covered by our test. The graph now looks as shown in Fig. 4.

In order to complete our graph we now have only the problem of drawing a smooth careful curve through the points that we have plotted. It is difficult to draw freehand as smooth a curve as is shown in Fig. 5. How-

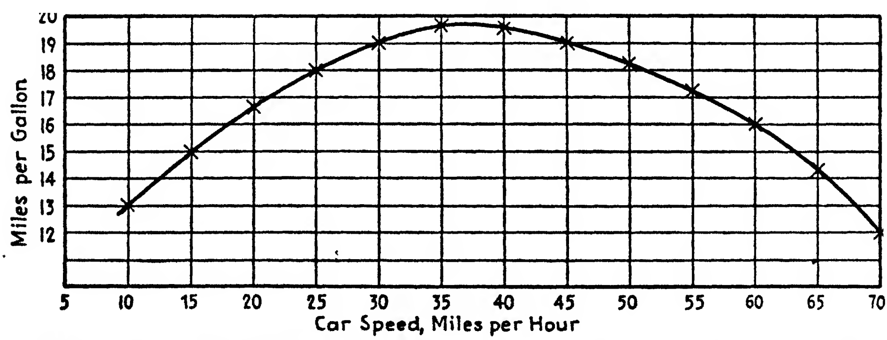


FIG. 5.—Continuous curve shows gasoline consumption at intermediate car speeds.

ever, an instrument known as a "French curve," which is found in every drafting room, is helpful in drawing smooth curves of all shapes.

Compare the finished draft, as shown in Fig. 5, with Table 1. You will surely agree that the graph represents the whole picture at a glance much better than the table. From the graph it is readily seen that the best gasoline mileage would be at car speeds between 30 and 45 mph. It is evident that above 60 mph the gasoline mileage gets worse very sharply.

This simple problem of graphing data in connection with an automobile fuel-economy test is presented here instead of an air-conditioning graph because almost everyone is familiar with automobiles and their problems. Furthermore, engineers employed by the automobile manufacturers spend many hours each year making tests such as this.

One more fact should be pointed out about the data in Table 1 and the finished graph in Fig. 5. Because of the nature of an automobile engine and because of the reasons that make gasoline mileage change, engineers know that this graph is a continuous function. As we remarked previously, most engineering graphs are continuous functions. Therefore gasoline-mileage figures for 13, 28, 42½ mph, or any other car speeds may be read between our actual plotted points. In other words, engineers know that if

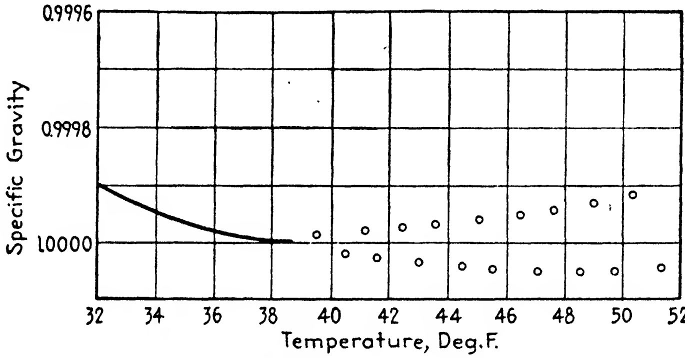


FIG. 6.—Specific gravity varies with temperature.

the gasoline mileage is 19.5 at 40 mph and 19.0 at 45 mph, then the gasoline mileage figure must be between 19.5 and 19.0 at car speeds between 40 and 45 mph.

Because this is a graph of a continuous function, an interesting question is raised. Suppose that it is possible to drive this particular car 7 mph and 75 mph in high gear. These points were not covered in our tests, so we do not have data for them in Table 1. The question is, are we justified in extending the ends of our curve in order to estimate gasoline-mileage figures for these car speeds not shown? For your own interest, extend the curve in Fig. 5 by drawing a freehand pencil line and see what values you estimate for the gasoline consumption at 7 and 75 mph.

The process of extending the curve is called "extrapolating." This simply means extending the curve of a graph beyond either or both end points. For the graph of our automobile data, automotive engineers would tell us that the values we estimated might be correct if we extrapolated carefully. However, automotive engineers could tell us this only because they have tested many automobiles and they *know the direction* of the curve from beginning to end. In other words, they have such complete knowledge of the data from which this curve was made that they could guess accurately different points beyond the given data.

Does this mean that we could take any ordinary curve that we may be using and extrapolate to get values not shown? The answer to this question is *emphatically* "No." We can extrapolate curves only when we are thoroughly familiar with their nature and when we know their *general direction* beyond the points for which we have exact data. Let us consider one example that shows the errors that can be incurred if we try to extrapolate a curve without knowledge of its general direction beyond our data. Let us assume that we have taken test data regarding the specific gravity of water at different temperatures, the results of which are shown in Table 2 and then graphed in Fig. 6.

TABLE 2. SPECIFIC GRAVITY OF WATER AT VARIOUS TEMPERATURES

Temperature, °F	Specific Gravity
32	0.9999
34	0.99995
36	0.99998
38	1.0000

This table does not represent actual test data to the degree of accuracy indicated by the specific gravity figures.

Take a pencil and continue the curve shown in Fig. 6. In other words, try to extrapolate this curve. In order to observe the general shape of this curve, our extrapolated line would have to lie somewhere between the two lines of chain dots shown in Fig. 6. Just where the extrapolated line would lie in between these limits would have to be determined by the extrapolating

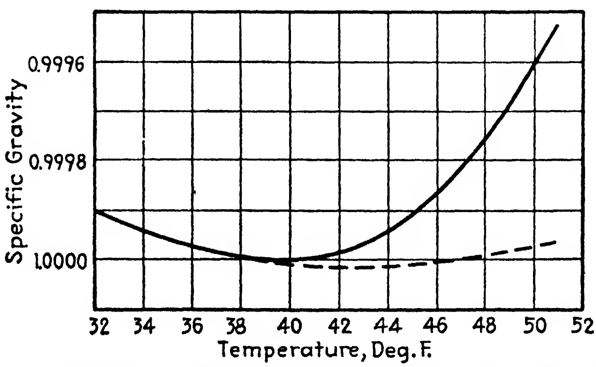


FIG. 7.—The change in specific gravity is greater above 45° F than below.

ability (or guessing ability) of the person doing the work. In any event, the extrapolated curve would show a specific gravity value at a temperature of 50° which would be just slightly above or below 1.000. The error in this case of extrapolation is very great. Figure 7 made from Table 3 shows complete data from a test made at temperatures up to 51°.

TABLE 3.—SPECIFIC GRAVITY OF WATER AT VARIOUS TEMPERATURES

Temperature, °F	Specific Gravity
32	0.9999
34	0.99995
36	0.99998
38	1.0000
40	1.0000
42	0.99998
44	0.99994
46	0.99986
48	0.99975
50	0.99960
51	0.99952

This table does not represent actual test data to the degree of accuracy indicated by the specific gravity figures.

A typical extrapolated curve is shown by a broken line on Fig. 7. Note how much this extrapolated curve differs from the correct values. This example has been presented to emphasize the fact that we can extrapolate curves only from one end or the other when we are certain of the general direction that the extended line will take.

Typical Sample Engineering Graphs. The graphs in Figs. 8 to 10 illustrate engineering data that are ordinarily presented graphically instead of in

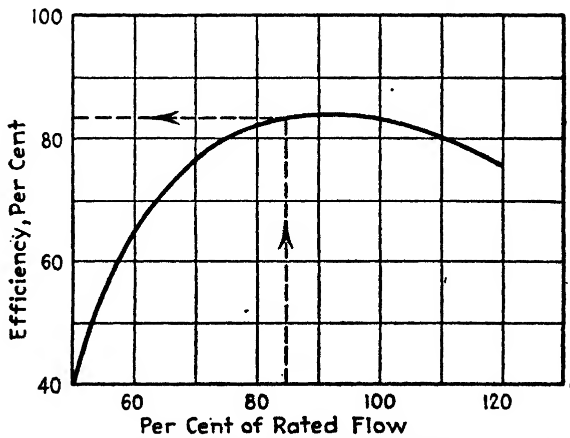


FIG. 8.—Above 90 per cent of rated flow, efficiency drops off.

tables. Each of these graphs shows a guideline marked with arrows. These guidelines show the method of reading each graph.

Note that in Fig. 10 our vertical axes show three different scales. The left vertical axis shows a scale that refers to curve A. The right-hand vertical axis shows a scale at the bottom that refers to curve B, and at the top it shows a scale that refers to curve C. Follow the guideline to understand the method of reading this graph.

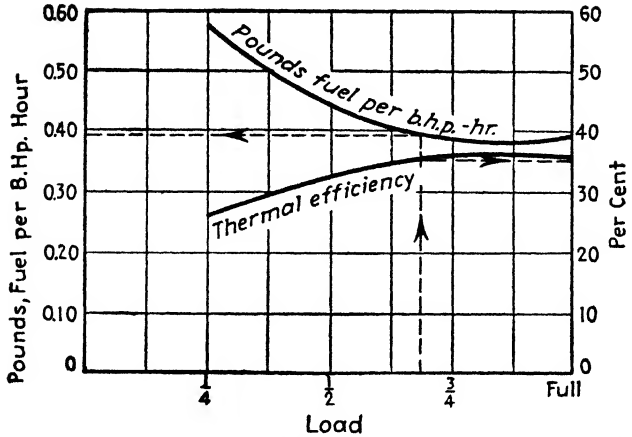


Fig. 9.--It is poor economy to run boilers for long periods of time under part loading.

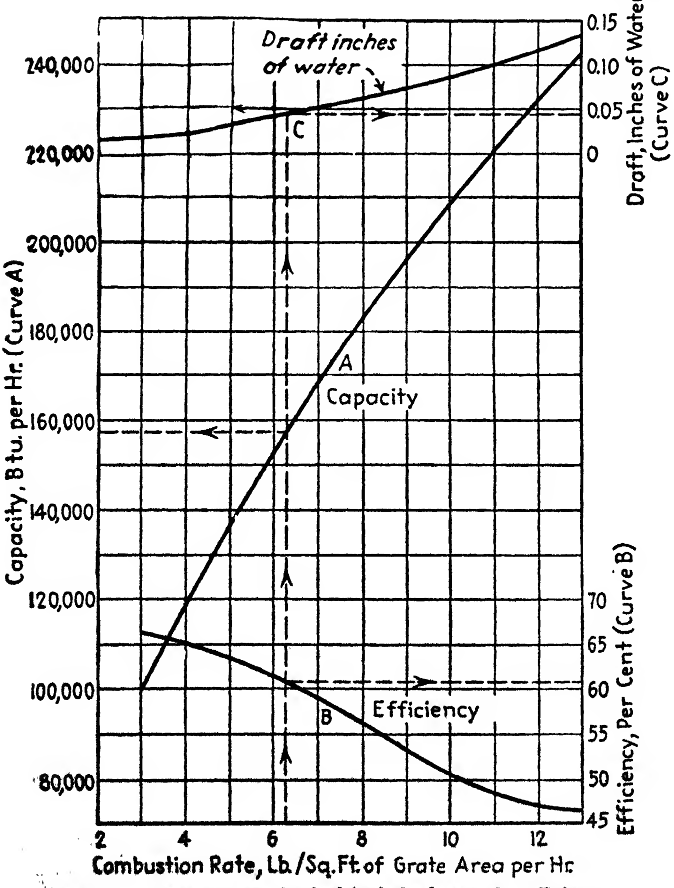


FIG. 10.—The thicker the fuel bed the lower the efficiency.

Figures 8 to 10 are typical engineering graphs. Data concerning furnaces that burn solid fuel are frequently presented in a manner shown in Fig. 10.

Typical Air-conditioning Engineering Graphs. In the selection of air-conditioning equipment, it is almost always necessary to refer to graphs in order to be able to get all the necessary information to select the proper

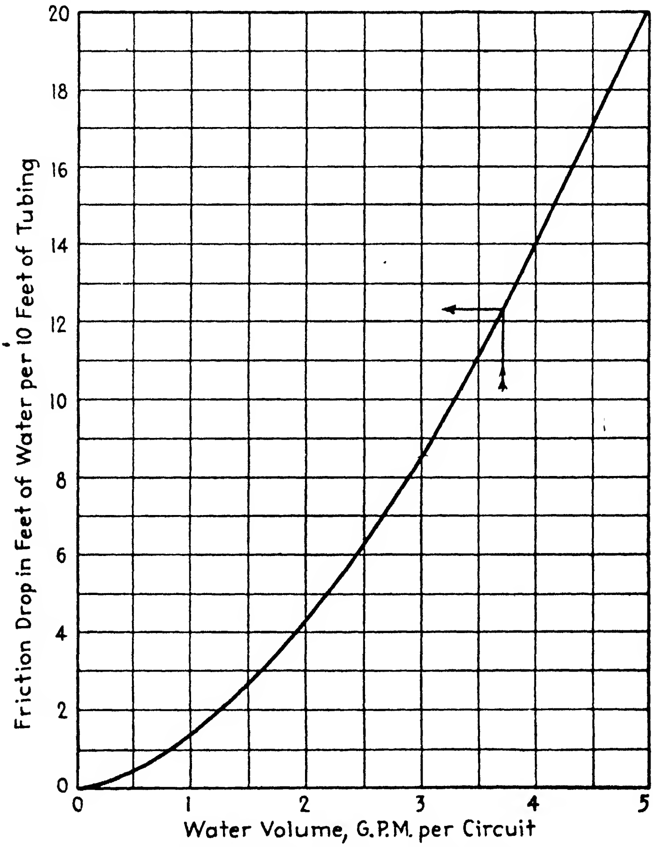


FIG. 11.—Friction drop varies approximately as the square of water volume.

equipment. At the present time we cannot make a detailed study of equipment-selection graphs. Nevertheless, we do want at this time to learn the mechanics of using the equipment-selection graphs themselves. Figures 11 and 12 shows two graphs which we are reproducing by the courtesy of Peerless of America, Inc. Figure 11 relates to the calculation of the resistances or friction of water flowing through a water-cooling coil. Let us do a test problem using this graph.

Suppose that we have a water-cooling coil with four water circuits requiring a total water flow of 12 gpm. Since our coil has four circuits, we divide

12 by 4, and a water flow of 3 gpm through each circuit results. Place your pencil at the point marked 3 gpm on the bottom axis of the graph in Fig. 11, run up to the curve, then continue to the left to the scale on the left axis. There you read 8.5 ft as the friction drop *per* 10 *ft* of tubing. As far as the use of the graph is concerned, the problem is now solved. In an actual case you would have to look up the number of feet of tubing in each

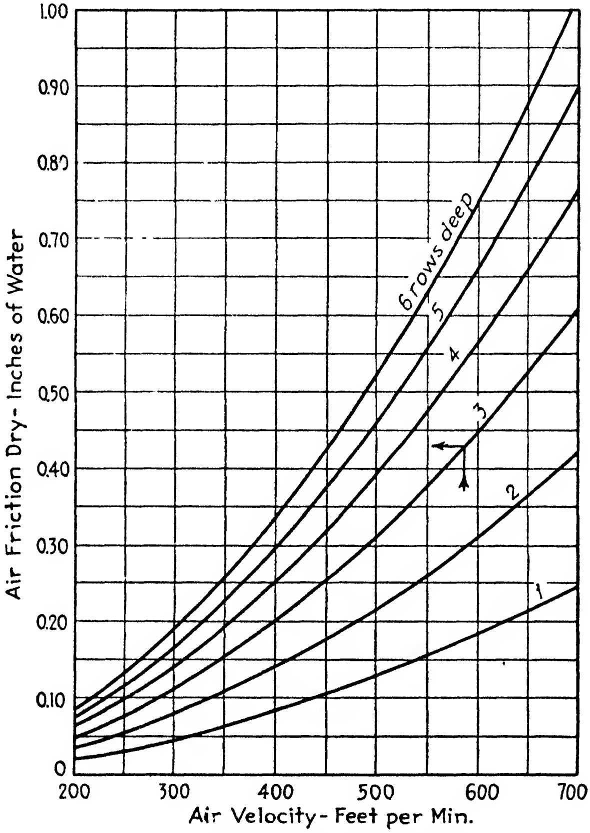


FIG. 12.—Air friction varies approximately as the square of air velocity.

circuit of your cooling coil. Suppose that this value were found to be 20 ft of tubing. The graph indicates that for *each* 10 *ft* of tubing there is a friction drop of 8.5 ft. Hence for 20 ft of tubing the friction drop would be twice 8.5, or 17 ft friction drop. (At this time we shall not define "friction drop" or other terms that appear on these sample graphs. These terms will be explained clearly later when we study the type of equipment to which the terms relate.)

Figure 12 is shown because it illustrates one complicating factor in connection with graphs. Note that there are six different curves plotted in

Fig. 12. This is because the particular kind of cooling coils to which Fig. 12 relates may be made with one to six rows of tubes. The one-row coil is about 1 in. thick, whereas the six-row coil is about 6 in. thick. Cooling coils of this nature resemble automobile radiators, and air is blown through them just as air is blown through an automobile radiator. The value shown on the vertical axis of Fig. 12, called "air friction," represents the resistance to blowing air through such a coil. Keeping in mind the comparison to an automobile radiator, you can understand that it would be easy to blow air through a coil only 1 in. thick. On the other hand, as the coil is made thicker, it becomes harder to blow air through the coil. Graphs of the type shown in Fig. 12 are presented by manufacturers to enable an air-conditioning engineer to calculate the frictional resistance that he must overcome in order to blow air through the cooling coil he has selected.

In order to use Fig. 12, *only one of the six curves is used at a time.* To read a value from Fig. 12, pick an air velocity on the bottom axis with which to start. Then assume a coil one to six rows deep. Run your pencil up vertically from the air velocity you have chosen until you reach the curve corresponding to the number of rows of depth in your coil. Then run your pencil to the left-hand vertical axis and read the value of the frictional resistance that will be encountered under your operating conditions.

An even more complicated graph is shown in Fig. 13.

In order to use Fig. 13, it is necessary for an air-conditioning engineer to know several things *before* referring to the graph. These quantities are as follows:

1. The wet-bulb temperature of the entering air. This is shown on the bottom axis of the graph at the left-hand portion of the graph. For New York City, 75° is a standard maximum design value for this factor; so the 75° line is printed in heavy black ink.

2. The condensing temperature. This is a factor relating to the refrigeration machine. For many ordinary conditions, 110° is a standard value for this factor.

3. The Btu per hour. This value represents the heat that the refrigeration machinery is removing from the substance it is refrigerating. It would be calculated before trying to select an evaporative condenser.

Let us assume that we wish to select an evaporative condenser for our three values given above. Let us take 75° for (1), 110° for (2), and for (3) a value of 200,000 Btu per hr. In order to select the proper evaporative condenser, read upward from 75° on the wet-bulb scale to the diagonal line marked "110° condensing temperature." Then read *horizontally* to the right until you reach the curve marked 200,000 Btu per hr. Then read vertically downward, and you will see that machine No. PWS-20 would be required. There are guidelines shown on Fig. 13 for other sets of conditions. These guidelines should help you in the use of this chart.

Study Fig. 13 until you thoroughly understand it, and do not leave the subject of graphs until this chart is familiar to you. At this time, since we have not yet become familiar with any of the engineering problems of air conditioning and have not encountered the equipment used to solve these problems, we shall not show additional equipment-selection graphs.

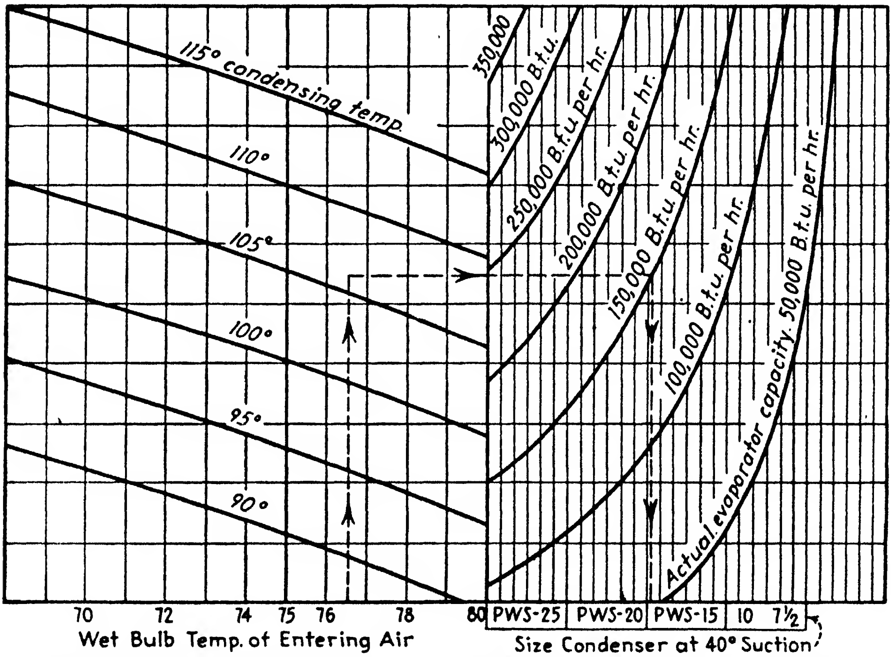


FIG. 13.—The air leaving an evaporative condenser is more humid than that entering.

The purpose of this chapter is twofold. (1) We want you to become familiar with graphs and learn how to read graphs that have been made by others. (2) We want you to learn how to make graphs so that you can present data that you may wish to explain to others.

Construction of Graphs from Test Data. When you record a table of data in the laboratory, the original data sheet used in the laboratory should be preserved. On this sheet no erasures should appear. If the member of the laboratory group who is recording the data should write down a wrong number, this should be crossed out and the correct number inserted with a note explaining the reason for so doing. The next step is to organize the figures contained in this rough data sheet, making corrections for any known instrument error. For instance if you are using a thermometer that you know reads 0.5°F too high, you must subtract 0.5°F from all temperatures that are read when using this particular thermometer. After doing this, your data should be available in a neat table.

TABLE 4

Centrifugal fan tip speed	Noise level, decibels	Point marks (See Fig. 14)
2,000	1.0	*a*
2,500	3.0	*b*
3,000	4.5	*c*
3,500	5.5	*d*
4,000	7.0	*e*
4,500	7.4	*f*
5,000	8.5	*g*
5,500	7.0	*h**
6,000	9.5	*i*
6,500	10.3	*j*
7,000	12.5	*k**
7,500	10.8	*l*
8,000	11.0	*m*
8,250	11.4	*n*
8,500	11.2	*o*
8,750	11.5	*p*
9,000	11.7	*q*
9,500	11.6	*r*
10,000	12.0	*s*

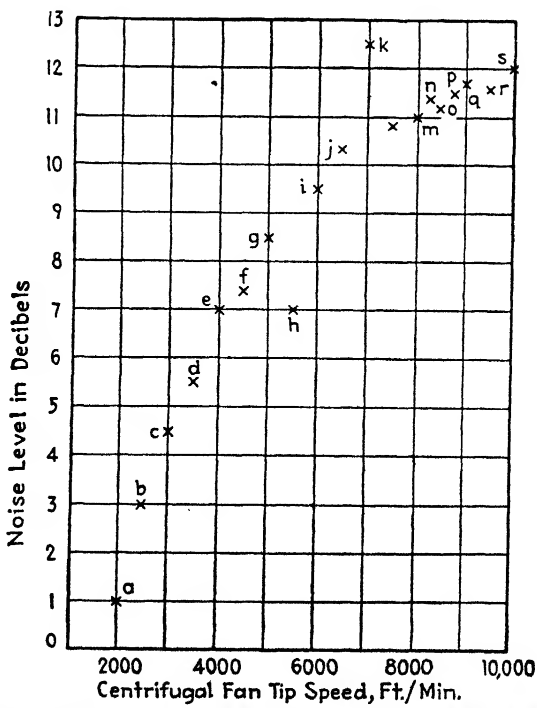


FIG. 14.—High tip speeds can be justified in industrial installations.

After studying this table you will decide upon the general size and shape of your graph and then draw the axes of the graph, marking the scales on each axis. Starting with the first value in your table, you will then plot a point on your graph representing each corresponding pair of values in your table. Let us suppose that Table 4 represented your finished corrected tabulation of your data.

Your graph, after plotting points from Table 4, would then appear as shown in Fig. 14. You can see at once that not all these points lie along a

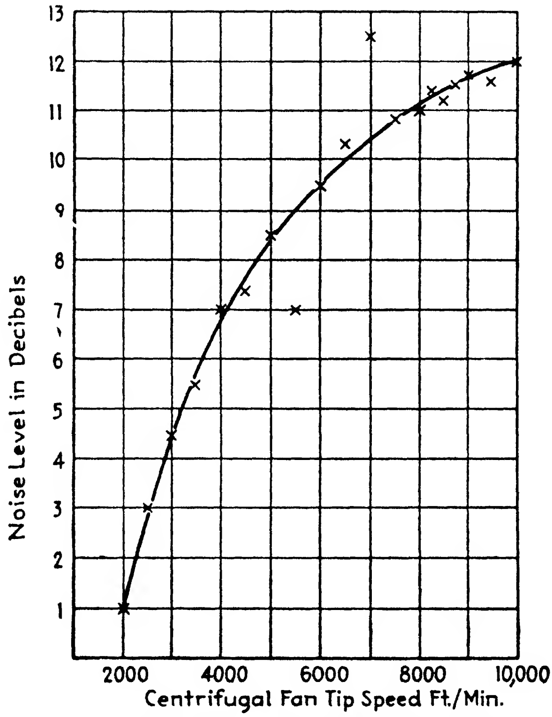


Fig. 15.—The duct system resistance will also determine the tip speed.

smooth curve. In order to draw your curve, you first reject points h and k (marked * on the table). These points are obviously incorrect. They must have been caused by some mistake in conducting the experiment. When we see at a glance that one or two points are badly in error, we should not let them influence our curve. Then take a French curve, the familiar drafting-room instrument mentioned before, and draw a smooth curve that seems to represent *good average values* of your data. This would give you a finished graph, as shown in Fig. 15.

The drawing of good curves through points from experimental data can be perfected only through experience and practice. For extremely accurate work, such as some of the equipment manufacturers must do in making their graphs, there is a mathematical method of getting an exact smooth curve. However, most graphs showing experimental data are made up by getting accurate experimental data in the first place. Such data will come close to following along a smooth curve, in which event an experienced laboratory engineer can draw a reasonably correct curve without the use of any additional calculations.

For the average air-conditioning man in the field today, the ability to read and interpret curves drawn by others is much more important than the ability to plot curves from test data. Even when we do encounter test data that must be presented graphically, it is ordinarily the result of a relatively rough field test, which need be presented only by a rough graph rather than an extremely precise one.

PROBLEMS

1. In a column write numbers from one up to your present age. Opposite each, write your weight at each age figure, as well as you can remember it. Draw a graph of the data shown in this table. Is the graph a continuous or a discontinuous function?

2. Define extrapolation. Explain the danger of extrapolating curves. In what circumstances is it fairly safe to extrapolate curves?

3. Figure 8 shows the efficiency of a typical pump. If the pump is operating at 75 per cent of rated flow, what would the efficiency be?

4. Figure 9 shows the fuel consumption and thermal efficiency of an internal-combustion engine. At 60 per cent load, what would be the fuel consumption and efficiency of this engine?

5. Figure 10 illustrates data regarding a typical coal-burning furnace. If we desire an output of 150,000 Btu per hr, find the following:

(*a*) Draft required, (*b*) efficiency, (*c*) pounds of coal per hour per square foot grate.

6. Using Fig. 13 choose the proper size of condenser for the following operating conditions:

Wet-bulb temperature of entering air	76°
Desired condensing temperature	110°
Evaporator capacity—220,000 Btu per hr	

7. Using the data given in Table 4, plot a graph. Your graph should resemble closely the one shown in Fig. 15.

CHAPTER III

PHYSICAL PROPERTIES OF AIR

The complete study of the physical properties of air, including the discussion of pressure, temperature, and volume, is too lengthy to cover in one chapter. This subject, including the several gas laws (remember gas laws all apply to air), will be discussed in this chapter and in Chap. IV.

Composition of Air. We have already told you that air is a mixture of several gases. As you know, air contains a good deal of the gas oxygen. Oxygen is necessary to sustain human and animal life. If a large number of people are locked in an absolutely sealed room, they will ultimately use up all the oxygen in the air. As this oxygen supply becomes diminished, the weakest of the people confined in the room will die. More and more of them will die of suffocation as the oxygen supply becomes further diminished. Even the most stalwart persons will die before the oxygen supply is completely exhausted.

Human beings would find life difficult and breathing a strenuous process if they were to be placed in a room where the air had only half its normal oxygen content. Air contains a good deal less than half oxygen under normal conditions. As a matter of fact, air contains about one-fifth oxygen, the remaining four-fifths being mostly the gas nitrogen, with a small amount of other gases. The gases in air and the relative proportions of these gases are shown in Table 1.

TABLE 1

Gases in Air	Per Cent by Volume
Nitrogen	78.5
Oxygen	21.0
Carbon dioxide	0.03–0.05
Argon	0.03
Other gases	0.42–0.44

The gas argon, which is shown in Table 1, is chemically a pure element, not a compound. You will note that an average sample of air contains about the same amounts of carbon dioxide and argon. Carbon dioxide you will recall is a chemical compound, not an element. Argon is an extremely rare gas and is shown in Table 1 only because it may be present in an average sample of air in about the same amount as the more familiar gas carbon dioxide. You will also note in Table 1 that air contains something less than ½ of 1 per cent marked "other gases" in the table. These other gases include the following:

Hydrogen	Helium	Ozone
Neon	Xenon	Carbon monoxide
Krypton		

Table 1 is not intended to show the exact percentages of the various gases in air which would be found in *every* sample of air that might be analyzed. Obviously, a sample of air taken from the center of New York City would differ in composition from a sample of air taken at Saranac Lake, and this second sample would in turn differ slightly from a sample of air chosen from some other point. However, Table 1 does show the composition of an average sample of outdoor air.

For practical work with air, we are ordinarily interested in the oxygen content either to sustain human life or to sustain the burning of fuel. For practical purposes, we consider that ordinary outdoor air is 21 per cent oxygen and 79 per cent nitrogen by volume. We are introducing a slight error by including a fraction of 1 per cent "all other gases," under the heading of nitrogen. However, this error is extremely small for two reasons: (1) there is only a small amount of these various other gases compared with the large amount of nitrogen; (2) the other gases behave in about the same way as the nitrogen when they are mixed with nitrogen, as they are in the air.

You may have wondered why it is possible for the air to continue to have the same proportions of oxygen year after year when we have been burning up this oxygen in fires and in the lungs of human beings and animals for many centuries. The story of how nature preserves this required balance of oxygen in the air is interesting. However, it is too long and too complicated a chemical and biological story for us to include here. One aspect of nature's mechanism for replenishing the oxygen in air is easily understood, and we wish to present it.

When a human being or animal fills his lungs with air, a great deal of the oxygen that has been taken into the lungs is used up. When a lungful of air is exhaled, its composition might be as follows:

	Per Cent
Nitrogen	79
Oxygen	11
Carbon dioxide	10

In other words, the human body uses some of the oxygen to burn carbon that has been taken in by the body as food. This burning process provides the body with heat and the energy with which to work. Of course the process is not quite so simple as this, and the amount of carbon dioxide exhaled may vary a great deal. The point is that human beings exhaust a portion of the oxygen in each lungful of air breathed, replacing this oxygen with carbon dioxide.

Most green-leafed plants, including trees, use air in an entirely different manner. In fact they use it in almost the opposite manner. For purposes of sustaining their life and growth, green-leafed plants require a great deal of carbon dioxide. Since there is so little carbon dioxide ordinarily present in the air (refer to Table 1), such plants must "breathe" large amounts of air to get their required carbon dioxide. The interesting thing is this: Plants take the *carbon* out of carbon dioxide and "exhale" the pure oxygen in return for the carbon dioxide that they have inhaled. This is just a part of the story of how nature maintains the proper proportion of oxygen in the air.

Before proceeding further, we must now agree on the use of the two words "air" and "atmosphere." Let us use the word atmosphere only when we mean the entire envelope of air that surrounds the earth. We shall use the word "air" whenever we speak of air in New York City, the tank of compressed air, air brakes, or any other ordinary reference to the substance known as air. Most of the time we shall use the word air, but there will be times when we wish to talk about all the air in the world, and then we shall use the word atmosphere.

Air is a Real Substance. Because we cannot see air, frequently we make the mistake of not realizing that it is actually a substance. Let us recognize that air *is* a real substance. Air is matter, and it has weight and a little of the "solidness" that we associate with wood, steel, water, or any other matter. In other words, let us not think of a glass or cup in the closet as being "empty." Just because the cup does not contain coffee, milk, or water, it is not *empty;* instead, it is *full of air.* If you have sufficiently accurate scales, it is possible to perform an interesting experiment. Take two identical metal cups, and seal them with airtight covers; leave cup *a* full of air, but connect cup *b* to a device that will *remove* all the air from inside the solid container. We could then weigh the two cups and easily see that cup *a*, which is full of air, is *actually heavier* than the truly empty container.

In order to realize that air is a real substance, recall the damage that can be done by a windstorm. After all, a wind is merely a large quantity of air moving rapidly. Air is a sufficiently real substance so that such a moving storm of air can blow over buildings, trees, bridges, etc. Also when aircraft of the heavier-than-air type (not balloons, blimps, or dirigibles) are in flight, machines weighing 60,000 and 700,000 lb are supported by air alone. In this case you have air supporting a weight that could be lifted mechanically only by means of a strong steel cable.

Atmospheric Pressure. You probably already know that the earth exercises a pull toward itself on all objects that are at or above the earth's surface. This pull, or attraction, is called the "force of gravity" or "gravitational force." The nature and the mathematics of gravitational force were first studied in detail by Sir Isaac Newton in England many years ago. Legend tells us that Newton first started to think along this line when he noticed an apple fall from a tree.

We have stated that air is a real substance and has weight. Now think a moment about the atmosphere (the envelope of air that surrounds the earth). If air did not have weight, would the atmosphere stay close to the earth, surrounding it like an envelope as it actually does? It probably would not. Probably the air, if it did not have weight just like any other substance, would travel off through space, and the people on the earth would have perished long ago for lack of oxygen.

Because air has weight, it is held in a narrow envelope around the earth. The earth is about 8,000 miles in diameter or approximately 25,000 miles in circumference. Around this spherical body is a layer of air that we call the "atmosphere." This layer is quite thin, being sufficient to support human life only for a distance of a few miles away from the earth's surface. The total thickness of the envelope of air is probably between 15 and 50 miles. However, scientists disagree on this point, as it is impossible to explore the atmosphere to an indefinitely high altitude.

In order to examine the atmosphere, especially regarding its weight at the earth's surface, let us imagine a square column of air measuring 1 in. on a side and extending from the earth's surface up to the highest point where there is any air at all. Since this column measures 1 by 1 in., its area is 1 sq in. Such a column is illustrated in Fig. 1. At point A, where the bottom of our column of air rests on the earth's surface, the weight of the entire column—the weight of the full height of the column of air—is resting on the surface of the earth. The weight of this column of air is known to be 14.696 psi at sea level on the earth's surface. In all our future work we shall use the value 14.7 instead of 14.696. For all ordinary purposes, 14.7 is sufficiently accurate, and we mention 14.696 merely because it is scientifically the exact value.

Look again at Fig. 1 showing our column of air 1 sq in. in area extending upward from the earth's surface. Of course, there is a similar column resting on *every* square inch of the earth's surface, but we need to consider only one of these columns, as shown in Fig. 1, for this discussion. At this time you should also recall the kinetic theory of gases as discussed in Chap. I. At the top of the earth's atmosphere, which is the level marked F in Fig. 1, the pressure would be zero. This is because there is no atmosphere (or air) above this level to press downward owing to the force of gravity. In terms of the kinetic theory of gases, there is no wall or obstacle above the air at the top of the atmosphere; hence the molecules flying about are perfectly free and do not even have other molecules above them against which to strike as they fly about. When you think of the upper limit of the atmosphere in this manner, you can readily understand why scientists disagree on just how high the earth's atmosphere does extend. No one can say how far a single molecule of oxygen or nitrogen may fly upward from the level marked *F* before being pulled back down again by the force of gravity.

Now consider the level marked *E* in our column of air. This level is approximately 5 miles above the earth's surface, which means that above

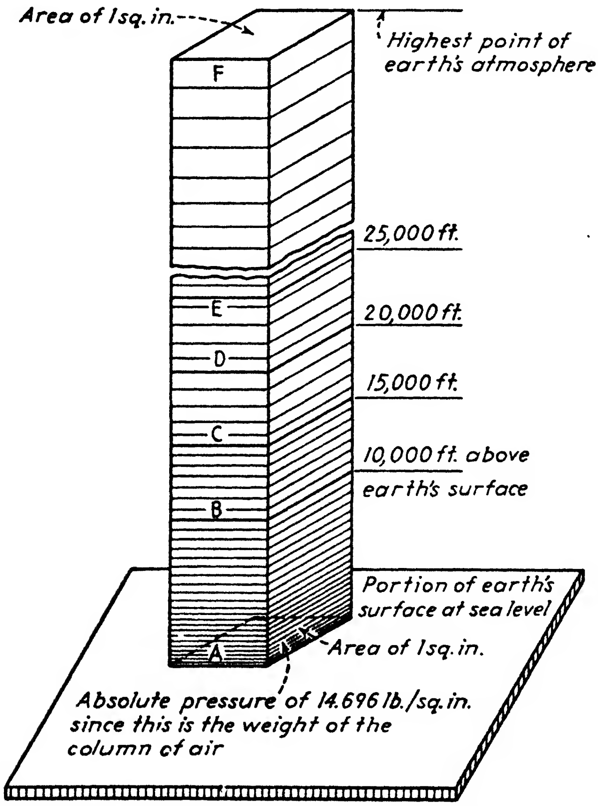


FIG. 1.—Notice the feeling of pressure on your eardrums as you descend in a fast elevator from the top floor of a high building.

it there is a column of air at least 10 miles high. Hence molecules that are flying about at the 25,000-ft level will strike against other molecules above (and all around) any point on the 25,000-ft level. If we consider one molecule of air that is flying about at the 25,000-ft level, the millions of other molecules at this point act somewhat like retaining walls to limit the free path of the flying molecule that we are considering. From this you can see that the molecules are much closer together at the level marked *E* than they are near the top of the earth's atmosphere. This explains, in the light of the kinetic theory of gases, how we should definitely expect to have measurable pressure at the 25,000-ft level.

Another way of realizing that there is a definite pressure at our *E* level is to realize that we have a column of air from *E* to *F* pressing down on the air below level *E*.

Following this reasoning further, we may investigate the air at the levels marked *D*, *C*, and *B*. For instance, pressing downward on the air at level *D* is the weight of the column of air from *E* to *F* plus the weight of the column of air from *D* to *E*. Since there is a heavier weight of air pressing down on the level at *C*, you can see that the pressure there must be greater than it is at the level *E*. Similarly, the further down toward the earth's surface, the greater the pressure. The pressures at altitudes above sea level are shown in Table 2.

TABLE 2.—ATMOSPHERIC PRESSURES AT VARIOUS ALTITUDES

Altitude, ft	Absolute Pressure, psi
Sea level	14.696
1,000	14.00
2,000	13.60
3,000	13.30
5,000	12.60
10,000	10.80
15,000	8.80

At an altitude of 15,000 ft above sea level, the absolute pressure of the air is almost one-half of the absolute pressure at sea level. In Chap. I we said that gases are elastic. This means that gases can be squeezed or compressed. Such compression of a gas increases its absolute pressure, and at the same time it crowds more molecules of the gas into a given volume of space. This means that the air close to the earth's surface, where the absolute pressure is highest, is more compressed. In other words, there are more molecules of oxygen and nitrogen in 1 cu ft of space at the earth's surface than there are in 1 cu ft of space at a higher level, such as 15,000 ft. When the molecules of a gas are packed more closely together, making 1 cu ft of the gas heavier, the gas is said to be more "dense." The air at higher altitudes, by contrast being less closely packed, is said to be more "rare" (or less dense) than the air close to the earth's surface.

Remember what we have said earlier about human beings and animals requiring the oxygen in air in order to exist. Let us imagine that a man's lung holds 1 cu ft of air (actually the capacity of a man's lung is only a small fraction of 1 cu ft). When a man fills his lung with air at sea level or at any level up to 1,000 or 2,000 ft, he gets sufficient oxygen in each lungful of air; breathing is easy and normal. At altitudes above 10,000 ft, where the air is more rare, a lungful of air contains much less oxygen. Hence at this high altitude human beings have difficulty in getting sufficient oxygen.

We must breathe more rapidly at these high levels, and physical exertion causes us to get "out of breath" much more easily at high altitudes. The so-called stratosphere airplanes that are designed to fly at 15,000 and 20,000 ft altitudes use mechanical compressors to fill the cabin with air that is compressed to an absolute pressure of 12 to 13 psi, so that passengers in the plane will not experience difficulty in getting sufficient oxygen with each breath.

Consider the following comparison concerning the pressure of water at different depths in a lake or swimming pool. Figure 2 shows a cross section of the earth and layers of the earth's atmosphere up to 15,000 ft

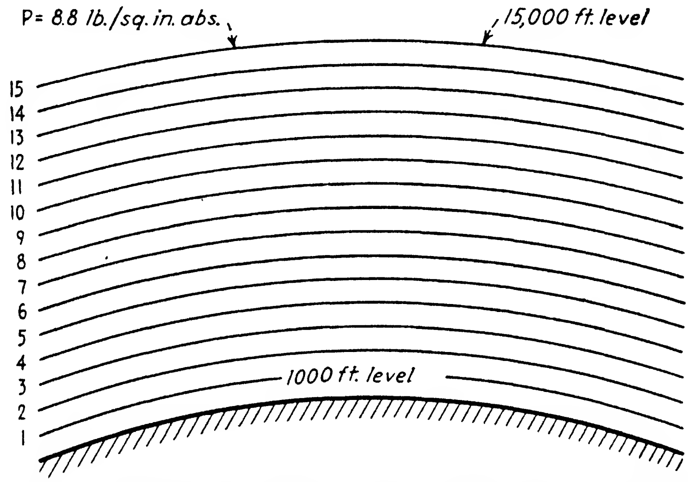


FIG. 2.—At altitudes above 10,000 ft it becomes difficult for one to breathe.

above sea level. Figure 3 shows water in a lake down to a depth of 75 ft. We may imagine a column of water extending from the bottom of the lake to the surface just as we imagine our column of air extending upward from the earth's surface.

If you have ever done any diving, you have certainly noticed the feeling of pressure on your ears if you go deeper than 8 or 10 ft below the surface of the water. Consider our column of water 1 sq in. in area in the lake. We have actually the same picture as we had in the case of our column of air, with two small but important differences. (1) Water is a much heavier substance than air. (2) At the top of our column of air in the earth's atmosphere, we have a pressure of zero, but on the top of our column of water we have a pressure of 14.7 psi abs caused by the earth's atmosphere pressing down on the water surface of our lake. The pressures at various depths in the lake are shown in Fig. 3.

Pascal's Principle. This law is named after the man who first discovered it and announced it to the world's scientific men. It merely states: "In a body of gas or liquid that is confined under any pressure, the pressure at any particular point acts equally in all directions." Hence as you walk along the street, the earth's atmosphere presses *downward* on your head, *upward* against the bottom of your foot, and *sideward* against your body, the pressure in all directions being 14.7 psi. You can readily see that this is true. Suppose that the earth's atmosphere pressed downward only, exerting no pressure sideways; this would tend to crush you just as you can crush a lump of putty by pushing down on it, without supporting the sides.

Similarly, when an automobile tire is full of air, the pressure is exerted against all points on the inside of the tire. Also when a steam boiler contains steam (which is a gas) under pressure, this steam pressure is exerted

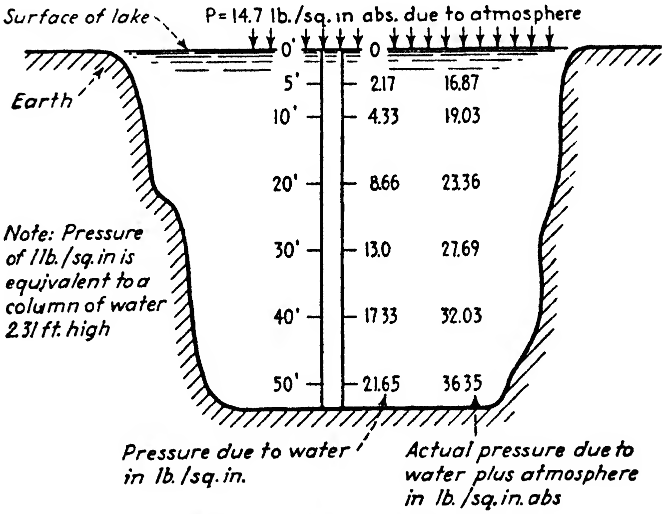


Fig. 3.—Pressure increases with depth below the surface.

upward against the top of the boiler, *outward* against the sides of the boiler, and *downward* against the surface of the water in the boiler. The water in the lower part of the steam boiler is under the same pressure as the steam in the upper portion of the boiler. The water then exerts its pressure against the sides of the boiler and also against the bottom of the boiler.

You need not remember the name of this law, stating that liquids and gases exert their pressure equal in all directions. It is important, however, that you remember the physical fact that the law states.

Absolute Pressure, Vacuum, and Gauge Pressure. We have already seen that the normal atmospheric pressure at the earth's surface is 14.7 psi. Suppose that we were to unscrew the pressure gauge from a steam boiler and the pressure gauge from the compressed-air tank at a gasoline station and lay the gauges on the table. These gauges would then read a pressure of 0 psi, but there is something wrong about this because we know that the pressure due to the earth's atmosphere is 14.7 psi. This apparent error is easily explained.

Since at all points on the earth's surface we experience this atmospheric pressure of 14.7 psi, this would be the pressure inside a steam boiler with the fire out. There would also be a pressure of 14.7 psi in a gasoline-station compressed-air tank when it was initially installed and the air compressor had not yet been connected. When we raise the pressure inside a tank or an enclosure, we always start with the atmospheric pressure of 14.7 psi, and then we raise the pressure to some point higher than the atmospheric pressure with which we started. We want a gauge that will tell us how much we have *raised the pressure* in a tank *above atmospheric pressure.* There is already a pressure of 14.7 psi in the atmosphere *outside* the tank or steam

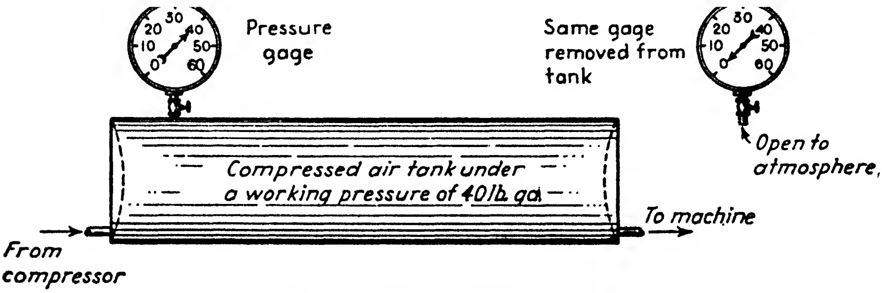


FIG. 4.—Zero psig is equal to 14.7 psia.

boiler. Suppose that we have raised the pressure inside the tank or boiler so that it is 100 psi greater than atmospheric pressure. The information we desire from a pressure gauge is an indication of the amount that we have increased the internal pressure of a tank above normal atmospheric pressure. See Fig. 4.

Because of the information that we want from a pressure gauge, ordinary pressure gauges are so constructed that when exposed to atmospheric air they read a pressure of 0 psi. This zero shown on the pressure gauge is an artificial zero pressure. It is not the absolute zero pressure. When measuring a pressure starting from our true absolute zero pressure, which exists at the top of the earth's atmosphere, we label our pressure "—— pounds per square inch absolute" (——psia). When we measure a pressure with an ordinary pressure gauge, starting from our artificial zero or starting point, we label the pressure "——pounds per square inch gauge" (——psig). ***In all gas laws that we have studied and that we shall study, where pressure is mentioned, we must use absolute pressure. We can never use gauge pressure in connection with gas laws.***

Hence we must have a quick and easy method of converting pressures back and forth from gauge pressure to absolute pressure and from absolute pressure to gauge pressure. Figure 4 illustrates graphically the means of making this conversion. However, the graph shown in Fig. 4 is of value only to learn this method. Actually we shall always make the conversion by one of the following simple formulas:

$$\text{Absolute pressure} = \text{gauge } P + 14.7 \ (\text{abs } P = \text{gauge } P + 14.7) \qquad (1)$$

$$\text{Gauge pressure} = \text{abs } P - 14.7 \ (\text{gauge } P = \text{abs } P - 14.7) \qquad (2)$$

where P = pressure, psi.

Suppose that we have an absolute pressure of 7 psi in a sealed tank. How can we read this pressure on a pressure gauge? From formula (2) above we would have

$$\text{Gauge } P = 7.0 - 14.7 = -7.7 \text{ psia} \qquad (3)$$

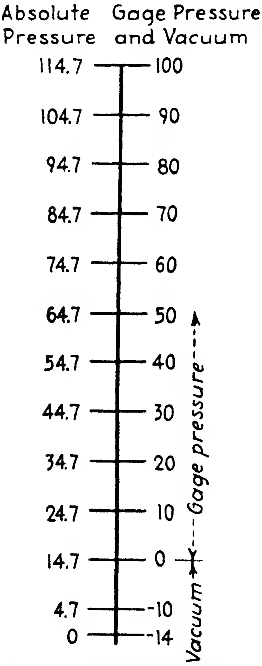


FIG. 5.—Graphical pressure conversion.

You can convert this by looking at the lower portion of Fig. 4. As actually constructed, some types of pressure gauges would read "−7.7 psi" if connected to a tank where the absolute pressure was 7.7 psi. However, instead of using a minus sign before pressures lower than 0 psig, some special gauges known as *vacuum gauges* would not read "pressure" at all. Instead, on such gauges the scale *above* 0 psig would be marked "pressure in pounds per square inch." *Below* zero the gauge would be marked "vacuum in pounds per square inch." Hence such a gauge would read "7.7 psi vacuum."

Whenever we have a gas or a liquid in an enclosed tank at a pressure below 0 psig, we call this condition a partial vacuum. We have a perfect vacuum only when we remove every molecule from within a tank, thus producing a pressure of absolute zero within the tank. In other words, the absolute pressure of *zero* is the same as a vacuum of 14.7 psi or a so-called perfect vacuum. (Actually it is impossible to create a truly perfect vacuum because we cannot remove every last molecule from a gas inside a tank. However, a cheap vacuum pump will produce a vacuum of 14 psi, and a good vacuum pump can easily create a vacuum of 14.6 psi).

Pressure in "Inches of Mercury." In Fig. 3 you can see that a column of water 34 ft high causes a pressure at the bottom of the column, due to the weight of water, of 14.7 psi. Because mercury is 13.59 times heavier than water, a very short column of mercury will create a pressure at the bottom of a column, due to the weight of mercury, of 14.7 psi. The height of such a column is 29.92 in. Thus a pressure of 14.7 psi is equivalent to three different columns of different substances

1. A column of air, the height of the earth's atmosphere—15 to 50 miles
2. A column of water 34 ft high
3. A column of mercury 29.92 in. high

In measuring pressures below 0 psig, ordinary gauges with dials and pointers are sometimes used. As we have already described, these read either "—— psi vacuum" or "minus —— psig." Frequently, however, the scale and dial of such a gauge is marked in "inches of mercury" instead of in "number of pounds per square inch." This would mean that for a perfect vacuum a gauge might read any one of the following, all of which are equivalent to an absolute pressure of zero:

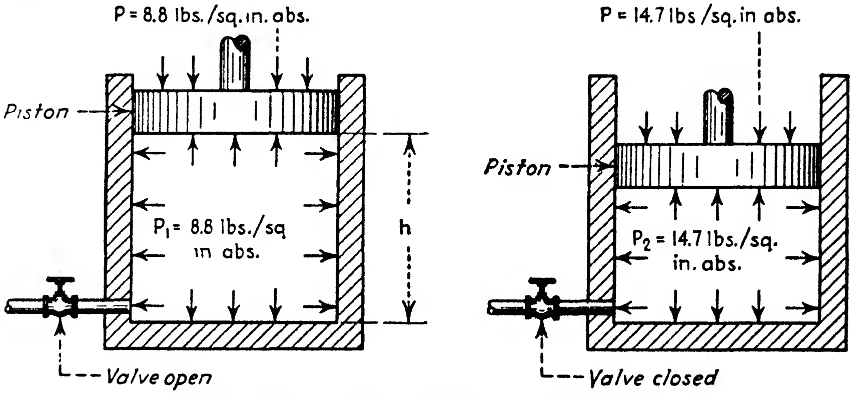
1. −14.7 psig
2. 14.7 psi vacuum
3. A vacuum of 29.92 in. Hg

Density and Specific Density. When we were considering the envelope of air around the earth, which we decided to call the atmosphere, we said that the air becomes more and more closely packed as we come down near the earth's surface. At that time we pointed out that 1 cu ft of air at the earth's surface would weigh more than 1 cu ft of air at a high altitude where the air would be less dense. You already have a general understanding of the words dense and density and know, for instance, that a block of iron is more dense than a block of sponge of the same size. However, in engineering work we use the word density in a more exact and definite manner. Therefore, we shall define density as follows: *Density* is the weight (or mass) *of matter per unit of volume.* In this definition, what do we mean when we say "unit of volume"? So long as we take the same unit of volume each time, it would not matter what unit we use. We could use a pint, a cup, a bushel, a gallon, or any other fixed unit of volume. However, in engineering work we do not use any of these. Instead, we use *one cubic foot* as our unit of volume. One cubic foot of water weighs 62.4 lb. We can say then that water has a density of 62.4 lb per cu ft. However, since we shall *always* use 1 cu ft as our unit of volume, we do not need to repeat the phrase "per cubic foot" every time we give the density of a substance. We simply say the density of water is 62.4 lb. Similarly, for air under certain conditions of temperature and pressure, the density is 0.075 lb.

In engineering work, we use the word "specific" frequently and in a special way. We shall soon have to be familiar with the terms "specific gravity," "specific heat," "specific density," "specific weight," and "specific volume." When we use the word "specific" in this manner, we mean always that the term is used in relation to one unit. In other words, when using specific density, we mean the weight of one unit of volume of a substance. Since we are using 1 cu ft as our standard unit of volume, when we speak of *specific density* of a substance we mean the weight of 1 cu ft of that substance. Hence the word density as previously defined is really the specific density.

When we speak of the specific volume of a substance, we shall mean the volume of one unit of weight of that substance. Because our unit of weight is 1 lb, *specific volume* will be used to mean the volume of 1 lb of a substance. Other terms involving this use of the word "specific" will be defined from time to time when we study the terms themselves.

The "Pressure-volume" Law of Gases. In studying the earth's atmosphere, we noticed that both the pressure and the density of the air became greater as we moved from a high altitude down to the earth's surface.



FIGS. 6 and 7.—Volume varies inversely as pressure.

Suppose that we were to have a tank composed of steel walls, with a *movable* piston in one end, and that we were to permit this tank to become filled with air 15,000 ft above the earth's surface. The tank would then appear as shown in Fig. 6. Suppose that we then close the valve that opened the inside of the tank to the atmosphere and that we locked the piston rigidly in place and brought the tank down to sea level. The tank would then remain exactly as shown in Fig. 6, except that the pressure outside the tank and on top of the piston would be 14.7 psia. Suppose that we now released the piston so that it is free to move. Since there is a greater pressure outside the tank than inside, the piston will be forced to move, squeezing or compressing the air inside the tank. The piston will move until it is as shown in Fig. 7, at which time the pressure inside and outside the tank will be both 14.7 psia.

This experiment is not an imaginary one. It can be and has been performed with an airplane. What can we learn from this experiment? Comparing Figs. 6 and 7, look for the most important changes that have taken place. Assume that the *temperature remains exactly the same.*

$$P = 8.8 \text{ psia} \qquad P = 14.7 \text{ psia}$$

The most important difference that you should have noted is this: The tank in Fig. 7 shows the gas within the tank at a *greater pressure* and at a *smaller volume* after it had been somewhat compressed. There is a simple mathematical law that applies to the compression of gases in this manner at constant temperature. *Remember that we are now dealing with a gas law. Our pressures must at all times be absolute pressure, not gauge pressure.* Stated in words, this gas law tells us that if we double the absolute pressure the volume will be reduced to one-half of the original volume. If we increase the absolute pressure to *four times* the original pressure, the volume will be reduced to *one-quarter* of the original volume.

Let us express this pressure-volume gas law in simple mathematics, using the following letters:

P_1 = original pressure, psia
V_1 = original volume, cu ft
P_2 = final pressure, psia
V_2 = final volume, cu ft

The gas law may then be stated in either of the following ways:

$$\frac{P_1}{P_2} = \frac{V_2}{V_1} \tag{4}$$

$$P_1V_1 = P_2V_2 \tag{5}$$

Equation (4), which may be stated in the following words, is clearer than our original word statement of the pressure-volume gas law: "original pressure *is to* the final pressure *as the* final volume *is to* the original volume." In making this statement, we are merely saying that, mathematically speaking, "as the pressure is increased, the volume is decreased *in direct proportion.*" Equation (5) is simply Eq. (4) written in a different form. To remember this pressure-volume law you may need to remember only one thing. Select either Eq. (4) or (5) or the word statement of the law, and remember one of these. Knowing one, it is easy to get the other two.

Since you may be unfamiliar with the mathematics involved in solving and using these equations of the pressure-volume gas law, we shall briefly go over this material so that you can readily solve such problems.

Proportions and Simple Equations. Although at this time we do not need many of the following mathematical symbols and abbreviations, we shall list them here for future reference:

( )	parenthesis	$\div$, /	divided by
[ ]	bracket	$\sqrt{\ }$	square root of
+	plus	$\parallel$	parallel
−	minus	$\perp$	perpendicular
$\times$, $\cdot$	times	$\angle$	angle

$\triangle$	triangle	a^3	a cubed
$''$	square inch	$>$	is greater than
$\bigcirc$	circle	$<$	is less than
s	distance (space)	$\therefore$	therefore
h	altitude (height)	rt.	right
st.	straight	A'	A prime
$\sqrt[3]{}$	cube root of	A''	A double prime
$=$	is equal to, equals, is equivalent to	A'''	A triple prime
		A_1	A sub 1
$\neq$	does not equal	A_2	A sub 2
a^2	square of a, a squared		

For those who feel able to solve simple equations and problems involving the use of these symbols, this section may be omitted. If you feel confident that you do not have to study this section, try the following example. If you cannot solve it correctly the first time, you should study this section carefully.

Example. We have 100 cu ft of nitrogen gas at a pressure of 0 psig. This gas is compressed at constant temperature to a final pressure of 65.3 psig. What is the final volume?

Solution. We must first convert our gauge pressures to absolute pressures.

$$0 \text{ psig} = 14.7 \text{ psia}$$
$$65.3 \text{ psig} = 80.0 \text{ psia}$$

The original pressure, *i.e.*, the pressure at the start, which our gas-law equation tells us to call P_1, is 14.7 psia.

Our final pressure P_2 is 80.0 psia.

Our original volume, which our gas-law equation tells us to call V_1, is 100 cu ft.

Our final volume V_2 is our "unknown."

Let us now tabulate this information before using our gas-law equation:

$$\begin{aligned} P_1 &= 14.7 \text{ psia (original pressure)} \\ V_1 &= 100 \text{ cu ft (original volume)} \\ P_2 &= 80.0 \text{ psia (final pressure)} \\ V_2 &= \text{unknown (final volume)} \end{aligned}$$

Our gas-law question is

$$\frac{P_1}{P_2} = \frac{V_2}{V_1} \tag{4}$$

We have known numerical values for three of the letters in the equation. Let us substitute the numerical values for these letters and rewrite the equation, using the numbers instead of the letters:

$$\frac{14.7}{80.0} = \frac{V_2}{100} \tag{8}$$

Note that in Eq. (8), V_2 still appears as a letter because we do not know the value of V_2

as yet. We can, however, easily solve the equation for V_2 as follows:

$$\frac{14.7}{80.0}(100) = \frac{V_2}{1} \tag{9}$$

$$V_2 = 18.4 \text{ cu ft} \quad \textit{Ans.}$$

Equation (9) tells us that to get V_2 we must multiply 100×14.7, which equals 1,470. We must then divide 1,470 by 80 to get our answer. (If you are familiar with the methods of solving a problem such as this, your solution will be much shorter than the one given here.)

In learning the simple mathematics necessary to solve equations, we must first learn the simple shorthand used in mathematical calculations. For instance, when we say 10×8, we mean 10 multiplied by 8. Dropping numbers and using letters for convenience, when we say $A \times B$ we mean A multiplied by B. This is written in three different ways in algebra, all of which mean the same thing:

$$A \times B = (A)(B) = AB \tag{6}$$

You may use any one of these forms that is convenient, but we suggest that you do not use the one at the left as the "times sign," the $\times$, may be confused with the letter x in your equation.

When we have fractions multiplied by single numbers, they also may be written in several different ways. Below are shown four quantities, all of which are equal, the form at the extreme right being recommended as the simplest.

$$\frac{a}{b}(c) = a\left(\frac{c}{b}\right) = \frac{1}{b}(ac) = \frac{ac}{b} \tag{7}$$

When we read the fraction $\frac{a}{b}$, we mean a divided by b, or $a \div b$. This is true of all fractions. The portion above the line is called the "numerator," and the portion below the line is called the "denominator." The parts of a fraction are frequently referred to as simply the *top* and *bottom* of the fraction.

In solving equations we may change the equation as much as we please provided we observe four rules:

1. *Add the same quantity* to both sides of an equation.
2. *Subtract the same quantity* from both sides of an equation.
3. *Multiply both sides of an equation by the same quantity.*
4. *Divide both sides of an equation by the same quantity.*

It does not matter how much we change the appearance of an equation so long as in our rearranging of the letters and figures, we obey the four rules given above, and others do not change the magnitude or size of either side of the equation. Hence since the two sides of the equation were equal to start with, they remain equal after we have performed identical operations on both of the two sides.

Frequently, we have a fraction on one side of an equation with our unknown quantity (which we wish to find) being either the top or the bottom of the fraction. Therefore, it is necessary for us to learn how to clear this sort of a fraction so that we finally get our unknown quantity "all by itself." Before doing this, however, make it a rule to arrange the equation so that your unknown is in the *top* of a fraction on one side or the other of the equation. The method of getting your unknown into the top of a fraction is illustrated by Eqs. (17) to (20). The equation given below illustrates clearing a fraction.

$$\frac{a}{b}(b) = \frac{b}{b}(a) = 1 \times a = a \tag{10}$$

Term a was part of the fraction "a divided by b." To clear this fraction we *multiplied the fraction by b*, with the results shown in Eq. (10).

However, if this fraction, which we have cleared, had been in an equation, according to rule 3, page 53, it would be necessary for us to multiply *both sides* of the equation by b to preserve our balance. Let us write the equation in which a is our unknown.

$$\frac{a}{b} = \frac{c}{d} \tag{11}$$

Now let us solve for a according to the method that we have just learned. When we multiply the left side of the equation by b, we have

$$\frac{a}{b}(b) = a \tag{12}$$

When we multiply the right side of the equation by b, we get

$$\frac{c}{d}(b) = \frac{cb}{d} \tag{13}$$

Let us now write Eq. (11)

$$\frac{a}{b} = \frac{c}{d} \tag{11}$$

and solve for a, handling both sides of the equation at once as we would ordinarily do. We now follow rule (3) and multiply *both sides* of the equation by b. which gives us

$$\frac{a}{b}(b) = \frac{c}{d}(b) \tag{14}$$

as we have seen. This may be written as

$$(a)\,\frac{b}{b} = \frac{cb}{d} \tag{15}$$

Since $b \div b = 1$, we now have our answer, thus

$$a = \frac{cb}{d} \tag{16}$$

Before trying to solve an equation as outlined in the preceding paragraph, make it a rule to get the unknown in the *top part* of a fraction on one side of the equation. This will always be possible in the case of equations arising out of gas laws. Let us write Eq. (17) in which we shall assign the following numbers to represent each letter:

$$\begin{aligned} a &= 4 \\ b &= 8 \\ c &= 6 \\ d &= 12 \end{aligned} \qquad \frac{a}{b} = \frac{c}{d} \qquad \begin{array}{|c|c|}\hline a & c \\ \hline b & d \\ \hline \end{array} \tag{17}$$

We give you these numbers so that you can check the following equations using letters. Equation (17) may be written in three other ways. From these various ways of writing the equation, you can see that you get any one of the four letters in the top of a fraction in the equation.

$$\frac{b}{a} = \frac{d}{c} \qquad \begin{array}{|c|c|}\hline b & d \\ \hline a & c \\ \hline \end{array} \tag{18}$$

$$\frac{c}{a} = \frac{d}{b} \qquad \begin{array}{|c|c|}\hline c & d \\ \hline a & b \\ \hline \end{array} \tag{19}$$

$$\frac{b}{d} = \frac{a}{c} \qquad \begin{array}{|c|c|}\hline b & a \\ \hline d & c \\ \hline \end{array} \tag{20}$$

The box structures shown on the right of the equation are for your convenience in learning this method. For a simple equation of this type, you can always draw a box structure of this nature. Starting with Eq. (17), draw your box structure. You can then turn this box structure upside down and stand it on its top, or you can turn it up on either of the two ends, thus giving you the other three forms shown in Eq. (18), (19), and (20).

If you wish a more detailed discussion of the solving of algebraic equations, we suggest that you obtain a first-year algebra book. However, we propose to review such parts of algebra as we need when we encounter equations that we must solve.

PROBLEMS

1. What are the most important gases in air? If we neglect all gases except oxygen and nitrogen in air, what are the proportions by volume of these two gases?

2. What is meant by the phrase, "an absolute pressure of 1 atm"? If the standard for 1 atm of pressure is taken at sea level, what is the absolute pressure in pounds per square inch on a mountaintop 2,000 ft above sea level?

3. Suppose that you have a *small* tank of compressed air, with pressure gauges inserted in the top, bottom, and sides of the tank. Would the pressure gauges read the same or differently? Why?

4. Draw a "ladder scale" showing the comparison between absolute pressure and gauge pressure from 0 to 100 psia. Draw a similar scale showing the comparison between vacuum in pounds per square inch, absolute, and in inches of mercury, for the range from 0 psig to a perfect vacuum.

5. Suppose that you have a telescopic tank containing nitrogen gas at a pressure of 60 psig. If the volume of this tank is decreased at constant temperature from an original volume of 50 cu ft until the pressure becomes 120 psig, what will be the final volume of the tank?

6. Suppose that a large commercial gas holder or gas tank contains 2,000,000 cu ft of gas at a pressure of 2 psig. If the movable portion of the tank is raised (increasing the volume) so that the final pressure is 1 psig, what will be the final volume of the tank if the temperature is constant throughout?

7. Suppose that you are using the cylinder of a steam engine as a tank containing oxygen gas. Since this cylinder has a movable piston, it could be used to compress gas. If the cylinder has a volume of 2 cu ft when the pressure within the cylinder is 0 psig, what will be the pressure in the tank if the piston is moved until the volume is decreased to 0.25 cu ft?

CHAPTER IV

PHYSICAL PROPERTIES OF OTHER GASES

The pressure-volume law, which we studied in the preceding chapter, applies to any gas that behaves according to our laws of perfect gases. Air, although a mixture of oxygen and nitrogen, behaves very closely according to this law. Before investigating two other laws of a similar nature, we wish to present one fundamental law called "Avogadro's law." This law might have been presented at the end of Chap. I, except that at that time we did not have a clear understanding of pressure and the difference between absolute pressure and gauge pressure.

Avogadro's Law. This basic scientific law is named after the scientist Avogadro, who discovered the truth of this law and announced it to the world. A proof of the truth of this law is beyond the scope of this text; so we must accept it on the strength of the fact that chemists and physicists accept it to be true.

Avogadro's law may be stated thus: "*Equal volumes of all gases*, at the same *temperature* and *pressure*, contain the *same number of molecules*." In other words, if we have a dozen tanks each of the same volume and all at the same temperature and pressure, but containing different gases, each tank will contain the same number of molecules regardless of what gas it may contain.

Let us suppose we have a tank containing 2 lb of hydrogen—exactly 2.000 lb—the hydrogen gas being at 32°F and 14.7 psia. For purposes of Avogadro's law, these temperature and pressure conditions are considered "standard conditions." Now let us suppose that we have two other tanks that are *exactly the same size* as the hydrogen tank; *i.e.*, the three tanks have equal volumes. We now put oxygen and nitrogen in the tanks 2 and 3, being careful that the oxygen and nitrogen are at our standard of 32°F and 14.7 psia. In this case, Avogadro's law tells us that each of the three tanks would contain the same number of molecules, although each tank contains a different gas. Refer to Fig. 1, which illustrates these three tanks, and the following discussion about them.

From Table 4 of Chap. I, we saw that a nitrogen molecule is fourteen times as heavy as a hydrogen molecule and that an oxygen molecule is sixteen times as heavy as the hydrogen molecule. We do not know how many molecules each of our three tanks contains, but *we do know that each tank contains the same number of molecules.* Let us pretend that each tank contains 3,000,000,000 molecules. (We want to emphasize that the hydrogen tank does not actually contain 3,000,000,000 molecules; we are just assuming this figure to help our explanation.)

The hydrogen tank, containing 3,000,000,000 molecules, contains 2 lb of hydrogen gas. The nitrogen tank also contains 3,000,000,000 molecules, *each one of which weighs fourteen times as much as a hydrogen molecule. Hence, in the nitrogen tank we must have fourteen times 2 lb, which equals 28 lb of nitrogen. Similarly, the oxygen tank contains 32 lb of oxygen.*

It is not possible to have carbon as a gas at our standard conditions of 32°F and 14.7 psia. Under these conditions, carbon is a solid, such as charcoal, coal, coke, a diamond, or one of its other solid forms. However, if it were possible to produce carbon in a gaseous state at our standard conditions, a tank of the same size as the tanks in Fig. 1 would contain 3,000,000,000 molecules of gaseous carbon. And, since a carbon molecule weighs twelve

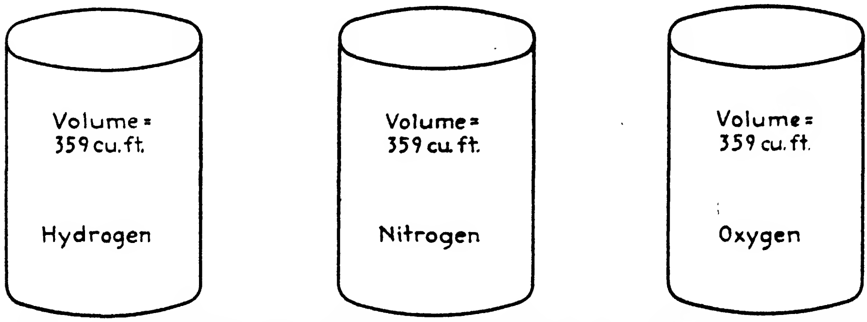


FIG. 1.—The gas weight in each case is its molecular weight in pounds at standard conditions.

times as much as a hydrogen molecule, the imaginary tank of gaseous carbon would contain 12 lb of carbon.

We can now state another gas law which is really just a corollary (a restatement) of Avogadro's law. This law is: "If we take the molecular weight of a gas and then take this *number of pounds* of that gas at 32°F and put the gas in a tank of the proper size so that the pressure will be 14.7 psia, then the *volume* of the tank *will be the same for any gas at all.*" When we take the number of pounds of a gas which is represented by the *molecular weight* of that gas, we call the quantity the "pound-molecular weight" of that gas. The pound-molecular weight of hydrogen is 2 lb; of oxygen, 32 lb; and so on. In other words, for the same pressure and temperature conditions, the pound-molecular weight of *any gas at all* occupies a certain *fixed volume.* At 32°F and 14.7 psia, which are our standard conditions in connection with Avogadro's law, this fixed volume is 359 cu ft. (Instead of using "pound-molecular weight," chemists frequently use the abbreviation "mole.")

For the various elements and combinations with which we have already become familiar, Table 1 shows what we have learned from Avogadro's law.

Note that Avogadro's law applies to *any gas*, regardless of whether it is an element or a compound:

TABLE 1

Element or compound (must be in gaseous state)	Volume at 32°F and 14.7 psia cu ft	Symbol	Molecular weight	Gas in tank, lb
Carbon	359	C	12	12
Hydrogen	359	H_2	2	2
Nitrogen	359	N_2	28	28
Oxygen	359	O_2	32	32
Sulphur	359	S	32	32
Carbon monoxide	359	CO	28	28
Carbon dioxide	359	CO_2	44	44
Water (steam)	359	H_2O	18	18

This table shows that 1 mole (one pound molecular weight) of any substance in a gaseous state has the same volume as 1 mole of any other substance also in a gaseous state, provided both substances are at the same temperature and pressure.

At the present time it may be difficult to realize the practical value of this material based on Avogadro's law. However, efficient operation of coal-burning, gas-burning, or oil-burning furnaces is dependent upon the knowledge of this law.

Temperature Scales. In most engineering work in the United States the Fahrenheit scale is used. Like all temperature scales, it is divided into *degrees*, and the temperatures read on this scale are labeled as *degrees Fahrenheit.* Degrees Fahrenheit is abbreviated °F. Laboratory scientists in this country, however, use the centigrade temperature scale. This scale is universally used in Europe both in engineering and laboratory work as well as for domestic purposes. Temperatures on the centigrade scale are labeled *degrees centigrade,* which is abbreviated °C. In spite of the fact that we use the Fahrenheit thermometer scale in this country for ordinary purposes, we must admit that it is not so logical as the centigrade thermometer scale. Before comparing these two scales, let us consider what we mean by the word "temperature."

Temperature is defined as "the intensity (or degree) of heat." This is a simple definition which states that we know of the different intensities of heat. Obviously, a steam-heated radiator, an ordinary floor, and a red-hot block of iron are each heated to different "intensities of heat." By having a temperature scale, we can express this as 212, 70, and 1800°F for the items just mentioned. (Remember, however, that *temperature* is not the same as *heat* content. Temperature merely measures, by means of some convenient scaled numbers, the intensity of the heat of any particular body.)

Figure 2 shows a graphical comparison of the Fahrenheit and centigrade scales. You will note that 212°F, the boiling temperature of water, is the same as 100°C. At the other end of the scale you see that 32°F, the freezing temperature of water, is equivalent to 0°C. In other words, the centigrade thermometer scale has been determined by arbitrarily calling the boiling temperature of water 100°C and by setting 0°C at the freezing temperature of water. The space between these points on a centigrade thermometer is then divided into 100 equal parts. We feel sure that you will agree that this is extremely logical. It ties in with our whole decimal system upon which United States money is based. Our United States money with 100 cents to 1 dollar, etc., is more logical than most foreign currencies. By looking at our Fahrenheit thermometer scale, however, you can see that it is not equally logical. Why is the zero of our Fahrenheit thermometer scale placed without apparent reason at a point 32°F below the freezing temperature of water? Why does the boiling temperature of water on the Fahrenheit thermometer scale come out at an odd figure such as 212°F?

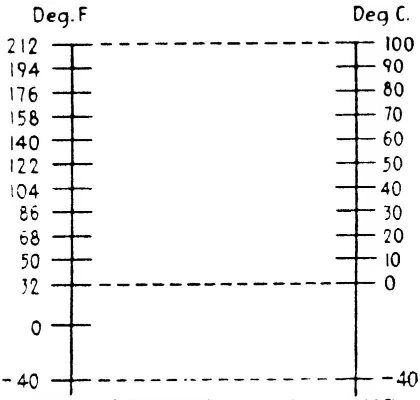


Fig. 2.—Minus 40°F is equal to −40°C.

The reason is that at the time the Fahrenheit thermometer scale was established the temperature we now know as 0°F was the lowest that could be produced in the laboratory. This temperature was chosen as zero for this temperature scale because it was believed to be the coldest temperature that man could obtain. (Early in 1939, scientists at the Massachusetts Institute of Technology announced that they had obtained a temperature slightly lower than −459°F.) In establishing the Fahrenheit thermometer scale and having picked a zero point, the other divisions on the scale were laid out in steps of equal degrees as we now find them.

What is the difference in temperature in degrees Fahrenheit between the freezing temperature and the boiling temperature of water? The boiling temperature is 212°F, the freezing temperature is 32°F, so the difference is 212 minus 32, which equals 180°F. You will remember that on the centigrade thermometer (see Fig. 2) the temperature difference between the freezing point and the boiling point of water is 100°C. This makes it possible for a simple pair of equations to be used in changing from degrees Fahrenheit to degrees centigrade, and vice versa. (*Note:* $^{100}\!/_{180} = {}^{5}\!/_{9}$.)

$$°F = \frac{9}{5}(°C) + 32 \qquad (1)$$

$$°C = \frac{5}{9}(°F - 32) \tag{2}$$

Examples. What Fahrenheit temperature is equivalent to 50°C?

$$\begin{aligned} °F &= \frac{9}{5}(50) + 32 \\ &= 90 + 32 \\ &= 122°F \end{aligned}$$

What centigrade temperature is equivalent to 77°F?

$$\begin{aligned} °C &= \frac{5}{9}(77 - 32) \\ &= \frac{5}{9}(45) \\ &= 25°C \end{aligned}$$

Using Eq. (2), show that $-40°C = -40°F$.

$$-40 = \frac{5}{9}(-40 + -32)$$

$$-40 = \frac{5}{9}(-72) = -40$$

Equations (1) and (2) apply regardless of whether the temperatures being converted are above or below zero on their respective scales. Remember to be careful of the plus and minus signs. The third example involves temperatures in which both the Fahrenheit and centigrade figures are below zero and, hence, must be carefully labeled with a minus sign.

Absolute Zero. In the preceding chapter we studied the meaning of absolute pressure and learned that a pressure of 0 psia meant absolutely *no pressure at all.* In other words, absolute zero in our absolute-pressure scale was a true zero. *There is no such thing as a pressure lower than absolute zero.* In the case of both the Fahrenheit and centigrade thermometer scales, as illustrated in Fig. 2, the temperature scales of course extend below zero. In other words, these are not absolute-temperature scales. They are, instead, somewhat like our gauge-pressure scale in that the zero point is an *artificially fixed point.* We know that we can have pressure below 0 psia, and we know that we can have temperatures far below 0°F or 0°C. You probably have heard of outdoor temperatures as low as −60°F.

To keep in line with the scientific foundation that we are building, we must hope to find that absolute zero in the temperature scales means a true zero, which is the lowest temperature that can possibly exist. Think now of the kinetic theory of gases, which we discussed in Chap. I. The kinetic theory of gases told us that molecules in a gas were flying about at high speeds, the molecules having great unoccupied spaces between them. It also told us that as we increase *the intensity of heat* in a gas, the kinetic energy or speed of the molecules increases. In this chapter we have just defined temperature as a measure of the intensity of heat in a substance. Hence we say that we have both *temperature* and the *kinetic energy of the molecules* in a gas as a measure of the intensity of heat in our gas. Thus we say that if we *remove* heat from a gas, we lower the intensity of heat in that gas, which we would measure by reading the temperature of the gas at a lower value. This process would be accompanied by a slowing up of the flying molecules of the gas.

In order to have a truly absolute zero temperature, we should have a condition where the intensity of heat is absolutely zero. This would mean that the kinetic energy of the molecules would be absolutely zero. Now if the molecules of a gas are not moving at all, what do you suppose happens? Imagine Madison Square Garden with 1,000 grains of white flour scattered through the atmosphere. This would be very like the molecules of a gas contained in Madison Square Garden as a tank. If these grains of flour suddenly collapsed in a pile on the floor of the Garden, the space they would then occupy would be practically zero compared with the huge space they previously occupied.

We must at this time state a fact that is accepted by scientists and ask that you accept it without further proof. The *absolute zero for temperatures* is actually taken at a point at which the heat energy of substances would be zero and hence the kinetic energy of their molecules would be zero. If we had a gas that could be cooled to absolute zero without becoming a liquid or compressing to a solid, this gas would shrink to zero volume with the temperature at absolute zero. Actually of course, there would be a small pile of molecules lying at rest, but these would be so small that we could not see them under the most powerful of microscopes. However, scientists are agreed that this condition represents zero volume for a gas.

We are now ready to begin the discussion of the temperature-volume gas law.

Temperature-volume Gas Law. You will recall that in the preceding chapter on the pressure-volume gas law the law held for changes of pressure and volume *when the temperature was held constant.* The temperature-volume gas law, which we are about to discuss, applies to changes in temperature and volume *when the pressure is held constant.* Hence before studying the law we want to show you that it is entirely possible to hold pressure constant while changing volume and temperature. A simple illustration of how this could be done in the laboratory is shown in Fig. 3.

Figure 3 shows a cylinder, the cross-sectional area of which is 10 sq in. The cylinder is filled with gas, and there is a weight on top of the piston, so that the weight plus the piston itself weighs 200 lb. Since we have a total force of 200 lb pressing down on the top of this confined gas and since the piston *area* is 10 sq in., there is a pressure of 20 psig on the gas in the cylinder.

Since atmospheric pressure is also pressing down on the top of the piston, the absolute pressure of the gas is 34.7 psia. From our discussion in Chap. I, you know that if we heat the gas in the tank shown in Fig. 3 the gas will expand and raise the piston. Likewise, if we cool the gas in the tank, the gas will contract, or shrink, and the piston will drop. However, in both these processes the pressure remains constant because the weight bearing down on the top of the gas remains constant. Commercially these principles are used in all the large gas tanks or gas holders of gas companies. In these a telescopic tank is used, and the upper portion of the tank acts both as a piston and weight.

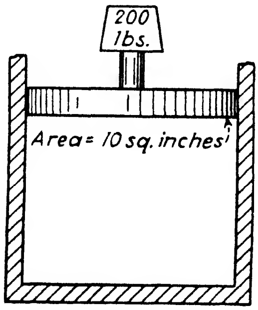


FIG. 3.—At constant pressure, volume varies directly with temperature.

Provided the pressure of a gas is held constant, if 1 cu ft of gas at 0°C is cooled 1°C the gas will lose $\frac{1}{273}$ of its original volume. Likewise it has been observed that if a certain volume of gas is cooled from 0 to −10°C at constant pressure, the gas loses $\frac{10}{273}$ its volume. The reverse of this would be true if the gas were heated. If we heat a volume of gas at constant pressure from 0 to +20°C, the gas would increase $\frac{20}{273}$ above its original volume. After we study one more example of this nature, you will probably be able to state the temperature-volume law of gases yourself. We wish now to point out that on the centigrade thermometer scale absolute zero, or 0°C abs, is 273° below 0°C. That is, 0°C *abs* equals −273°C. If you look at scales *C* and *D*, the two right-hand scales on Fig. 4, you will see the comparison between the centigrade and the absolute centigrade thermometer scale.

Now let us give one more example of a gas cooling at constant pressure and decreasing in volume. If we cool a gas at constant pressure from 300°C abs to 150°C abs, the gas will decrease $\frac{150}{273}$ of its original volume. This means that it loses approximately one-half of its original volume and its final volume must be approximately one-half of its original volume. You should now be ready to state the *temperature-volume gas law* in words. It can be stated thus: "If any volume of gas is cooled at constant pressure, the volume decreases in *direct proportion* to the decrease in the absolute temperature." You will note that in this definition we simply state "absolute temperature"; we did not say whether the absolute temperature should be measured in degrees Fahrenheit or in degrees centigrade. The reason for wording the definition in this manner is that the temperature-volume gas law applies equally well so long as the temperatures of the original and final conditions are both expressed in *degrees absolute on the same temperature scale.*

It does not make any difference whether we use the centigrade or Fahrenheit scale.

It is fortunate for us that this is true, since all our work is done on the Fahrenheit temperature scale. Refer now to columns A and B, the two left-hand columns in Fig. 4, which show the *ordinary Fahrenheit* temperature

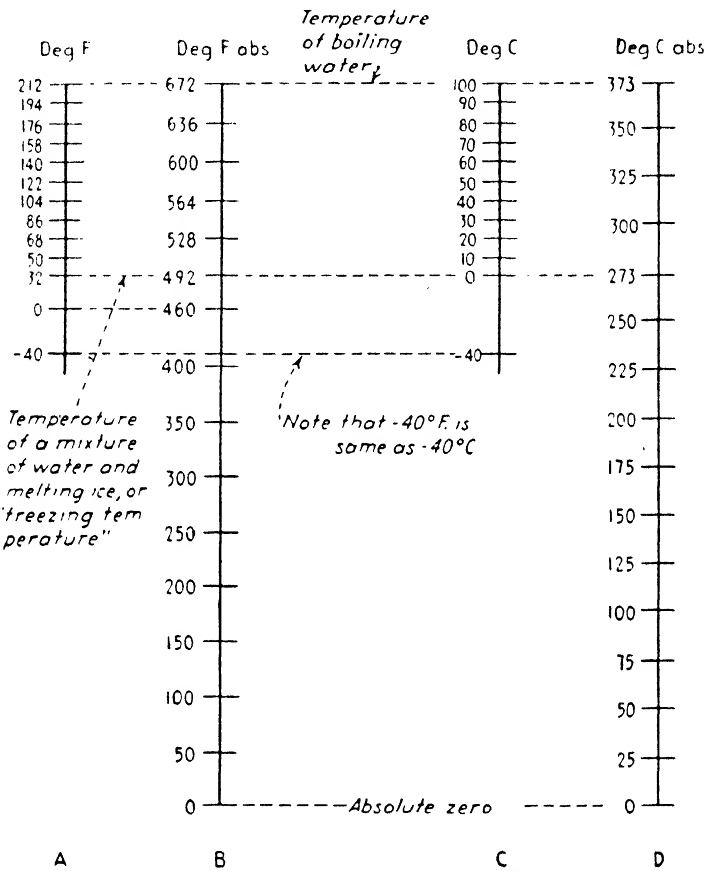


FIG. 4.—Comparison of temperature scales.

scale and the *absolute Fahrenheit* temperature scale. From this you see that 32°F corresponds to 492°F abs. Hence 0°F equals 460°F abs. By the same token, −460°F equals 0°F abs. Check these statements against the graphical interpretation shown in Fig. 4 to be sure that you understand them.

For simplicity in changing the ordinary Fahrenheit and the absolute Fahrenheit scales you need only remember this key "460." You should keep a mental picture of the two left-hand columns in Fig. 4. However, you may also use the following two equations to convert between these two scales:

$$°\text{F abs} = °\text{F} + 460° \tag{3a}$$

$$°\text{F} = °\text{F abs} - 460 \tag{3b}$$

We are now ready to state our temperature-volume law of gases as an equation, which is the only way that it is of any use to us. Be sure to remember three things: (1) the statement of this law in words, (2) that the *pressure* must be held *constant* during the change in temperature and volume, (3) that temperatures must be *absolute temperatures*, which for us will mean temperatures in degrees Fahrenheit absolute. The temperature-volume law may be stated in an equation as follows:

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \tag{4}$$

If our unknown in this equation is V_2 or T_2, Eq. (4) is not in a convenient form for solving. As you remember from the preceding chapter, we can rewrite Eq. (4) as shown below in case V_2 or T_2 are our unknowns.

$$\frac{V_2}{V_1} = \frac{T_2}{T_1} \tag{5}$$

If either V_2 or T_2 were to be the unknown in a problem, we would substitute in our gas-law equation as written in Eq. (4), instead of Eq. (5). For your convenience, we present below the solution of Eq. (5) for both V_2 and T_2 as the unknowns.

$$V_2 = \frac{T_2}{T_1}(V_1) \tag{6}$$

$$T_2 = \frac{V_2}{V_1}(T_1) \tag{7}$$

Before leaving this gas law, let us solve one practical problem.

Example. Suppose, that in a large metropolitan gas tank the temperature increases because of unusual conditions from 25 to 90°F, "owing to heat from the sun," the original volume of the tank having been 1,000,000 cu ft. We must, of course, assume that the gas company uses this tank for idle storage on this particular day and does not remove any of the gas from it. Let us proceed to solve this problem in a manner similar to the solution of our typical pressure-volume gas-law problem in the preceding chapter.

Solution. Remember that this is a gas-law problem. Hence temperatures must be absolute temperatures; so we must convert the temperatures to absolute from °F as given. We do this by using Eq. (3*a*), thus:

$$°\text{F abs} = °\text{F} + 460 \tag{3a}$$

$$T_1 = 25° + 460° = 485°\text{F abs}$$

$$T_2 = 90° + 460° = 550°\text{F abs}$$

What is our unknown in this problem? It is the *final volume*, or V_2. We can now use our gas-law equation, in any of the forms we have in Eqs. (4), (5), (6), or (7). Using the basic form from Eq. (4), we have

$$\frac{V_1}{V_2} = \frac{T_1}{T_2}$$

Substituting numbers we then have

$$\frac{1,000,000}{V_2} = \frac{485}{550}$$

from which

$$V_2 = \frac{550}{485}(1,000,000)$$
$$= 1,135,000 \text{ cu ft} \quad \textit{Ans.}$$

We could have used a short cut in this problem by using our gas-law equation as written in Eq. (6), thus:

$$V_2 = \frac{T_2}{T_1}(V_1)$$

Substituting numbers directly, we get

$$V_2 = \frac{550}{485}(1,000,000)$$
$$= 1,135,000 \text{ cu ft} \quad \textit{Ans.}$$

Temperature-pressure Gas Law. Suppose that we have gas contained in an absolutely sealed container, as shown in Fig. 5. You already know from the kinetic theory of gases that if we heat this gas the molecules of the gas will increase in speed and will strike harder against the walls of the container, thus increasing the pressure within the tank. If we assume that the tank is sufficiently strong so that it does not bulge or bend, i.e., if the volume of the tank remains constant, we have a gas law that will tell us exactly how much the pressure will be increased for a certain temperature increase. In this law, not only must the volume remain constant, during the temperature and pressure change, but also the temperatures must be absolute temperatures *and* the pressures must be *absolute pressures.*

The *temperature-pressure gas law* states: "If the volume of a gas remains constant, the absolute pressure changes in direct proportion to a change in the absolute temperature." Stated as an equation, this law is

$$\frac{P_1}{P_2} = \frac{T_1}{T_2} \tag{8}$$

For your convenience we show this equation restated below in Eq. (9) and then show Eq. (9) solved for both P_2 and T_2, which are often our unknowns.

$$\frac{P_2}{P_1} = \frac{T_2}{T_1} \tag{9}$$

$$P_2 = \frac{T_2}{T_1}(P_1) \tag{10}$$

$$T_2 = \frac{P_2}{P_1}(T_1) \tag{11}$$

Problems are solved by means of this gas law exactly along the lines explained in the two problems we have shown solved by the other two gas laws, previously explained.

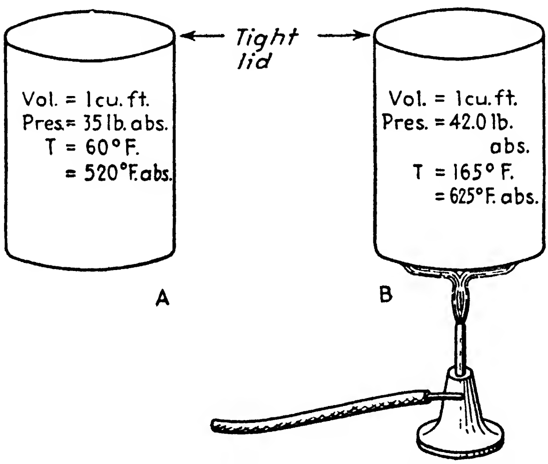


FIG. 5.—At constant volume pressure increases with temperature.

Summary of Temperature, Pressure, and Volume Laws of Gas. We wish to give you now brief statements of the three gas laws relative to temperature, pressure, and volume, which we have discussed in this chapter and the preceding chapter.

Pressure-volume Law. With the *temperature* held constant, the volume of a gas changes in inverse proportion to a change in absolute pressure.

$$\frac{P_1}{P_2} = \frac{V_2}{V_1}$$

Cross-multiplying, this equation may be written

$$P_1V_1 = P_2V_2$$

Temperature-volume Law. With the *pressure* held constant, the volume of a gas changes in direct proportion to a change in its absolute temperature.

$$\frac{V_1}{V_2} = \frac{T_1}{T_2}$$

Temperature-pressure Law. With the *volume* held constant, the absolute pressure of a gas changes in direct proportion to a change in its absolute temperature.

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}$$

Before studying this chapter further, we suggest that you solve some of the problems at the end of this chapter to test your understanding of the above gas laws and the use of the equations required to solve them.

Dalton's Law of Partial Pressures. Dalton's law is another gas law named after the scientist who discovered it and announced it to the world. Suppose that we have a mixture of two or more gases in a tank. For a simple illustration, let us assume a tank of air that we shall consider to be 21 per cent oxygen and 79 per cent nitrogen by volume, the pressure being 85.3 psig. As you know, this is equivalent to 85.3 + 14.7 = 100 psia. Dalton's law states that, when we have two gases *mixed physically* in this manner, the total absolute pressure in the tank is due partly to each individual gas. This means that in our tank of air at a pressure of 100 psia part of that 100 psia is caused by the oxygen and part is caused by the nitrogen. If we could suddenly remove all the oxygen from the tank, this portion of the total pressure would disappear and our pressure gauge would drop to a new value representing the pressure due only to the nitrogen. In a tank such as the one we are imagining, how much of the pressure is due to the oxygen and how much of the pressure is due to the nitrogen?

The answer to this is logical and just what we might expect. You will recall that we just stated that the air was 21 per cent oxygen by volume and 79 per cent nitrogen by volume. According to Dalton's law, the oxygen then causes 21 per cent of the total pressure of 100 psi, and the nitrogen causes the other 79 per cent of the total pressure; i.e., the *partial pressure* of the oxygen is 0.21 (100) = 21 psia. Similarly, the *partial pressure* of the nitrogen is 0.79 (100) = 79 psia. We have not yet given you a brief accurate statement of Dalton's law. Before doing so we wish to point out two fundamental facts that are easy to forget about this law. (1) The two or more gases must be *physically mixed only* — the law will not apply if any chemical combination takes place between the various gases. (2) The total pressure caused by all the various gases and the partial pressure of each gas alone must be stated in pounds per square inch *absolute* — the law will not apply if gauge pressures are used.

Dalton's law may be stated briefly thus: "*The total pressure within a closed vessel containing a physical mixture of several gases is equal to the sum of the partial pressures of the various gases;* and the partial pressure of each individual gas is the same portion of the total pressure as the per cent by volume of the particular gas in relation to the total volume of the tank."

Example. Suppose that we have a tank with an absolute pressure of 10 psi and a volume of 5 cu ft, containing hydrogen. Suppose that we have another tank containing 100 cu ft of nitrogen at a pressure of 10 psia. If we now pump nitrogen by means of a compressor into the hydrogen tank until the absolute pressure is 50 psia, find the following:

1. The per cent by volume of hydrogen and nitrogen in the small tank
2. The partial pressure of the hydrogen and the partial pressure of nitrogen in the small tank
3. If the large tank originally containing the nitrogen were a constant-pressure telescopic tank, how much of its volume would have decreased because of the nitrogen that was removed by the pump and forced into the small tank?

Solution. 1. From Dalton's law we know that the absolute pressures of the hydrogen and nitrogen are in the same ratio as the percentage by volume of these two gases. However, we shall find the answer to part 2 and then get the solution to part 1.

2. The partial pressure of the hydrogen is not changed at all. It was originally producing a pressure of 10 psia, and it still is. The problem tells us that the final total pressure is 50 psia, the 40 psia *increase* has been caused by the *nitrogen* that was forced into the tank by the pump. We can now make a table, as shown in Table 1.

TABLE 1

Gas	Partial pressure	Per cent of total pressure
Hydrogen	10	20
Nitrogen	40	80
Total mixture	50	100

This table gives the answer to part 2 of our problem. It also gives the answer to part 1. Since hydrogen gas is 20 per cent of the total pressure, its volume under the final conditions must be 20 per cent of the total volume in the tank, and 20 per cent of 5 cu ft is 1 cu ft. Similarly, the volume of nitrogen under the final condition in the small tank is 80 per cent of 5 cu ft, or 4 cu ft.

3. You may already have realized that the answer to this portion of the problem after doing parts 1 and 2 is no longer a Dalton's law problem. Using the pressure-volume law, assuming the temperature to be constant, let us think of the nitrogen alone in the large tank, where the nitrogen was originally. The pressure was 10 psia, and we wish to find out how many cubic feet of this nitrogen were removed from this tank. Hence we can

write

$$P_1 = 10 \text{ psia} \quad \text{Original conditions of the nitrogen}$$
$$V_1 = \text{unknown}$$

Now as to *final conditions*, parts 1 and 2 of our solution have told us that we have 4 cu ft of nitrogen at an absolute pressure of 40 psi. (The hydrogen occupies the other 1 cu ft of our 5-cu ft tank and causes the other 10 psia of our total pressure of 50 psia.) Since we know the final conditions of our nitrogen, we can write

$$P_2 = 40 \text{ psia} \quad \text{Final conditions of the nitrogen}$$
$$V_2 = 4 \text{ cu ft}$$

Then write the equation for the pressure-volume law.

$$\frac{P_1}{P_2} = \frac{V_2}{V_1}$$

If we substitute in this equation the known and unknown values that we have just written, we get

$$\frac{10}{40} = \frac{4}{V_1}$$

then

$$V_1 = (4)\ \frac{40}{10} = 16 \text{ cu ft}$$

A few other problems involving Dalton's law are given at the end of this chapter. We suggest that you try these to be sure that you are familiar with the law and how to solve problems involving the use of Dalton's law combined with our other gas laws.

If you have any doubt as to the practical value of Dalton's law or if you have any idea that it is too theoretical to have a place in this text, let us remind you of a few facts.

1. The Electrolux gas refrigerator works with no moving parts whatsoever and produces refrigeration by means of a small gas flame. This refrigeration principle was invented and applied to domestic refrigeration by two young engineers *who knew Dalton's law and knew it well.* In Chap. XII, we shall study the Electrolux refrigeration cycle and see that these engineers applied Dalton's law to a mixture of two gases and managed to tease one gas into forgetting entirely about *total* pressure and being conscious only of its own small *partial* pressure. (The result of this was to cause the gas at a low partial pressure to boil at temperatures of 10°F, or thereabouts.)

2. More closely connected with air conditioning is another application of Dalton's law. All air contains water in the gaseous state. You know that a pan of water placed on a window sill or elsewhere will eventually "dry up," in spite of the fact that it never reaches the boiling temperature. This water changes from a liquid to a vapor state, and the molecules of gaseous water mix with the molecules of the air. Dalton's law is the explanation of this phenomenon, and it is more important as indicated by the fact that *humidity* (water in air) is a more *complicated* problem in air-conditioning work than is the *temperature* of air.

PROBLEMS

1. Draw the centigrade absolute and Fahrenheit absolute thermometer scales from absolute zero up to the boiling temperature of water. Show the points corresponding to −40°F, 0°F, 0°C, the freezing point of water (careful), and 70°F.

2. If 20 cu ft of air is compressed at constant temperature from 0 to 58.5 psig, what will the final volume be?

3. If 15 cu ft of gas at 0°F is heated at constant pressure to 110°F, what will the final volume be?

4. A tank of gas (volume unknown) is at 30 psig and 40°F. If this gas is heated to 340°F, what will be the final pressure?

5. What is the partial pressure of oxygen in air at atmospheric pressure? What is the partial pressure of nitrogen?

6. Distinguish between a chemical combination of two gases and a physical mixture of the gases.

7. A 10-cu ft tank is open to the atmosphere. Equip the opening with a perfectly tight valve, and pump in hydrogen until the pressure gauge on the tank reads 58.8 psi. Then pump argon gas into the tank until the pressure gauge reads 85.3 psi. (*a*) Find the partial pressure of all gases in the tank. (Note there are four gases in the tank.) (*b*) State the percentage by volume of each gas in the tank. (*c*) If the contents of this tank are permitted to expand at constant temperature into a larger tank such that the pressure is 0 psig in the second tank, how many cubic feet of each gas will there be in the second tank?

CHAPTER V

HEAT, WORK, ENERGY, AND THEIR MEASUREMENT

A good deal of the material that you will meet in the early part of this chapter will not be entirely unfamiliar to you. You will have an elementary knowledge from previous chapters. This has been arranged deliberately since the entire subject of heat, work, and energy is very important and is difficult to study in detail without having had an earlier introduction.

Definition of Heat. It is impossible to define heat in a simple clear definition that would not be criticized by some scientists and engineers. You have already learned about the effect of heat upon the kinetic energy or speed of motion of molecules. You know that heat can boil water, and you also know that by means of heat steam engines, gasoline engines, and diesel engines are able to *do work*. In some cases, these engines run electric generators sending electrical energy out over wires; this electrical energy can then be reconverted into heat in electric heaters, toasters, etc. Also, this electrical energy can be converted into *mechanical work* by means of an electric motor.

We, therefore, wish to give you the following simple definition of heat. *Heat* is a form of *energy*. Scientists and engineers will agree that this is a correct statement, but as soon as we try to make this definition longer and put it in clearer terms, there is considerable disagreement as to just what heat actually is. We shall, therefore, consider heat simply as a form of energy, which it unquestionably is. We can readily visualize how adding heat to a body increases the internal energy of that body, for we have formed a picture of the speed of motion of the molecules increasing owing to the addition of heat. By *internal energy* of a body we mean the energy contained within that body.

British Thermal Unit. We have already distinguished between heat content and temperature, and we know that they are entirely different concepts. Temperature, you will remember, is simply a measure of the intensity of heat content and may be expressed on any of four different temperature scales. It is necessary that we have a measure of *heat content itself*. In other words, we know that to heat 1 lb of water from 70 to 80° (a 10° temperature increase) requires a certain amount of heat. It is obvious that it would require twice as much heat to increase the temperature of 1 lb of water 20°; also it would require twice as much heat to heat 2 lb of water through our original 10° temperature range. According to our concept of heat, we think of it as something that we actually *add* or *put into* a body or substance whose temperature has been raised.

Since water is a common and familiar substance, it has been chosen in connection with setting up a means of measuring heat. The unit in which we shall measure heat is known as the British thermal unit, abbreviated Btu. We have previously defined a Btu as the *quantity* or the *amount* of heat required to *raise 1 lb of water 1°F.* This definition is not absolutely exact; it assumes that exactly the same amount of heat is required to heat 1 lb of water from 32 to 33°F, from 71 to 72°F, from 165 to 166°F, or from 211 to 212°F. Actually, the amount of heat required to raise 1 lb of water 1°F is slightly different at different points between the freezing temperature and the boiling temperature of water. A pound of water might require as little as 0.998 Btu to increase its temperature 1°F under certain conditions and as much as 1.005 Btu to increase its temperature 1°F under certain other conditions.

When we encounter a slight variation such as this, we ordinarily find it convenient to use an *average* value. This has been done in laying down the exact scientific definition of 1 Btu. One Btu is rigidly defined as "$\frac{1}{180}$ of the amount of heat required to raise the temperature of 1 lb of water from 32 to 212°F at atmospheric pressure." This is the same as saying that it takes 180 Btu to heat 1 lb of water from 32 to 212°F, which is a temperature increase of 180°F. One Btu is then $\frac{1}{180}$ of this total quantity of heat, or it is an *average* value of the amount of heat required to raise 1 lb of water 1°F.

As you will see in the next paragraph we are going to neglect in all our practical work any slight variation in the amount of heat required to increase the temperature of 1 lb of water 1°F under different conditions. We are going to assume that 1 Btu will increase the temperature of 1 lb of water 1°F, regardless of the initial temperature of the water.

Specific Heat. When we mentioned the word "specific" in an earlier chapter we told you that it would always pertain to *unit values.* Since the meaning of "unit" is "one," this means that specific will always pertain to 1 lb, 1 cu ft, or *one* of some other unit. *Specific heat* is defined as follows: "The specific heat of any substance is the *amount of heat required to raise 1 lb of that substance 1°F.*" We have already seen that for water this specific heat is 1 Btu, since this is how the Btu itself has been defined. If you say that the specific heat of a certain matter is 0.15, you mean that it takes 0.15 Btu of heat to raise the temperature of a 1-lb block of this matter 1°F.

Specific heats may be thought of in another way. In elemental textbooks, the specific heat of any substance is frequently defined in a slightly different way from the definition that we have given you. This definition is: "The specific heat of a substance is the *ratio* of the heat required to raise the temperature of 1 lb of the substance 1°F, to the heat required to raise the temperature of 1 lb of water 1°F." In other words, when using this definition we might say that the specific heat of a certain substance was 0.75, meaning that it took three-fourths as much heat to increase the temperature of 1 lb of this substance 1°F as would be required to heat 1 lb of water 1°F. We feel that this thought of specific heats of other substances as a ratio to the specific heat of water is a needlessly complicated method of thinking of specific heat.

Earlier in this chapter we have mentioned that the specific heat of water may vary slightly from exactly 1.000. This variation is extremely small, perhaps from 0.998 to 1.005. Under ordinary conditions the variation would be even less than this amount, so that for all practical purposes we are entirely correct in considering the specific heat of water as being exactly 1 Btu.

Addition or Removal of Heat from a Body. Data are available giving the specific heat of all ordinary substances including all chemical elements and also such things as ordinary metals, woods, cloths, masonry, and paper. In other words, if we need the specific heat of almost any substance, the value can be obtained from a table or handbook. When we increase the temperature of a body of some substance, regardless of its weight or its specific heat, we obviously are able to increase its temperature only because we have put a certain amount of heat—a certain number of Btu—into the body. We need a simple formula that will enable us to calculate this total amount of heat added to a body when we increase its temperature. Upon developing such a formula we shall find that the same formula also operates if we *remove* Btu from a body to *lower* its temperature. Let us consider a simple case that we can solve without a formula before looking at the formula to fit all cases. Suppose we are increasing the temperature of 2 lb of water 5°F. We know that *each* of the 2 lb of water would require 1 Btu if heated 1°F. We are increasing the temperature 5° instead of 1°, so we find ourselves multiplying five times 2 Btu and getting 10 Btu as a result. In solving this simple problem regarding water, we have not mentioned as a number the specific heat of water. We actually use the specific heat of water as "1" at the start of the problem in saying that it took 2 Btu to increase 2 lb of water 1°F.

Suppose we have a substance whose specific heat is 0.5, and we wish to increase the temperature of 2 lb of this substance 5°F. How many Btu are required? In this case we may be able to get some benefit from thinking of specific heat as a ratio to the specific heat of water. If the substance in this illustration were water with a specific heat of 1, we would merely say that the answer was 10 Btu. Since our substance has a specific heat of 0.5, which is one-half of the specific heat of water, we would immediately say that the answer is one-half of the answer we get when using water, or 5 Btu.

Having solved the preceding two illustrations without a formula, you can probably now write the formula giving the total amount of heat added to a body when its temperature is increased, regardless of its weight or specific heat. From the two preceding examples we saw that we had to *multiply* by the number of *pounds* of the substance if we had more than 1 lb. We also *multiplied* by the *specific heat*, which did not affect the problem if the specific heat were 1, but which was important if the specific heat were different from 1. The weight and the specific heat were multiplied by the temperature increase in our particular problem. Hence we may set up the following formula:

$$Q = cw\,(T_2 - T_1) \tag{1}$$

where Q = total Btu added
c = specific heat of the substance Btu per lb
w = weight of the substance, lb
T_1 = original temperature, °F
T_2 = final temperature (after heating), °F

In Eq. (1), the term $(T_2 - T_1)$ will always be a temperature difference. If we heat a substance from 70 to 80°, this term would be "(80 − 70)," which equals 10°. When we mention a temperature change or a temperature difference, instead of writing $(T_2 - T_1)$ we may frequently use an abbreviation simply meaning "temperature difference." This abbreviation is ΔT. Hence we may write

$$(T_2 - T_1) = \Delta T \tag{2}$$

Equation (2) simply shows us two equal expressions for a temperature difference. Using this new abbreviation, Eq. (1) now becomes

$$Q = cw(\Delta T) \tag{3}$$

The only precaution that must be observed in using this abbreviation is that ΔT is a *minus* quantity if we are *cooling* a body. Suppose we are cooling a body from 90 to 75°. T_1 is then the original temperature, or 90°; T_2 is the final temperature, or 75°. We then have

$$\begin{aligned} (T_2 - T_1) &= \Delta T \\ \Delta T &= (75° - 90°) \\ \Delta T &= -15° \end{aligned}$$

As soon as you become more familiar with these terms, you will simply glance at a problem of this nature and say, "ΔT is 15°—cooling makes it minus 15°." In other words, you will not need to set up the expression $T_2 - T_1$.

We wish to give you three examples of the use of this equation:

Example 1. Suppose that you have 2,000 lb of water to be heated from 50 to 55°, how many Btu will be required?

Solution. Substituting in Eq. (1), we have

$$Q = 1(2{,}000)(55° - 50°) = 10{,}000 \text{ Btu} \quad \textit{Ans.}$$

Or substituting in Eq. (3), we have

$$Q = 1(2{,}000)(5°) = 10{,}000 \text{ Btu} \quad \textit{Ans.}$$

Example 2. Suppose that you have 20 lb of water to be heated from 72 to 212°, how many Btu are required?

Solution. Substituting in Eq. (1), we have

$$Q = 1(20)(212° - 72°) = 2{,}800 \text{ Btu} \quad \textit{Ans.}$$

Or substituting in Eq. (3), we have

$$Q = 1(20)(140°) = 2{,}800 \text{ Btu} \quad \textit{Ans.}$$

Example 3. Suppose that you have 24 lb of a substance having a specific heat of 0.5 which you wish to *cool* from 59 to 46°, how many Btu will be *removed* from the substance?

Solution. Substituting in Eq. (1), we have

$$Q = 0.5(24)(46° - 59°) = 12(-13) = -156 \text{ Btu} \quad \textit{Ans.}$$

Or substituting in Eq. (3), we have

$$Q = 0.5(24)(-13°) = -156 \text{ Btu} \quad \textit{Ans.}$$

The equation that we have just been discussing may be said to be the most fundamental in the entire subject of thermodynamics, or study of the flow of heat. The hot water used in hot-water heating systems reacts according to this formula, picking up Btu in the boiler in the cellar and losing Btu in the radiator and the piping systems. Similarly, the warm air in a winter air-conditioning system has Btu added to it in the conditioner, and the air loses Btu in giving up heat to the conditioned space. The reverse of this action is true of the cool air in the summer air-conditioning system, since this cool air picks up Btu in the conditioned space and then has these Btu removed from it in the conditioner. Finally, the chilled water in any large central air-conditioning plant, or the chilled brine in any cold-storage warehouse or ice plant, gives up and takes on Btu according to this important basic formula. Air-conditioning engineers use this formula many times.

Work and Energy. *Work* in the engineerng sense is entirely different from work in the layman's sense. Suppose that you were to stand with a desk weighing 100 lb on your back for an hour or two. This would be almost impossible as your muscles would grow very tired, and you would certainly feel that you had been doing work. If you were to carry this desk for several street blocks, the street being perfectly level, you would also feel that you were doing a great deal of work. Yet in the engineering sense of the word "work," you did absolutely none unless you moved the object upon which work is being done against a *force*. In the engineering sense, work is done only when a force is applied against a resistance and when that force then overcomes the resistance and moves a definite distance. Mathematically, work is defined as "*force in pounds* multiplied by the *distance in feet* through which the force moves." This we may write as an equation as follows:

$$W = F(d) \tag{4}$$

where W = work, ft-lb
F = force, lb
d = distance, ft

To come back to our problem of holding a 100-lb desk on your back, suppose that you climb a flight of stairs so that the desk is raised 10 ft. Since the desk weighs 100 lb, the earth's gravitational force is exerting a downward pull of 100 lb on the desk. If you overcome this force by a force that your back applies in an upward direction and if you raise this desk 10 feet in the air, you have done 1000 ft-lb of work. This is arrived at as follows:

$$W = F \times d$$
$$1{,}000 \text{ ft-lb} = 100 \text{ lb} \times 10 \text{ ft}$$

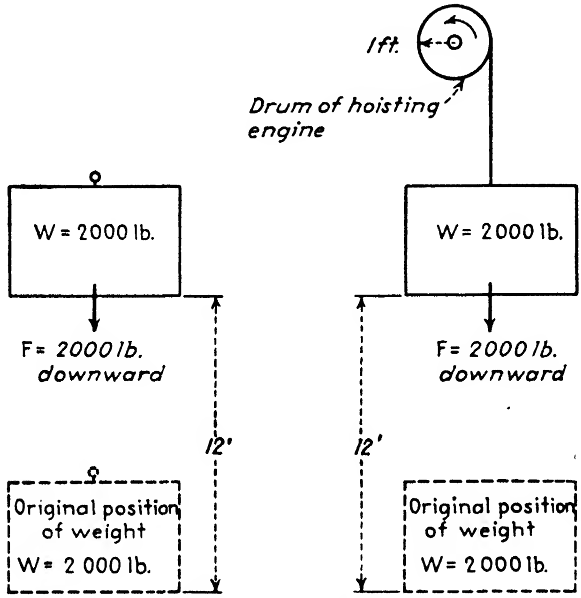
Figure 1 shows a weight of 2,000 lb being lifted vertically 12 ft; hence the work done is

$$12(2{,}000) = 24{,}000 \text{ ft-lb}$$

It does not make any difference whether this weight is being lifted up by an imaginary giant hand or by the cable from the revolving drum on a hoisting engine. Figure 2 illustrates this same weight being lifted through the same distance as in Fig. 1, the only difference is that in Fig. 2 the weight is being lifted by steel cable, as it would actually be lifted by a hoisting engine. In the case illustrated in Fig. 2, the hoisting engine does 24,000 ft-lb of work in lifting the weight.

When weights are lifted vertically, as illustrated in Figs. 1 and 2, the work done is not wastefully expanded or lost beyond recovery. Consider the counter weight that is used in all modern elevators. When the elevator car is at the bottom of the shaft (at the first floor), the counterweight is at the top of the shaft. A steel cable runs from the counterweight up and around a steel drum connected to a motor and then down, connecting to the top of the car. The elevator motor does work in lifting the counterweight to the top of the shaft. However, this counterweight is heavier than the empty elevator car, so that by releasing the brakes the counterweight can fall, thus lifting the elevator car itself. Elevators do not actually operate in this manner (by simply releasing the brakes). However, if they did, you could see that we would get back the work that had been done on the counterweight in lifting it upward. In falling downward, the counterweight would work on the elevator car in lifting it.

So far we have not mentioned friction in our discussion of work. We do not wish to go into a detailed discussion of friction, but we wish to mention that whenever two surfaces rub against each other, one substance tends to



FIGS. 1 and 2.—Work is equal to force times distance traveled.

"drag" against the other and make the motion difficult. If the two surfaces are smooth and well lubricated, the friction is small. If the two surfaces are rough and not lubricated at all, the friction is great. Suppose that you have a 200-lb block of concrete resting on an absolutely level cement floor. If you were to tie a rope around such a concrete block, you know that it would require the application of a strong force or pull on the rope in order to "drag" the concrete block along the floor. Figure 3 shows our 200-lb concrete block and assumes that a force of 50 lb would be required to make the block slide along the floor.

In this case we are not raising the block vertically at all, nevertheless we are exerting a force in order to move the block, so we are doing work. If we move the block 1 ft, we do 50 ft-lb of work; if we move it 5 ft, we do 250 ft-lb of work; etc. The fact that we are exerting our force to overcome frictional "drag," or resistance, does not alter the picture at all. The point is that we are applying the force that moves through a distance equal to the amount we move the weight, and this comes within the scope of our original definition of work.

We wish now to define the word "energy" as we use it in engineering work. So far we have not defined energy, although we have spoken of kinetic energy, heat energy, and electrical energy. Since energy is so varied in its form, we must give it a general definition that will fit all forms of

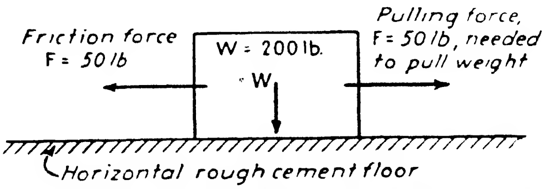


Fig. 3.—The force required to set the block in motion is greater than that to keep the block in motion.

energy. Considerable thought on this matter by scientists has resulted in the following definition: "*Energy* is the *capacity* or the *ability* to do *work*." This is a simple but important definition, and you will find that it fits any form of energy whatsoever.

Relation between Work and Heat. We have just defined energy. Heat has already been defined as one form of energy. Hence we find three terms—heat, energy, and work—closely linked together. We now wish to give you the mathematical relationship between mechanical work ($W = Fd$) and heat, which is a form of energy. This relationship is expressed by two equations, which say the same thing in different ways:

$$778 \text{ ft-lb} = 1 \text{ Btu} \qquad (5)$$

$$\tfrac{1}{778} \text{ Btu} = 1 \text{ ft-lb} \qquad (6)$$

Now let us examine some examples showing how heat energy and mechanical energy can be converted from one form to the other.

Figure 4 illustrates two grinding wheels, or millstones, immersed in a tank of water. The bottom millstone is attached to the tank so that it cannot turn, while the upper millstone is caused to rotate by an electrical motor.

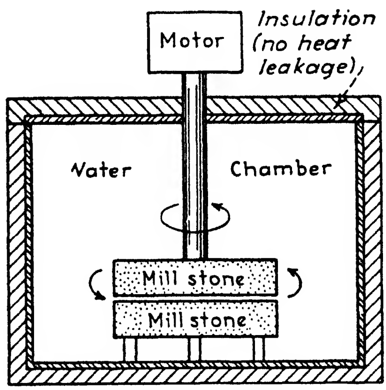


Fig. 4.—Only heat of friction is added to water.

The two millstones have rough surfaces and are pressed together so that there is good rubbing contact. The tank is insulated with a material that prevents heat from leaking out of the tank. Let us assume that this insulation is so good that no heat at all leaks out. In this case all the work that the electric motor does in turning the upper millstone is used to overcome friction between the two millstones. All this mechanical work is converted into heat at the rubbing surface between the two millstones. The two stones will become warm and will ultimately heat up the water in the tank. Suppose we know the weight of the two stones, the specific heat of the stones, and the weight of the water, and suppose we measure the temperature inside the tank before starting the motor and again after a certain amount of work has been done. In this manner it is possible by means of a careful experiment to verify that 778 ft-lb of work produces 1 Btu of heat.

So long as our insulation does not permit the loss of any Btu, there is no Btu loss whatsoever in this experiment. In other words, 778 ft-lb of work

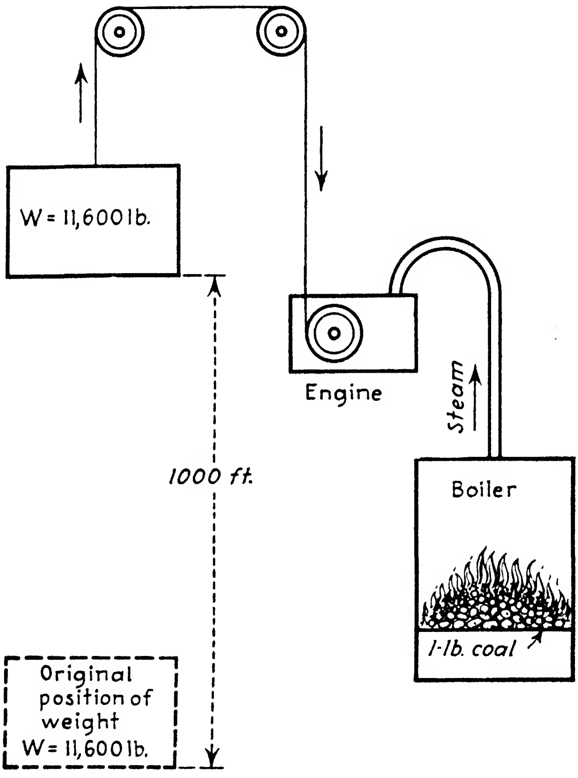


FIG. 5.—Energy transmission losses reduce effectiveness.

done on the upper grinding stone is converted into exactly 1.000 Btu; not so much as a fraction of 1 per cent of the mechanical work is lost or fails to appear as heat. You may regard this as an absolute rule: "the conversion of mechanical work into heat through friction is 100 per cent efficient," *i.e.*, none of the work is lost.

Because of friction and many other factors that we are not going to discuss at the present time, the conversion of heat energy into mechanical work presents a different story. We stated that 1 Btu is equivalent to 778 ft-lb.

This means that 1,000 Btu is equivalent to 778,000 ft-lb. Now consider 1 lb of a good grade of bituminous coal. A pound of such coal when burned with perfect combustion will produce a surprisingly large amount of heat—about 15,000 Btu. If we could get these 15,000 Btu into steam in a boiler and could then run an engine, still without losing any of our energy, we could do about 11,600,000 ft-lb of work. This means that the heat energy in 1 lb of soft coal could lift an 11,600-lb weight (almost 6 tons) 1,000 ft up in the air. This is illustrated in Fig. 5. Unfortunately, when we convert energy from heat energy into mechanical energy we always lose about 65 per cent of the energy that our fuel was capable of producing when burned.

This is a surprising fact if you have not known it before. But it is true that the best engines for the conversion of heat energy from fuel into mechanical work must sustain over-all losses of 65 per cent of the original heat energy. In other words, our best engines are 35 per cent efficient. Ordinary automobile engines are 24 to 27 per cent efficient at best. We now want to define efficiency mathematically as efficiency plays an important part in all engineering work.

$$\%\ \text{eff} = \frac{\text{work done}}{\text{energy input}}(100) \tag{7}$$

Looking at the right-hand side of this equation you will see that the fraction will always be less than 1, since the work derived from a machine will always be less than the energy put into the machine. Efficiency will not be taken up again in this chapter, but we shall have frequent use of our knowledge of the meaning of efficiency and how to calculate it in our later work. For instance, the fans used to handle air in an air-conditioning system and the pumps used to circulate brine and water through pipes have efficiencies at best of slightly over 80%, depending on size. Since every commercial air-conditioning job contains at least one fan, you can see that it is impor ant that we have both a general idea and a mechanical concept of the meaning of efficiency.

Horsepower. Turn back to Fig. 2. Here we have a hoisting engine lifting a weight 12 ft and doing 24,000 ft-lb of work on the weight. Such a weight-lifting operation could be performed by a very small hoisting engine, in which event it might take 20 min or ½ hr to lift the 1-ton weight 12 ft vertically. On the other hand, a large and powerful hoisting engine might lift a 1-ton weight 12 ft in 2 sec. This brings up the question of time: How many seconds or minutes are required to do a certain number of foot-pounds of work? Obviously, the time required to do a certain number of foot-pounds of work depends upon the size and power of the engine doing the work. The only new factor that we have introduced so far in this section is *time*. Using this one new factor we can now define the general term "power." *Power is the time rate of doing work.* In other words, the power of an engine measures *how fast* it can do work. In our English and American systems of units, we measure power by a unit called "horsepower," which is abbreviated hp. One horsepower is defined as being 33,000 ft-lb of work *per minute*, or 550 ft-lb of work *per second*. Thus, you see how the concept of time enters into the horsepower unit and provides a measure of time rate of doing work.

In Fig. 2, we have work done on this weight in the amount of 24,000 ft-lb. Let us assume that engine x does this work in 10 min, while engine y does this work in 5 sec. Let us now figure the horsepower of the engines, assuming no losses through gears, pulleys, etc.

Engine x:

$$\begin{aligned}\text{Work done} &= 24{,}000 \text{ ft-lb}\\ \text{Time required} &= 10 \text{ min}\\ &= 600 \text{ sec}\end{aligned}$$

Since this engine does 24,000 ft-lb of work in 10 min, in 1 min it does work as follows:

$$\frac{24{,}000}{10} = 2{,}400 \text{ ft-lb of work in 1 min}$$

Then

$$\text{Engine hp} = \frac{2{,}400}{33{,}000} = 0.073 \text{ hp}$$

Engine y:

$$\begin{aligned}\text{Work done} &= 24{,}000 \text{ ft-lb}\\ \text{Time required} &= 5 \text{ sec}\\ &= \tfrac{1}{12} \text{ min}\end{aligned}$$

Since this engine does 24,000 ft-lb of work in $\frac{1}{12}$ min, it can do

$$12 \times 24{,}000 = 288{,}000 \text{ ft-lb of work in 1 min}$$

Then,

$$\text{Engine hp} = \frac{288{,}000}{33{,}000} = 8.75 \text{ hp}$$

In both of these examples we used the same procedure to find engine horsepower. The problem gave the rate at which the engine was doing work; we then performed two steps. (1) We found how much work the engine could do in exactly 1 min. (2) We then divided the amount of work that our engine could do in 1 min by the amount of work equivalent to hp, *viz.*, 33,000 ft-lb. The result of this division gave us our engine horsepower.

If you have difficulty in understanding this method of finding engine horsepower, try this simple example. Suppose you have an engine that does 6,600 ft-lb of work in 6 sec. Six seconds is $\frac{1}{10}$ min; so your engine can do 66,000 ft-lb of work in one whole minute. Knowing that 1 hp is 33,000 ft-lb per min, you can see that your engine must be 2 hp since it can do work at just twice this rate. In other words, engine horsepower equals 66,000 divided by 33,000, which equals 2 hp.

The Transfer of Heat by Radiation. One of the two basic laws of thermodynamics may be stated as follows: "When we have two *adjacent* bodies at different temperatures, heat always flows from the hotter body to the colder body." Notice the word "adjacent" in this definition. For purposes of this definition, the earth is considered adjacent to the sun, and we know that heat flows from the sun to the earth. If we place a red-hot block of iron on a stone floor, touching another block of iron at 70°F, heat would

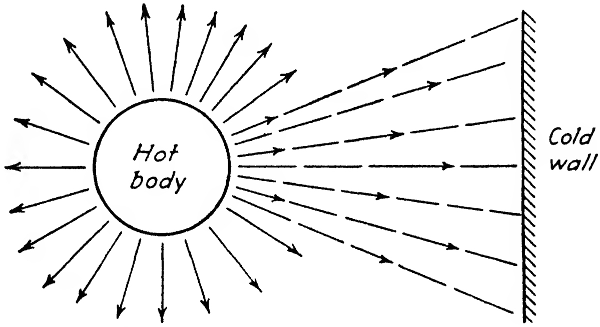


FIG. 6.—Transfer of heat by radiation.

flow from the hot body to the colder body. In other words, the word "adjacent" in this definition means any two bodies which we are considering in a problem of heat flow from one body to the other. There are three methods by which heat may be transferred. The first two of these methods that we shall discuss are a direct inference from the law we have just stated. The third method is not so direct a corollary of the law.

When we have a hot body "adjacent" to a cold body (a steam-heated concealed radiator in a room), heat will flow from the hot body to the cold body, as stated in our thermodynamic law. This method of transfer of heat, when the two bodies are not touching and when no air stream is moving between them, is called "transfer of heat by radiation." The sun transfers heat to the earth entirely by this method. The sun is 93,000,000 miles away from the earth, and there is no air or other gas in most of the intervening space; yet the sun radiates or throws heat to the earth.

It is extremely cold in this space between the earth and the sun. Scientists have ascended in balloons to heights at which the temperature was far below 0°F. In the so-called "outer space" between the earth and the sun, scientists believe that the temperature is probably absolute zero.

From this you can see that radiated heat is a peculiar type of heat. Radiated heat is given off from a hot body in a form that scientists believe is similar to light waves and radio waves. These waves (if that is what they are) are *cold.* However, these waves are a form of energy. On striking any object at all, even on striking the molecules of oxygen and nitrogen in the earth's atmosphere, the energy in these radiated heat waves is changed from this wave form of energy into the simple heat energy with which we are familiar. A bad case of sunburn is an example of how real the heat from these waves can be (see Fig. 6).

Heat Transfer by Conduction. This is the simplest form of heat transfer. You certainly have had the experience of having a sterling-silver spoon in a cup burn your fingers, or of having an iron poker become too hot

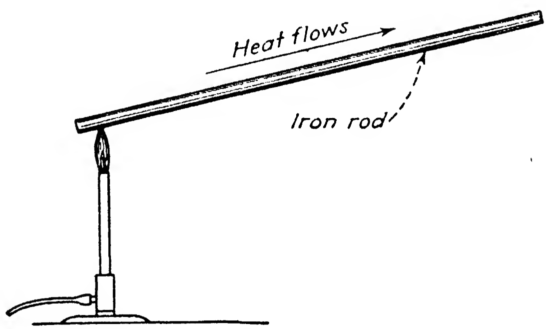


FIG. 7.—Transfer of heat by conduction.

to hold. In either of these cases, heat has been transferred through the metal from molecule to molecule. The source of heat raises the temperature of one end of the spoon, and the molecules at this point begin to vibrate at a greatly increased speed. They then collide with adjoining molecules, the impact of the collision speeds up the adjoining molecules, thus passing along the heat through the metal. When two blocks of iron, one hot and one cold, are placed against one another so that there is metal-to-metal contact, heat is transferred by both radiation and conduction (see Fig. 7).

Heat Transfer by Convection. Whenever we transfer heat actually in an engineering process or when we discover a heat-transfer phenomenon in nature, ordinarily at least two, if not all three, of our methods of heat transfer are combined in the process. In fact, we may make a definite statement that we cannot have heat by *radiation alone* or by *conduction alone.* As to heat transfer by convection, which we are about to define, it is not impossible to imagine a case of heat transfer by convection alone, but actually when we find convection as the most prominent means of heat transfer in any actual process, we nevertheless find that some heat has been transferred also by radiation or conduction or both.

Let us now consider perhaps the most familiar example of heat transfer by convection, after which we shall define heat transfer by this means. Our typical and most familiar example is that of an open uncovered radiator in a room. Let us point out at the start that such a radiator, of course, radiates heat (heat transfer by radiation) to the entire room; the radiator also heats the air just adjoining the hot iron by means of conduction. Also if a person or object touches the radiator, heat flows because of the direct contact.

We can now talk about heat transfer by convection without fear of confusion. In the previous paragraph we stated that the air that was in contact with the hot iron of the radiator became hot by means of conduction. According to our temperature-volume gas law, what happens to a gas when it is heated in this manner at constant pressure? Our gas law tells us that air thus heated would expand to a greater volume than its original volume. This would mean that a cubic foot of the heated expanded air would weigh less than a cubic foot of the room air. This heated lighter air rises from the radiator exactly as a lighter-than-air balloon rises from the earth. As this warm air rises toward the ceiling, leaving the space between the coils of the radiator, it is replaced by air from the portion of the room near the floor. This room air at perhaps 70° now comes in contact with the radiator, becomes heated and hence lighter in weight or lower in density. The second batch of air that has been heated rises in its turn as soon as its temperature has been increased. Thus, cool air is removed from the floor of the room by natural circulation and warm air is caused to circulate upward from the radiator and across the ceiling as indicated in Fig. 8.

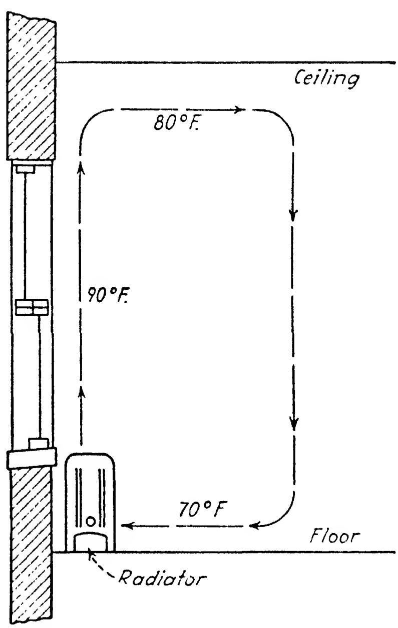


FIG. 8.—Transfer of heat by convection.

It is difficult to give a brief definition of heat transfer by convection. From the example in the preceding paragraph you may write your own definition, or you may adopt one something like the following: "When we apply heat by means of conduction or radiation to some point in a system or circuit containing a gas or a liquid, we then cause natural circulation of the liquid or gas. The heated liquid or gas travels elsewhere in the system, carrying the heat that it itself contains. After giving up its heat to another portion of the system, the gas or liquid returns by natural circulation to be reheated by conduction and radiation."

This definition speaks of a circuit or system and also mentions that either a liquid or gas may be used. This may seem confusing, but it is necessary if our definition is to cover all examples of heat transfer by convection. Do not worry for the moment about using either a liquid or a gas, as we shall soon give you an example of heat transfer by convection using water. Also do not be concerned about the use of the phrase "system or circuit." In our example of convection or currents in a room, our room was an enclosed system; we assume that air does not escape from the room. By "system or circuit" we mean the walls, pipes, or combination of walls, pipes, and tanks that confine the gas or liquid that we are using to transfer heat by convection. Let us briefly present several other examples regarding heat transfer by convection so that we can form a clear picture of this important means of heat transfer.

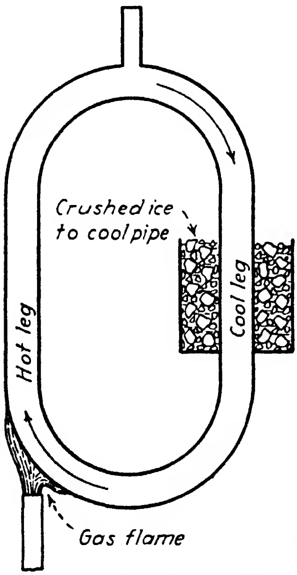


Fig. 9.—Flow by convection in a closed circuit.

Figure 9 gives the simplest illustration of convection flow in a closed circuit. Imagine that the loop of piping shown in Fig. 9 contains ordinary water. Before lighting the gas flame, the water in the right-hand vertical section of Fig. 9 is cold and so is all the rest of the water contained in the circuit. Therefore, all the water in the circuit is at the same density.

Now let us light the gas flame beneath the left-hand vertical section of pipe. Soon the gas flame will heat the water in this "hot leg" of our loop. As soon as the temperature of this water is increased, it expands so that its density becomes less. Remember that when we say the density of a substance we mean its weight per cubic foot or its weight per cubic inch. Now that we have warmed the water in one leg of our circuit, we have made that water lighter in weight than the cold water in the opposite leg of our circuit. Since we have a closed piping circuit, you can see that we have the effect of an unbalanced seesaw. The water in our cold leg being heavier than our water in the hot leg opposite, circulation will start in the direction indicated by the arrows. The heavier water in the cold leg will drop, and the lighter water in the hot leg will rise.

For a practical application of this principle, refer to Fig. 10. This illustrates the conventional domestic hot-water tank installation. In considering this hot-water tank, let us imagine at the start that all the water is cold. When we light the gas flame, convection circulation starts exactly as in the loop that we have just studied. However, in the present case the coil of pipe above the gas flame is our hot leg and the tank itself is our cold leg. After this system has been in operation for 5 or 10 min, the upper section of the tank is filled with hot water.

This hot water does not mix with the cold water below because it is lighter. Furthermore, the convection flow takes place very slowly and there is no agitation or stirring action. However, even if the tank is half-filled with hot water, the heaviness of the cool water in the bottom half of the tank is sufficient to cause convection circulation to continue. As you know the tank will finally become completely filled with hot water. At this point, the gas flame would be shut off either automatically or manually.

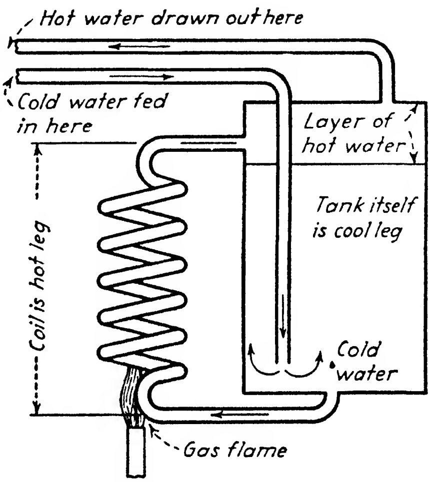


FIG. 10.—Conventional hot-water tank installation.

After a considerable quantity of hot water is drawn out of this tank through an open hot-water faucet, cold water flows into the tank to replace the hot water taken out. The cold water is led by means of a pipe inside the tank, *down to the bottom of the tank*. The discharge of this pipe is equipped with a screen or with baffles to prevent any agitation or stirring. In this manner we can draw off half a tankful of hot water, after which we shall still have the top half of the tank full of hot water and the bottom half of the tank filled with the new cold water. After this the gas flame could be started again at any time, and convection flow will occur.

A further practical illustration of convection hot-water flow is shown in Fig. 11. Here the hot legs are kept hot by means of heat that is added to the water in the boiler. This hot water then gives up Btu and cools in the radiator. Hence the water in our cool leg is lower in temperature and heavier so that convection flow takes place. (*Note:* This type of heating system, although popular 15 years ago, is obsolete today. As you will learn in a later chapter a small circulating pump is used to assist the convection or gravity circulation.)

A warm-air heating installation without a fan is another practical illustration of convection circulation. In this case the warm ducts or pipes form the hot leg; the rooms in the house, including the stair well between the first and second floor, form the cool leg. (With the improvement in fans and with the development of winter air conditioning, this type of system has also become practically obsolete today. We shall study the modern version of this heating system in a later chapter.)

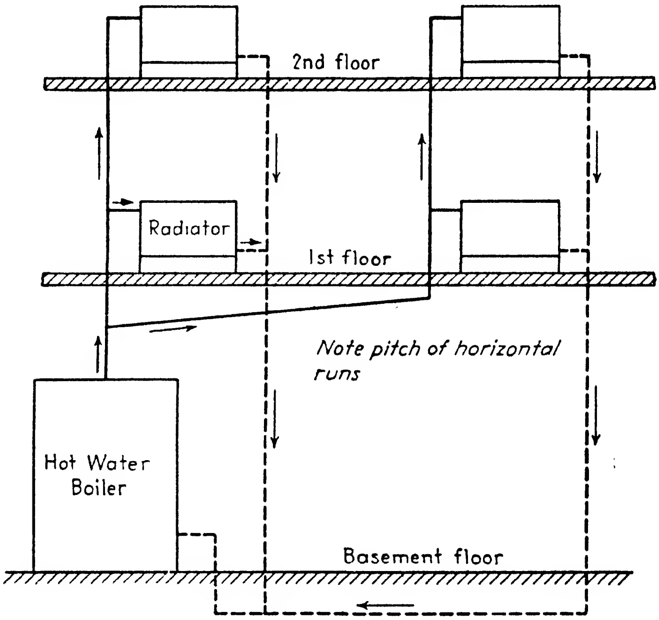


FIG. 11.—Typical hot-water heating system.

Sensible and Latent Heat. We have already made clear by means of the kinetic theory of gases that a considerable addition of heat is necessary to change a liquid to a gas. Since our next chapter is a detailed study of steam and its properties, we want to define certain quantities. We have special names to refer to heat that raises the temperature of a body without affecting its state, and for heat that changes the state of a body without affecting its temperature.

Sensible heat is defined as heat that increases the temperature of a body without effecting a change of state.

Latent heat is defined as heat that changes the state of a body without affecting its temperature.

You can see that these names are very logical. For instance, suppose that you have heated 1 lb of water from 70 to 150°F. This heat which you have added could be "sensed" or felt either by one's hand or by a thermometer. On the other hand, suppose that you have boiled at atmospheric pressure 1 lb of water at 212°F into 1 lb of steam at 212°F. Although you have added a large quantity of heat, almost 1000 Btu, the effect of this added heat could not be felt or measured by a thermometer.

In our definition of latent heat you see that we do not say whether it was changing the state of a body from solid to liquid or from liquid to gas. This is because it is latent heat regardless of which of these cases we may have. When latent heat causes a change of state from a liquid to a gas, we call it "latent heat of vaporization." When latent heat causes a change of state from a solid to a liquid, we call it "latent heat of fusion." In the next chapter you will find an interesting graph showing the heat content of water from ice at 30°F to steam at above 212°F. Sharp breaks or "jumps" in this graph illustrate the latent heat of fusion and the latent heat of vaporization.

PROBLEMS

1. Define "heat." In what unit do we measure heat? What is the difference between heat and temperature? What is meant by "specific heat"? Is the specific heat of a given substance always the same?

2. Using the following symbols, write an equation that will give the quantity of heat added to or removed from a body during any temperature change. State what each symbol represents.

$$Q$$
$$c$$
$$w$$
$$T_1$$
$$T_2$$

3. Suppose that you have 100 gal of a refrigerating brine that weighs 11.5 lb per gal and has a specific heat of 0.65 (typical calcium chloride brine). If this brine cools from 35°F to −5°F, how many Btu are removed from the brine?

4. Define "work" in words and by means of an equation. How much work is done when a steel erecting hoist lifts a 12-ton girder 400 ft vertically?

5. If the steel hoist mentioned in Prob. 4 required half an hour to lift the girder, what horsepower engine would be required if the hoist mechanism had an efficiency of 65 per cent? At the same efficiency, what horsepower engine would be required to lift the girder in 20 min?

6. If an automobile weighing 4,200 lb falls from a cliff 120 ft high, how much heat will appear when the car crashes? (The work that the falling car does in crashing is the same as that which would be required to hoist the car vertically to the top of the cliff.)

7. Heat may be transferred by conduction, convection, and radiation. Define these three forms of heat transfer.

8. Define sensible heat and latent heat.

CHAPTER VI

THERMODYNAMIC PROPERTIES OF STEAM

Introduction to Steam Tables. You will remember that we discussed the changing of a liquid to a gas upon the application of heat. At that time we learned that many substances can exist as gases, and even as solids, even though we are most familiar with them as liquids. It is our purpose in this chapter to study one particular substance—water. Water can exist in a solid form (*ice*), in a liquid form as we usually find it, or in the form of a gas, which we call steam. In order to avoid confusion we shall, from now on, refer to water by the chemist's name, which is H_2O. We shall do this in order to have a name for this substance that will be true, regardless of its *state*. If we did not make this distinction now, confusion would arise because we think of "water" as the liquid form of H_2O only.

The boiling of water is familiar to all of us in the ordinary case of boiling in an open kettle on a stove. You are aware of the fact that cold water must be heated for a considerable period before it begins to boil; also, many of you have had the experience of cooking frozen foods. Frozen foods contain a great deal of water, so that when you drop a frozen brick of green peas into a kettle of boiling water, it has the same effect as if you had dropped a lump of ice into the boiling water. That effect is temporarily to stop the boiling. After the ice (in the green peas) has melted and the entire cold mass has been heated, boiling begins again.

We now wish to use this general knowledge as well as that which we discussed previously. This makes clear the study of the actual properties of H_2O under all the various conditions that have been investigated and tabulated by scientists. Throughout this chapter, references will be made to columns in the Steam Tables, which appear in the Appendix.[1] To learn this material most thoroughly and easily, we suggest that you refer to these tables while studying the text.

"Sensible Heat" or "Heat of the Liquid." You will recall that the definition of 1 Btu is "the amount of heat required to raise the temperature of one pound of water one degree Fahrenheit." When we boil cold or cool water, we find it necessary to raise its temperature. To raise this temperature we must add sensible heat to the water. Knowing the specific heat of water, which is approximately 1.00 Btu, we can calculate the amount of sensible heat that must be added to 1 lb of water in order to raise its temperature 10°, 50°, 100°, or any amount that we may wish to find.

[1] See Tables I and II. Tables with roman numerals are given in the Appendix.

This sensible heat, which we add to water in order to raise its temperature, is given a special name in this case. That name is "heat of the liquid," or "heat of the fluid." In the Steam Tables, we represent this quantity by the symbol h_f.

Fortunately, we do not have to calculate these h_f values for heating water. This information is given in carefully worked out Steam Tables. The data in these tables are the result of an extensive research study made by Prof. Joseph H. Keenan and Prof. F. G. Keyes, of the Massachusetts Institute of Technology.

In order to show these h_f and other heat values most conveniently in the tables, it has been decided to assume a starting point or "reference point" at which we agree to call h_f for water *zero*. This point is taken at 32°F. In other words, we know that 1 lb of H_2O at 32°F really does contain some heat. However, we pretend that it contains zero heat or zero Btu at this condition. Now, if we were to use our formula

$$Q = W(T_2 - T_1)c \quad \text{(here } Q \text{ is the same as } h_f\text{)}$$

to calculate the heat added in heating 1 lb of water from 32 to 50°F we have

$$Q = h_f = 1(50 - 32°)1.00 = 18 \text{ Btu}$$

Similarly, we can calculate other h_f values, always assuming that the specific heat (c) of water is exactly 1.00, and make a table as follows:

Temp., °F	Calculated heat content, Btu per lb	h_f from Steam Tables—exact value
0	0	. . .
50	18	. . .
100	68	. . .
104	72	. . .
150	118	. . .
200	168	. . .
212	180	. . .

However, the specific heat of water is not exactly 1.00 so that there are slight errors in the values that we have just calculated. All the exact values are given in the Steam Tables. After reading the next paragraph, fill in the exact h_f values from the Steam Tables in the blank spaces in the above table.

Look on the first page of Table I.[1] Refer to column (7) in the table under the heading "Enthalpy, saturated liquid, h_f." You will note that the value in the table opposite 32°F is 0 Btu. *For any temperature above* 32°, you may read from this table that h_f, or sensible heat in Btu, that would be required to heat 1 lb of water up to that temperature. For instance, note that at 50° the value is 18.06 Btu. At 104° the value is 71.91 Btu, and at

212° h_f equals 180.00 Btu. Be sure that you read and check these three values in the tables. Now fill in the blanks in the table in the preceding paragraph.

Latent Heat of Vaporization. You will recall from earlier chapters that whenever a substance changes its state from a liquid to a gas this change is made possible by the addition of heat to that substance. In the liquid state the molecules of the substance are closely bound together and can be imagined as vibrating back and forth through limited paths. When the substance changes state to a gas, the addition of a large amount of heat energy is required to increase the kinetic energy (energy of motion of these molecules). When this kinetic energy is increased sufficiently, the molecules can break loose and fly off on free paths, the substance then being a gas. For instance, with water boiling on the stove at an absolute pressure of 14.7 psi at a temperature of 212°, the latent heat of vaporization is 970.3 Btu for 1 lb of water.

Column (8) of Table I of the Steam Tables shows these latent heats of vaporization for water under different conditions. This column is headed by the symbol h_{fg}. You can consider this as meaning the heat from *fluid* to *gas*, if it helps you to remember the symbol. Below are listed some latent heat values taken from Table I. Be sure that you can locate these values in the table and check the latent heat (or h_{fg} value) given.

TABLE 1

Boiling Temperature, °F	Latent Heat of Vaporization, h_{fg}
32	1075.8
150	1008.2
212	970.3
300	910.1
400	826.0
500	713.9
600	548.5
650	422.8
675	332.6
700	172.1
705.4	0

See Fig. 1 for a graph of the data in this table.

With regard to Table 1, we want to call your attention especially to two things: (1) Note that the latent heat decreases as the boiling point of the water rises from 32 to 705.4°F. (2) We want to emphasize that these h_{fg} values do not represent the *total* amount of *heat* contained in 1 lb of water above 32°F after it has been boiled. Instead, they represent just what the symbol h_{fg} implies, *i.e.*, the heat required to change 1 lb of liquid water, *which is already at the boiling temperature*, from a liquid to a gas.

It would now be possible for you to find the *total heat* for any of these conditions by simply adding the heat of the liquid h_f to the latent heat of vaporization h_{fg}. However, this value is needed so frequently and is so important in work with steam that the Steam Tables show this value in

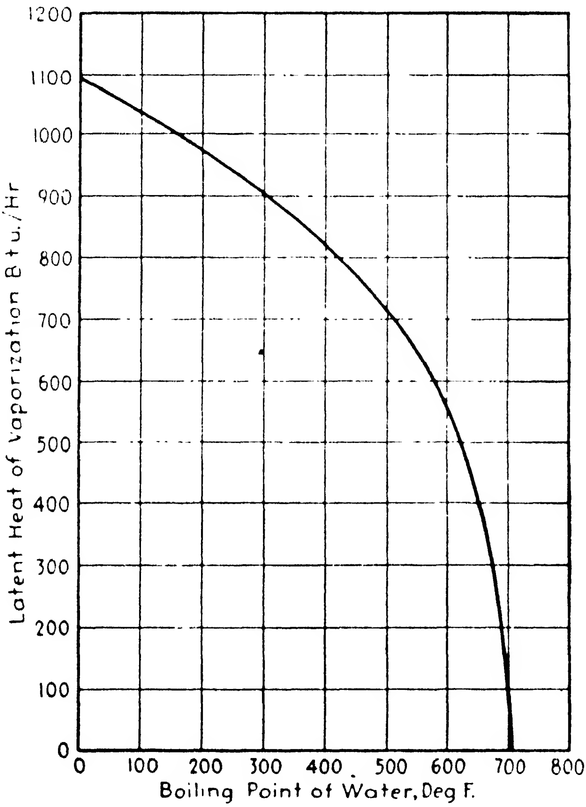


FIG. 1.—Latent heat vs. boiling point of water.

column (9). This column is headed h_g, which you can remember as meaning *total heat of the gas* or *heat of the vapor*. In other words, we may write the mathematical expression

$$h_f + h_{fg} = h_g = \text{total heat} = \text{enthalpy} \qquad (1)$$

This is the clearest way to show this relationship.

In the left-hand column on the next page, five conditions are listed in which we commonly find the substance H_2O. In the right-hand column, in order to make the picture absolutely clear, the *state* of the substance is shown in each case.

Ice	Solid state
Water	Liquid state
Wet steam	Mixture of liquid and gas
Dry saturated steam	Gaseous state
Superheated steam	Gaseous state

Of the above, you are of course familiar with water and ice, but per aps the other three cases should be defined.

Dry Saturated Steam. This is H_2O that contains all the required heat of the liquid and *all* the required heat of vaporization to cause it completely to change state from a liquid to gas. However, it does not contain any additional heat whatsoever.

Wet Steam. When a quantity of H_2O has all its heat of the liquid added and only a portion of its latent heat of vaporization, the result is a mixture of some dry saturated steam and some small particles of liquid water, both being at the same temperature. This mixture is known as *wet steam.* For instance, if to 1 lb of water we add all the h_f and then one-half the h_{fg} that would be required to produce dry saturated steam, we boil away only one-half of our pound of water. Our resultant wet steam is then said to have 50 per cent quality. If we boil away 75 per cent of our liquid water by adding 75 per cent of the h_{fg} that would be required to boil it completely, the resultant mixture is said to be 75 per cent quality. Wet steam of 75 per cent quality is also described as being "75 per cent dry" or as being "25 per cent wet."

Superheated Steam. When a pound of water contains all its h_f and all its h_{fg} for its particular temperature and pressure conditions, we have seen that it is called "dry saturated steam." If this steam then is heated further, *i.e.*, if more Btu are put into this steam, the steam becomes superheated. All the Btu in addition to those contained in the dry saturated steam are called superheat. Since the superheated-steam tables are somewhat different in their arrangement from the pressure and temperature tables, we shall not discuss the use of superheated-steam tables until later in this chapter. The total heat contained in superheated steam is represented by the symbol h.

To summarize the foregoing discussion regarding the heat properties of steam, let us tabulate the various heat values:

Heat of the liquid or heat of the fluid	h_f
Latent heat of vaporization or heat required to change from fluid to gas	h_{fg}
Total heat of the vapor or of the dry saturated gas	h_g
Superheat (or sensible heat added to dry saturated steam). There is no letter given to represent these additional Btu of superheat. Instead, the symbol h is used to represent the total heat, $h_f + h_{fg} +$ Btu of superheat, that is contained in superheated steam.	

Expressed as an equation this means

$$h = h_f + h_{fg} + \text{superheat Btu}$$

or

$$h = h_g + \text{superheat Btu} \tag{2}$$

If you wish to find the total heat contained in some wet steam, it is obtained by the following formula:

$$H = h_f + (h_{fg} \times \text{per cent quality}) \tag{3}$$

If you wish to find the total heat contained in any dry saturated steam, it is not necessary to add h_f and h_{fg}, as taken from the Steam Tables, but instead, you may look up directly the value h_g. The total heat of the superheated steam as represented by the value h may be found directly from the superheated-steam tables, which we shall discuss later.

Specific Volume. For any substance the *specific* volume is the volume in cubic feet of *one pound* of that substance at given temperature and pressure conditions. Since steam is a substance and liquid water is also a substance, one would expect to find data about the specific volume of both water and steam. This is true, and these values are given in columns (4), (5), and (6) of Table I.[1] These columns are headed v_f, v_{fg}, and v_g. The meaning of these symbols is similar to that of the symbols for total heat; *i.e.*,

v_f = specific volume of the liquid H_2O or water
v_{fg} = change in volume (increase) during evaporation or boiling
v_g = total volume of the dry saturated vapor
v = total volume of superheated steam

With regard to these three symbols, we again have an equation for dry saturated steam:

$$v_g = v_f + v_{fg} \tag{4}$$

For wet steam we have the equation

$$\text{Specific volume} = v_f + [v_{fg} \quad (\% \text{ quality})] \tag{5}$$

For superheated steam we have the equation

$$\text{Specific volume} = v = v_g + \begin{array}{l}\text{increased volume due to} \\ \text{added (sensible) superheat}\end{array} \tag{6}$$

The values for entropy given in columns (10), (11), and (12) of the Steam Tables are not necessary for us in our study of steam and are beyond our scope. Later, when we study refrigerants, we shall have to work with entropy, but we shall neglect this quantity entirely in our study of steam.

Temperature-pressure Relationship. We have now discussed all the columns in the Steam Table except column (2) in Table I. This column is

[1] Tables with roman numerals are given in the Appendix.

headed "Absolute pressure, psi." This brings us to a very important relation. Imagine a kettle boiling on a stove, the spout open, and all air having been driven out by the steam. In this case you know that the temperature of the boiling water would be 212°F, and you also know that the pressure within the kettle is 14.7 psia. Now imagine that the spout of the kettle is partly plugged, the lid being fastened tightly. This would cause the pressure inside the kettle to rise somewhat; let us say that it rises to 20 psia. A thermometer inside the kettle would then read almost exactly 228°F. Let us now imagine a third case in which a vacuum pump is connected to the spout of the kettle, drawing away the steam as fast as it is formed. Let us imagine that this vacuum pump reduces the pressure within the kettle to 1 psia. Under this condition the temperature within the kettle would then read almost 102°F.

Let us tabulate the pressures and temperatures cited in the previous paragraph:

$$P = 20 \qquad T = 228°$$
$$P = 14.7 \qquad T = 212°$$
$$P = 1 \qquad T = 102°$$

We have been careful to state that the air must have been driven out of the kettle so that we have steam alone above the water. This is so that we will not meet complications of Dalton's law on partial pressures because of having a mixture of gases in the kettle. Now, so long as there is steam alone above the water, there is *one and only one temperature* that corresponds to *each pressure.* Similarly, for dry saturated steam above water, there is *one and only one pressure* that corresponds to each *boiling temperature.* This is one of the most important fundamental laws in connection with steam; therefore we have illustrated it by means of a rough graph (Fig. 2).

Turn to the first page of Table III in the Appendix and you will find the first three columns from the left having titles as follows:

Column (1)	Column (2)	Column (3)
Absolute pressure, psi	Saturated water	Saturated steam

Looking down column (1), notice the bold-faced figures; notice also the ordinary figures enclosed in parentheses shown just below each bold-faced figure. The bold-faced figures show absolute pressure in pounds per square inch as indicated by the upper portion of the title of this column. The figures in parentheses are merely reprinted from Table II. They show the saturation temperature corresponding to each pressure. The use of parentheses around the lower portion of the title to this column (saturation temperature) is to make it clear that this part of the title refers to the saturation temperatures, which are also shown in parentheses.

Now, note columns (2) and (3) of the table. Here again information is reprinted from Tables I and II. You will note that for each pressure we have values for the following, each of which is indicated by a letter or letters.

v.	Specific volume
h.	Total heat, Btu per lb
s.	Entropy

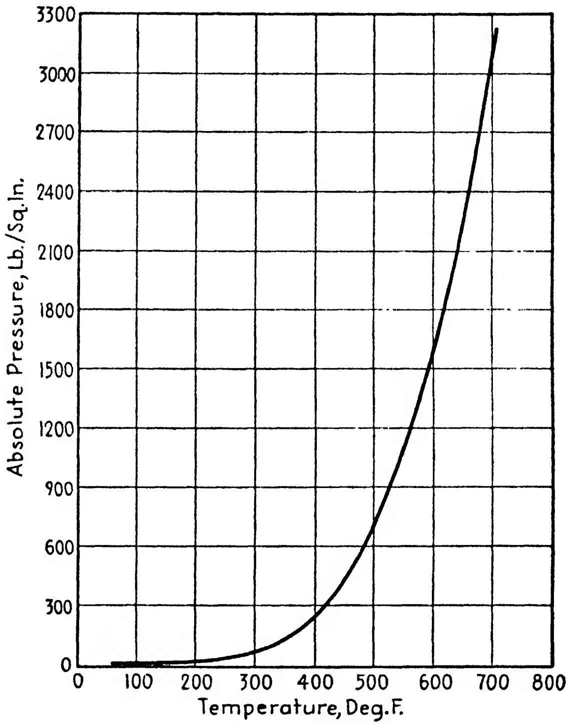


FIG. 2.—Boiling-point diagram for water.

Of these we have decided that we are going to omit the entropy value from consideration. The specific volume and total heat values we have already discussed. The only new value is that of "superheat, °F." If steam has been superheated such that its temperature is 100° above the saturation temperature at its given pressure, it is then said to have 100° of superheat. Do not confuse this term "superheat," which is measured in degrees Fahrenheit, with *heat*, which is always measured in Btu. In other words, the degrees of superheat and the Btu of superheat are two entirely different things.

Note that in columns (2) and (3) there is no value shown for degrees of superheat. This is because, as mentioned above, the values in these two columns are not connected with superheated steam but are reprinted from Tables I and II on saturated steam. Hence, *in these two columns only*, the values of v mean v_f in the saturated-water column and v_g in the saturated-steam column. Similarly, *in these two columns only* the values given opposite h mean h_f in the saturated-water column and h_g in the case of the saturated-steam column. These values, which are reprinted from the saturated tables, as well as the saturation-temperature figures shown in parentheses, are reprinted in the superheated-steam tables so that it will not be necessary for the user of the tables to turn back to the previous table.

To use the superheated-steam tables, one must be able to locate the proper small square of figures in the middle part of any page in the tables. In the simplest case this may be done if we know the temperature and the absolute pressure. Suppose the absolute pressure is 20 psi and the temperature is 400°F. Run your finger down the left-hand column to 20 psi abs, then to the right until you reach the column under "400°" at the top of the page. You will then read the following:

v	25.43
h	1238.3
s	1.8386

Knowing the temperature and the pressure is the simplest way of finding values in the superheated-steam tables. These values can be found by many other ways, of which we shall discuss a few.

Example. Refer to the figures given in the preceding paragraph for superheated steam at 20 psia at 400°F. Suppose now that you were given the following data and were asked to find values in the superheated-steam tables.

1. Steam having a saturation temperature of 227.96°F
2. This steam having 172.04° of superheat, or 227.96° + 172.04° = 400°F

From these data you can immediately determine that the pressure of the steam is 20 psi abs by looking in column (1). Next, move to the right in the table, noting the values given for temperature, °F until you reach 400°. You have located the same group of figures as we found in the preceding paragraph by a slightly different method.

Similarly, if we have any information that enables us to determine the absolute pressure of the superheated steam, we can find all information concerning that sample of steam, provided we know any one of the following: (*a*) its temperature in degrees Fahrenheit, (*b*) its specific volume, and (*c*) its total heat.

The methods required to locate the proper vertical column in Steam Table III are the same with any one of the three pieces of information listed above. In all three instances you locate the proper column by substantially the same manner as explained in the example above.

Interpolation. So far in our use of the Steam Tables and in our use of all other tables in the text, we have always looked up values of data that we can find *given exactly* in the tables. Suppose that you try to find the properties of saturated steam at 63 psia. When you look at the table, you will find that it does not show this absolute pressure of 63 psi. It shows data for 60 psia and for 70 psia. For a simple case such as this, you must take values three-tenths of the way between the values that are given in the table, because 63 psia is three-tenths of the way between 60 and 70. This process is called "interpolation."

You will remember that in our chapter on graphs we called the extending of a graph beyond either end by the name "extrapolation." Interpolation means estimating from points of a curve in between two known values of data. Interpolation also means estimating values in various tables which are in between values actually present in the tables. Beyond the sample illustration given in the preceding paragraph, the best method of studying interpolation is to take a concrete example.

Example. Let us find the properties h_f, h_{fg}, and h_g for dry saturated steam at a pressure of 63 psia. Looking in our Steam Tables we find values given for 60 and 70 lb. Let us, therefore, set these values up in a table for ourselves in which we shall interpolate.

	p	t	h_f	h_{fg}	h_g
	60	292.71	262.09	915.5	1177.6
	70	302.92	272 61	907.9	1180.6
Difference	10	10.21	10.52	7.6	3.0

From the foregoing table you see that we have written our heat properties as well as our saturation temperatures for the two conditions printed in the table. We have then done a simple arithmetic subtraction and have found the differences in each property we want to find between the two values known. It is simple to interpolate. We are trying to get properties at 63 psia. The difference in pressure between 60 and 70 lb is of course 10 lb, as shown in the table. Our actual steam represents three-tenths of the way between 60 and 70. We, therefore, interpolate as follows:

$$\frac{3}{10}(10.21) = 3.061$$

$$\frac{3}{10}(10.52) = 3.156$$

$$\frac{3}{10}(7.6) = 2.28$$

$$\frac{3}{10}(3.0) = 0.9$$

The quantities we have calculated in the equations above must be added to the values shown in the table at 60 psia. This will give us the answers to our problem: (Note minus sign in the case of h_{fg}. Why?)

t	h_f	h_{fg}	h_g
292.71	262.09	915.5	1,177.6
+ 3.061	+ 3.156	− 2.28	+ 0.9
295.771	265.246	913.22	1,178.5

As some of you may have realized, we could have performed this interpolation another way. We could have used the fraction $\frac{7}{10}$ instead. If, however, we had taken seven-tenths of our difference for each property, we would have had to subtract from the steam-table data at 70 lb. In our solution we, of course, added to the steam-table data shown for 60 lb. In other words, we have two known points 60 and 70 for which we know all properties. We desire the properties at 63 lb, and these we can find in one of two ways. (1) We can think of them as being three-tenths of the way

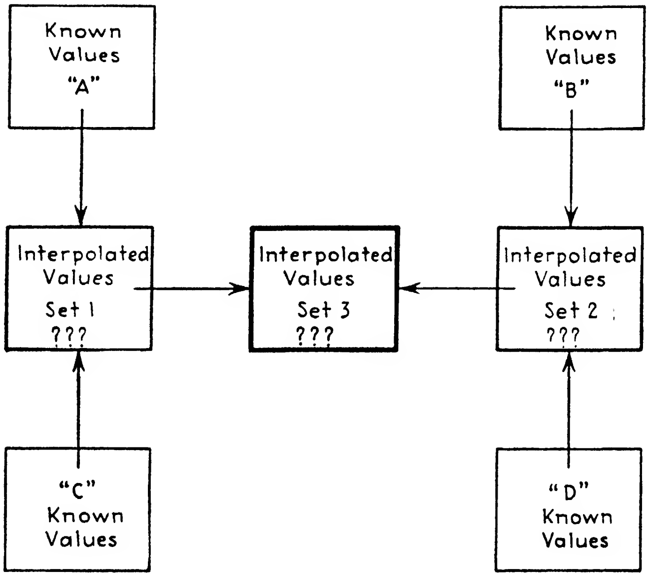


FIG. 3.—Interpolation skeleton diagram.

from 60 *to* 70. (2) We can think of the properties we want as being seven-tenths of the way *from* 70 *to* 60.

In order to interpolate in Table I, the temperature table, the same procedure is followed as described above. In order to test your ability to interpolate in this table, try one of the problems given at the end of this chapter. However, interpolation in the superheated-steam table is more difficult so that we wish to present the solution of such an interpolation in complete detail.

Example. Let us find the number of degrees of superheat, the specific volume, and the total heat content of 1 lb of steam at a temperature of 540°F and at an absolute pressure of 42.5 psia. The other two properties are represented in the tables by the symbols v and h, respectively. Turn to Table III, the superheated steam table, and locate as near as you can the temperature and pressure conditions that we have given. You will find that the nearest temperature values are 500 and 600°. Our interpolation would then be done as shown in the sketch in Fig. 3. Table III does not give the property "degrees of superheat," represented by the symbol *Sh*, but it is easily calculated. Assume that temperatures between 500 and 600° are not tabulated in Table III.

In Fig. 3, you see that the four outside squares are labeled as *known* values; that is because we can find these in the table. From Fig. 3 you can see that the square *A* represents the properties of steam at a pressure of 40 psia and at a temperature of 500°F Square *B* shows values at this same pressure but at a temperature of 600°F.

Squares *C* and *D* show values for properties of steam at 45 psia, square *C* being at 500°F and square *D* being at 600°F. Using Fig. 3 as a model, let us set up Table 2 to show the numbers between which we are going to interpolate. The question marks in Table 2 are the same as set 1 and set 2 of the interpolated values shown in Fig. 3. As the arrows in Fig. 3 indicate, we are going to find the values that we have represented by question marks in our table. In other words, at temperatures of both 500 and 600° we are going to find the properties of steam at an absolute pressure of 42.5 psia. This means two complete interpolation problems.

TABLE 2.—SUPERHEAT INTERPOLATION PROBLEM

(1)	(2)	Temperature, °F		
		500 (19)	540	600 (24)
40 (267.25)	v h	14.168 1284.8		15.688 1333.1
42.5 (270.84)	v h	??? ???	→ XXX XXX ←	??? ???
45 (274.44)	v h	12.580 1284.4		13.935 1332.8

(The saturation temperature at $P = 42.5$ lb is a simple interpolation between the saturation temperatures at 40 lb and 45 lb). After having completed these two interpolation jobs, we have the properties of steam at the pressure we wish, but our temperature of 540° is in between the temperatures of 500 and 600°. Therefore, we have a third interpolation to perform. The complete solution of this problem is presented below.

Solution. First, we interpolate in the 500° column, proceeding as we did for one ordinary simple interpolation.

Taking differences at 500°, column (19),

	v	h
	14.168	1284.8
	−12.580	−1284.4
Difference =	1.588	0.4
½ difference =	0.794	0.2

We have taken one-half of these differences because 42.5 lb is one-half of the way between 40 and 45. We must now subtract these from the original known values for 40 psia, thus:

14.168	1284.8
− 0.794	− 0.2
13.374	1284.6

The same process is followed for our interpolation in the 600° column, column (24). The results of these two interpolations can then be tabulated, corresponding to the question marks in Table 2.

v	13.374	XXX	14.812
h	1284.6	XXX	1332.95

We now interpolate to get the values marked XXX in Table 2 and in the table above. Since our actual steam has a temperature of 540°F, our new values are four-tenths of the way between the values we have just calculated for 500 and 600°, respectively.

We now must interpolate between the values for 500 and 600° which are shown at the top of this page. We must get four-tenths of the differences in the values shown above, thus:

$$4/10 \text{ difference for } v = \frac{4}{10}(14.812 - 13.374) = 0.575$$

$$4/10 \text{ difference for } h = \frac{4}{10}(1332.95 - 1284.6) = 19.3$$

Adding these to the values at 500° we get our final answer.

For steam at 540°F and 42.5 psia, the properties are

$$v = 13.949 \text{ cu ft per lb}$$
$$h = 1303.9 \text{ Btu per lb}$$

To get the "degrees of superheat" of the steam is easy and could have been done at the start. The saturation temperature is 270.74°F (see Table 2). The steam is actually at 540°F, so

$$Sh = 540 - 270.74 = 269.26°$$

The foregoing examples should have made you familiar with the various types of interpolation that are necessary in using the Steam Tables. There is one fact that we wish to mention now in order to simplify your use of the Steam Tables. For any gauge pressures higher than 0 psig, add 15.0 instead of 14.7 to convert from gauge pressure to absolute pressure. This error of 0.3 psi is slight, and the use of the even figure 15.0 will avoid a great deal of unnecessary interpolation that might be caused by using the correct value of 14.7. For instance, if you had a gauge pressure of 5 psig, this could be called either 19.7 or 20.0 psia. Obviously, the 20.0 value makes for much easier use of the Steam Tables. You may use this short-cut in all ordinary steam problems for the pressure of steam itself, when you do not have steam mixed with other gases.

Let us now consider a practical steam problem such as is encountered in the design of most large steam-heating systems and live-steam systems in industry.

Example. Suppose we have a radiator that needs 22,000 Btu of heat per hour in order that it may give this heat to a certain drying room. The radiator is connected to a steam boiler by means of piping as shown in Fig. 4. In the steam main AB, 10 per cent of the steam sent out of the boiler condenses and runs back to the boiler as water, never reaching point B. Referring to points A, B, C, and D, marked on the piping in Fig. 4, the steam conditions at various points are as follows:

TABLE 3

Station	Gauge pressure	Absolute pressure	T, °F	Quality	h_f	h_{fg}	h_g
A......	5	..	..	95%	..	..	..
B......	5	..	..	95%	..	..	..
C and D.	5	..	..	0	..	0	0

Find:

1. How many pounds of steam must enter and pass through the radiator each hour?

2. How many pounds per hour of steam must leave the boiler and start up the steam main past point A?

3. In part 2 you found how many pounds per hour of steam left the boiler and you know that this steam reenters the boiler as water, some at point A and some at point D. Find how many pounds of water per hour enter the boiler at A and at D and also the temperature and heat content of the water at these two points.

4. If the boiler operates at an *over-all efficiency* of 60 per cent and burns coal having a heat value of 12,000 Btu per lb, how many pounds per hour of coal will be burned?

5. Assume that proper insulation of the steam main will stop *all* heat loss in that pipe, thus preventing the condensation of steam in pipe $A - B$. How many pounds per hour of coal could be saved by insulating this pipe?

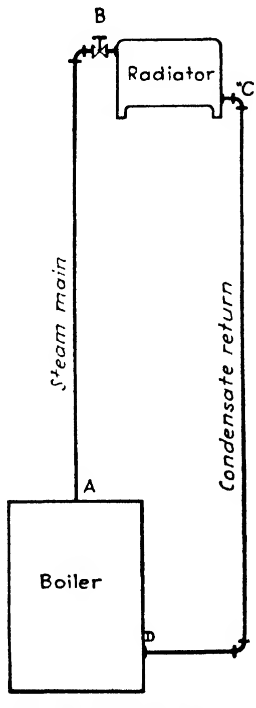


FIG. 4. — Simple steam-heating hookup.

In solving this problem, we suggest that you spend about 1 hr working on it before looking at the solution given below. As to general procedure, we shall make a few suggestions. (1) Fill in the blanks in Table 3, which gives you part of the data needed for the problem. When you fill in these blanks, be careful that you calculate h_{fg} and h_g on a basis of steam at 95 per cent quality instead of keeping the values shown in the table for 100 per cent quality. (2) Remember that if 10 lb of steam left the boiler, only 9 of them reach point A at the entrance to the radiator. (3) Remember your definition of efficiency; in the case of this boiler you only *get out* (in the form of steam) 60 per cent of the energy that you *put in* in the form of coal. With this help you should be able to get a reasonably correct solution to this problem in considerably less than 1 hr. With practice, you can complete and check a solution like this in about 15 min.

Solution. Referring to Table 3,

$$h_{fg} = 960, \text{ for steam of } 100\% \text{ quality}$$
$$h_{fg} \text{ (at } 95\% \text{ quality)} = 0.95(960) = 912$$
$$h_g = h_f + h_{fg}$$
$$= 196 + 912 = 1108$$

We also look up the temperatures and h_f value to insert in Table 3. With the spaces filled in, the table should then appear as follows:

Station	Gauge pressure	Absolute pressure	T, °F	Quality	h_f	h_{fg}	h_g
A......	5	20	228	95%	196	912	1108
B......	5	20	228	95%	196	912	1108
C and D.	5	20	150	0	118	0	0

1. The radiator requires 22,000 Btu per hr; each pound of steam entering the radiator (*at point B*) contains 1108 Btu and then leaves as water having 118 Btu. Hence, 1108 − 118 = 990 Btu given up in radiator by each pound of steam.

Therefore

$$\frac{22{,}000}{990} = 22.2 \text{ lb steam per hour} \quad \textit{Ans.}$$

2. Let x = steam that *must leave the boiler* if 1.00 lb is to *reach the radiator*, 10 per cent being lost in pipe AB. Then

$$0.90(x) = 1.00$$
$$x = \frac{1.00}{0.90} = 1.11 \text{ lb}$$

and

$$22.2(1.11) = 24.6 \text{ lb steam per hour must leave boiler} \quad \textit{Ans.}$$

3. The water running back through pipe BA to the boiler is at 228°F; the water running back through pipe CD is at 150°F. We then have

At A: (24.6 − 22.2) = 2.4 lb per hr of water

At D: Just what goes through the radiator, 22.2 lb per hr of water. From the steam tables (data already tabulated),

At A: h_f = 196 Btu per lb
At D: h_f = 118 Btu per lb

4. *Leaving* the boiler we have 24.6 lb of steam with a total heat content of h_g = 1108 Btu per lb.

Entering the boiler we have, from part 3, water entering the boiler as follows:

At A: 2.4 lb per hr, h_f = 196 Btu per lb
At D: 22.2 lb per hr, h_f = 118 Btu per lb

We set up an equation that says

Total heat in steam leaving − total heat in water entering =?

The heat in the water entering the boiler is

$$2.4(196) + 22.2(118) = 3090 \text{ Btu per hr}$$

Then

$$\begin{aligned} Q &= 24.6(1108) - 3090 \\ &= 27{,}257 - 3090 \\ &= \text{Btu per hr } 24{,}167 \end{aligned}$$

For slide-rule work we "round off" this figure and write

$$Q = 24{,}200 \text{ Btu per hr}$$

This is the heat added by the boiler to the water in heating and boiling the water. From each pound of coal we actually get as useful heat 60 per cent of 12,000 Btu; so

$$\frac{24{,}200}{0.6(12{,}000)} = 3.375 \text{ lb of coal per hour} \quad \textit{Ans.}$$

5. Each hour pipe loss causes the boiler to take in 2.4 lb of water at 228°F ($h_f = 196$) and to drive this off as 95 per cent quality steam ($h_f = 1108$). Hence pipe loss Q is easy to get.

$$\begin{aligned} Q &= 2.4(1108 - 196) \\ Q &= 2.4(912) \\ &= 2189 \text{ Btu per hr} \end{aligned}$$

and

$$\frac{2189}{0.6(12{,}000)} = 0.304 \text{ lb per hr coal saving} \quad \textit{Ans.}$$

There are several other steam problems given on the problem sheet at the end of this chapter. Solve as many of these problems as you feel are necessary to make you familiar with the Steam Tables and to give you an understanding of the properties of steam. This knowledge is important of itself, but more than that it is the foundation of our study of water vapor in air.

PROBLEMS

Note: In solving the following problems, read the problem over *twice* and draw a sketch that clearly illustrates the statement of the problem. Next, look up all required information in the Steam Tables. You are then ready to set the Steam Tables aside and proceed with the solution of the problem.

1. A boiler is fed with city water at a temperature of 60°F and produces dry saturated steam at 100 psig. If this boiler has an output of 400 lb of steam per hour, find the total heat added per hour to the water that passes through the boiler. If "feed" water at a temperature of 180°F could be substituted for the cold city water, how much saving (reduction) could be effected in the heat quantity you have just calculated?

2. Suppose that you desired to cool water at the rate of 10 lb per min from 70 to 60°F by evaporation of a portion of the water in a vacuum chamber. What vacuum must be maintained? For each 10 lb of water cooled, approximately how many pounds of water must be evaporated?

3. A boiler delivers steam at 92 per cent quality and at a pressure of 15 psig. This steam is then condensed in heating coils and returns to the boiler as water at 150°F. Between the boiler and the heating coils, 5 per cent of the *total quantity* of steam is lost through condensate returns whence it returns to the boiler as water at 228°F. The heating coils require a total of 900,000 Btu per hr. How many pounds of steam per hour are required at the entrance of the heating coils? How many Btu per hour must be transferred from the combustion chamber into the water in the boiler? Using oil having a heat value of 140,000 Btu per gal and assuming an over-all boiler efficiency of 63 per cent, how many gallons of oil per hour must be burned?

4. Ten pounds per minute of superheated steam at 30 psig and at a temperature of 300°F are condensed in a shell-and-tube condenser by a water flow of 25 gal per min (1 gal of water weighs 8.33 lb). If the cooling water enters the condenser at a temperature of 80°F, what will its temperature be on leaving the condenser? (*Note:* In a shell and tube condenser, the cooling fluid and the gas being condensed do not mix. Hence, in this problem the hot water resulting from the condensation of the steam leaves the condenser by a separate outlet leading into a hot well; it does not mix with the cooling water.)

CHAPTER VII

INTRODUCTION TO PSYCHROMETRY

Psychrometry is the study of air and water vapor. Hence when we say "the psychrometric properties of air," we mean all the properties of air that are affected by the presence of more or less water vapor mixed with the original dry air. Actually, there is no such thing as bone-dry air, or air without the *slightest trace* of water vapor. All the air we find in the atmosphere contains some water vapor mixed with it, although the amount may be extremely slight. Even in the laboratory it is impossible to produce bone-dry air.

Mixtures of Air and Superheated Steam. The best way for you to understand the basic principles involved in studying psychrometry is by working a problem that can be solved by means of Table II[1] of the Steam Tables. Let us take steam at 1 psia and at a temperature of 120°F. From the Steam Tables, the saturation temperature corresponding to this pressure is 101.74°F. Since our steam is at 120°F, it is obviously superheated slightly more than 18°. Now suppose that we cool the steam to 103°F. By such a cooling process, we remove *sensible heat only; i.e.*, we remove most of the superheat in our steam. If we continue cooling the steam, as soon as we get a fraction of a degree below 101.74° some steam condenses back to a liquid and the *pressure decreases.* Remember that when we have steam over water, there is one and only one pressure that corresponds with each boiling temperature. In the past we have called this boiling temperature "saturation temperature," meaning "saturated with latent heat of vaporization, but containing no superheat." We could equally well call these boiling temperature "condensation temperatures," because in cooling steam below these temperatures some of the steam condenses and the pressure drops. Therefore, in the problem we are discussing we can call 101.74°F the boiling temperature, the saturation temperature, or the condensation temperature of our steam.

From our gas laws we know that if we cool such a saturated vapor at constant volume, so that its pressure decreases, the density of the vapor would become less, because part of the vapor condenses out as liquid. In the case of the steam that we are discussing, this would mean that a given volume of space would contain less pounds of steam, and hence, the space would contain less Btu of latent heat. Remember this mention of the

[1] Tables with roman numerals are given in the Appendix.

reduced latent heat per unit volume owing to the reduced density of the steam. We shall have to do with heat content on later pages.

In the foregoing discussion, we have talked of one gas or vapor alone. Suppose, however, that we had steam mixed with nitrogen, the proportions of the two gases being such that the partial pressure of the steam was 1.0 psia. The saturation temperature or condensation temperature of the steam is dependent only upon the partial pressure of the steam itself. Instead of the nitrogen, we may substitute a mixture of nitrogen and oxygen, which we call air, and the problem is still unchanged. Suppose that you have a mixture of air and steam flowing through a duct. Suppose that you know that this mixture is at a temperature of 120°F, a pressure of 14.7 psia, and that it contains 6.8 per cent steam by volume.

In solving this problem we must first recognize that we have Dalton's law to consider. From Dalton's law we can set up certain information about our problem in a table.

	Per cent by volume	Partial pressure
Steam	6.8	1.0
Air	93.2	13.7

The partial pressures of the individual gases composing the air are of no importance to us; so we have simply considered air as if it were one gas. Check the partial pressures shown in the above table to be sure that we have used Dalton's law correctly. Now that we have separated our two gases, steam and air, according to Dalton's law, we can consider one gas separately, forgetting completely about the other. Our steam is at a partial pressure of exactly 1.0 psia. From this you can see that the steam behaves exactly the same in this case where it is mixed with air as it did in the case of steam alone, which we discussed at the beginning of this chapter. The Psychrometric Tables are made up by showing saturated water vapor mixed with air at various temperatures from 0 to 200°F.[1] For each temperature, the corresponding saturation pressure for steam is shown. Therefore, these saturation pressures are in each case the partial pressure caused by the water vapor. The water vapor is always mixed with the proper amount of plain dry air such that the total pressure throughout the entire table is 14.7 psia. As we progress from 0 to 200°F in the table, the partial pressure of the water vapor in moist air increases from 0.0185 to 11.425 psi. The dry air in the mixture is responsible for the remaining pressure such that the total is constant at 14.7, or atmospheric pressure. As the vapor pressure of

[1] See Table V.

the water—the partial pressure of the low-pressure steam—changes, the saturation temperature or *condensation temperature* changes with it. In psychrometry, we call this temperature the "dew point" of the moist air. We use *moist air* or *mixture* to describe a mixture of bone-dry air plus water vapor.

The Psychrometric Tables show this clearly. For each temperature, the vapor pressure of the water vapor is shown which corresponds to that temperature. The relative proportions of dry air and water vapor are then shown in several ways, which we shall discuss shortly. We have our moist air then as a mixture that obeys Dalton's law, the two gases in the mixture being *air* and *water vapor*. The water vapor is *dry saturated steam* for each temperature shown in the tables. From this we can deduce several things:

1. If our moist air as shown in the table is *cooled*, some water vapor will *condense* and fall out as "rain."

2. If our mixture stays at the same temperature, no more water vapor can be added.

3. If we heat our mixture, the original vapor becomes superheated but its vapor pressure remains as it was. We could then add more water vapor, increasing the partial pressure due to the water, because the increase in temperature brings with it a new higher saturation pressure.

First Psychrometric Definitions. Before we begin to study and analyze the Psychrometric Tables, we want to agree upon three definitions pertaining to the mixture of air and water vapor. From now on when we use the word "air" by itself, we always mean air plus whatever amount of water vapor is mixed with it. When we wish to mean the *dry air portion only* of the mixture, we shall refer to it as "dry air." This will make unnecessary the continued repeating of reference to the moisture or water vapor, which we know is always present anyway. The three terms we wish to define now are 100 per cent saturated air, partly saturated air, and dew point of the air. These terms are defined below. The concept of air as a sponge that will hold more or less water vapor, depending on its temperature, is a useful way to think about psychrometry. Therefore, we give definitions on this basis at the right below and the true technical definitions to the left.

When the vapor pressure of the water vapor in air is equal to the saturation pressure corresponding to the temperature of the mixture, the air is said to be 100 per cent *saturated with moisture.*

When air is holding the maximum amount of moisture which it can hold as a vapor at its particular temperature, the air is said to be 100 per cent *saturated with moisture.*

When the vapor pressure of the water vapor in air is less than the saturation pressure corresponding to the temperature of the mixture, the air is said to be *partly saturated with moisture.*

When air is holding less than the maximum amount of moisture which it can hold as a vapor at its particular temperature, the air is said to be *partly saturated with moisture.*

Note: When air is partly saturated with moisture, it may, of course, be one-half, one-quarter, or three-quarters saturated. This degree of saturation is expressed as a per cent and is called *relative humidity.* Do not use this as a definition of relative humidity as we shall give you an exact definition shortly.

The saturation temperature corresponding to the vapor pressure of the water vapor is called the *dew point of the air.*

The *dew point of air* is the lowest temperature to which the air can be cooled without the condensation or "raining out" of some water vapor. If air is cooled a fraction of a degree below its dew point, some water vapor will condense.

Measurements of Pressures in Millimeters of Mercury. The very low vapor pressures of water that we find in psychrometric work are ordinarily measured in millimeters of mercury. For instance, at 35°F the corresponding pressure of water is 0.10 psia. Rather than think of this as $\frac{1}{10}$ psi, it is more generally expressed in terms of millimeters of mercury.

Turn to Table II of the Steam Tables. Note that for absolute pressures below atmospheric pressure the table shows pressures both as "psi" and as "in. Hg." The latter means "inches of mercury" and is used more frequently as a measure of these low absolute pressures. From an earlier chapter you recall this relation,

$$1 \text{ atmosphere} = 14.7 \text{ psi} = 29.92 \text{ in. Hg}$$

Hence:

$$1.0 \text{ psi} = 2.03 \text{ in. Hg}$$

For very small pressures, the height of the mercury column is usually measured in the metric system instead of in inches. The units are called millimeters (abbreviated "mm"), and the equivalents are

$$1 \text{ in.} = 25.4 \text{ mm}$$

$$29.92 \text{ in.} = 760 \text{ mm}$$

$$\frac{1}{25.4} \text{ in.} = 0.0394 \text{ in.} = 1 \text{ mm}$$

Soon we shall be speaking of pressures of only a few millimeters of mercury, so we want to get accustomed to this idea. From the foregoing, keep the equivalents between pounds per square inch and inches of mercury in mind, so that you can *visualize* what we mean when we speak of "an absolute pressure of 10 mm Hg":

$$1.0 \text{ psi} = \text{approximately } 53 \text{ mm Hg}$$
$$10 \text{ mm Hg} = \text{just less than } \tfrac{1}{5} \text{ psi}$$

The Psychrometric Tables. Turn to Table V of the Psychrometric Tables. Let us first study the headings of the columns in the table.

Temp, °F (1)	Pressure of saturated vapor		Weight of saturated vapor			
			Per cubic foot		Per pound of dry air	
	In. Hg (2)	Psi (3)	Pounds (4)	Grains (5)	Pounds (6)	Grains (7)

The temperature figures shown in column (1) represent the temperatures of 100 per cent saturated air. As indicated by its title, this table is based upon 100 per cent saturated air. Reading opposite a temperature of 70°F, we see that columns (2) and (3) give us the partial pressure of the water vapor in 100 per cent saturated air at a temperature of 70°F. If you check these partial pressures by looking opposite 70° in Table I of the Steam Tables, you will see the following comparison:

	Pressure of saturated vapor at 70°F	
	In. Hg	Psi
Steam Tables	0.7392	0.3631
Psychrometric Tables	0.7387	0.3628
Both, to three significant figures	0.739	0.363

You see there is a slight disagreement between the two tables in the fourth figure after the decimal point. We shall simply forget about this as we shall ordinarily use data only to three significant figures, because that is all we can use on our slide rules. The meaning of columns (2) and (3) should be evident after our study of the Steam Tables and our preliminary discussion in this chapter.

Note that the major heading above columns (4), (5), (6), and (7) is the same. In other words, all four of these columns give us the "weight of the saturated vapor," in its different forms. In order to understand these columns we must use one new unit, "grain." One grain is a small fraction of a pound, and the equivalents are

$$7{,}000 \text{ grains} = 1.0 \text{ lb}$$

$$\frac{1}{7{,}000} \text{ lb} = 1 \text{ grain}$$

In column (4) opposite 70°, find the figure 0.0011507; dropping the "07" from the end, this figure is read "one hundred fifteen one hundred-thousandths" pound. You can see that it is difficult to think in terms of such minute fractions of a pound. Even in column (6) you will see that the fractions of a pound are so small that they are difficult to use. In almost all our work we shall not use columns (4) and (6); instead we shall use columns (5) and (7). Looking opposite 70° you find in column (5) 8.055 grains per cu ft, and in column (7) you find 110.2 grains per lb of dry air. These figures can be made to mean something to us; for if we see one sample of air containing 8 grains per cu ft and another containing 12 grains per cu ft, we can instantly recognize that the second sample contains 50 per cent more water vapor than the first sample.

Weight of Water Vapor in Air. As we have said, columns (4), (5), (6), and (7) give the weight of the saturated vapor. Columns (4) and (5) give this on a different basis from columns (6) and (7). Columns (4) and (5) take 1 cu ft of our air, which is, as we know, a mixture of air plus water vapor. The weight of the saturated water vapor is then shown *per cubic foot* of this *mixture*. Such a cubic foot might be 98.5 per cent dry air and 1.5 per cent water vapor. Columns (4) and (5) show us the weight of the water vapor per cubic foot of the moist air as we actually find it.

Columns (6) and (7) give the weight of the water vapor on an entirely different basis. From the heading of columns (6) and (7) you see that the weight of vapor is given *per pound* of *dry air*. Look at the values opposite 70°F; you see there 0.01574 lb of water vapor per pound of dry air or 110.2 grains of water vapor per pound of dry air. This means that we would have the following table:

Dry air...........	1.0000 lb
Water vapor......	0.0157 lb
Mixture..........	1.0157 lb

In other words, we imagine the water vapor and dry air as two separate distinct gases, even though their molecules are actually mixed and intermingling. We then imagine, as shown in the tabulation above, a sample of moist air just the proper size so that it contains 1.0000 lb of dry air plus water vapor to saturate it at this temperature. Stated in still another way, columns (6) and (7) show us, for any temperature, what *weight of water vapor* can be mixed with 1 *lb of pure dry air* in order to give us a 100 per cent saturated mixture.

Taking the values from our table at a temperature of 70°F, we see that 1 lb of dry air will hold 110.2 grains of water vapor, under which conditions it is 100 per cent saturated. We can tabulate this information from both columns (6) and (7):

	Pounds	Grains
Dry air	1.0000	7,000
Water vapor	0.0157	110.2
Mixture	1.0157	7,110.2

Suppose that we then want to consider 1.0000 lb of *mixture* and want to know the percentage of dry air and the percentage of water vapor in this pound of mixture. We just multiply 1 lb by the fraction representing the portion of each substance contained in the pound of mixture.

$$\frac{7,000}{7,110}(1.0) = 0.9845 \text{ lb dry air}$$

$$\frac{110}{7,110}(1.0) = 0.0155 \text{ lb water vapor}$$

Further Psychrometric Definitions—"Humidity." The information given in columns (4), (5), (6), or (7) (any one of these columns) is called *absolute humidity*. In other words, absolute humidity is the weight of saturated vapor expressed either *per cubic foot of moist air* or *per pound of dry air*. We gave you an idea of the meaning of relative humidity earlier in this chapter. Let us first tabulate some information from the Psychrometric Tables before defining relative humidity.

Saturation Temperature, °F	Weight of Moisture per Pound of Dry Air, Grains
40	36.36
50	53.38
60	77.21
70	110.2
80	155.5

The best way to understand relative humidity is to take an example. Suppose you have air at 70°F that you know contains 55.1 grains of moisture per pound of dry air. This air can hold 110.2 grains of moisture per pound of dry air if it is fully saturated. For convenience, we shall now abbreviate "grains of moisture per pound of dry air" as "grains per lb." Relative humidity may then be expressed as the following fraction:

$$\% \text{ R.H.} = \frac{55.1}{110.2}(100) = 50\%$$

Expressed in general terms, we can then define *relative humidity* as the ratio of the *actual absolute humidity divided by the absolute humidity of saturated air at the temperature being considered.* This we can express as a simple fraction as shown below:

$$\% \text{ R.H.} = \frac{\text{Actual grains per pound}}{\begin{array}{c}\text{max. grains per pound that could}\\ \text{be held at the given temperature}\end{array}}(100)$$

In other words, we may say, "If air holds a certain percentage of the maximum moisture it could hold at its particular temperature, that percentage is the relative humidity."

It is necessary that you understand the idea of relative humidity before going on to some of the other psychrometric properties. For this reason, we are presenting below two sample problems involving relative humidity. The first of these problems requires you to find the relative humidity of a certain sample of air. The second problem, which is more involved, is a practical air-conditioning problem in which you determine the relative humidity that will exist in an air-conditioned room.

Example. Given air at 80°F and a water vapor pressure of 0.791 mm of Hg, what is the relative humidity?

Solution. From the Psychrometric Tables we find that saturated air at 72° has a vapor pressure of 0.79058 mm Hg, which taken to three significant figures is 0.791 mm Hg. This air contains 118.2 grains of vapor per pound of dry air; *saturated* air at 80₀ would contain 155.5 grains per lb. Hence we have the equation

$$\% \text{ R.H.} = \frac{118.2}{155.5}(100) = 76\%$$

Example. Air is at 50°F, 100 per cent saturated (such as might leave a summer conditioning unit). This air is then *heated* to 80°F and has 25.9 grains of moisture per pound of air *added* (as might occur in an air-conditioned room); what is the final relative humidity?

Solution

Absolute Humidity	Grains per Pound
Initial........................	53.4
Added........................	25.9
Resultant....................	79.3

Since the room air is at 80°F, it could hold 155.5 grains per lb, *if fully saturated.* Hence the relative humidity of the air is

$$\% \text{ R.H.} = \frac{79.3}{155.5}(100) = 51\%$$

Let us turn now to columns (8) and (9) of Table V. These columns show the property of gases with which we are already familiar, *viz.*, the specific volume. The headings of columns (8) and (9) are reproduced below:

Volume, cu ft barometer, 29.92 in. Hg	
Of 1 lb of dry air (8)	Of 1 lb of dry air + vapor to saturate it (9)

You see, therefore, that both columns are based upon a temperature pressure of 14.7 psia, or an absolute pressure of 29.92 in. Hg. Column (8) shows a volume of 1 lb of bone-dry air. As you know, it would be impossible to get this air exactly bone dry, but since we can get air containing less than 1 grain of moisture per pound of dry air, it is useful for us to have the specific volume of dry air alone. Furthermore, we use the value in column (8) in calculating the specific volume of some partly saturated air. This calculation is explained on page 116.

Column (9) shows the specific volume of 1 lb of dry air mixed with sufficient additional water in a gaseous state so that we have a saturated mixture. Since the total pressure is still 14.7 lb and since we have added some additional gas to our original dry air, the volume of the resulting mixture is greater than the volume of the 1 lb of dry air alone. Look at the values for the specific volume at 70°F in both columns (8) and (9).

Specific Volume	Cubic Feet per Pound
1 lb dry air at 70°	13.34
1 lb dry air at 70°, plus vapor to saturate it	13.68
Difference	0.34

If you write down these values as shown above, you will see that the difference between them is 0.34 cu ft per lb. This difference represents the added volume caused by the addition of sufficient water vapor completely to saturate 1 lb of dry air at 70°F.

Suppose that you had 1 lb of dry air at 70°F containing 55.1 grains of moisture. We have already calculated relative humidities, and you can readily calculate that this air would have a relative humidity of 50 per cent. In other words, this air would contain one-half as much water vapor as is required completely to *saturate* the air. To get the specific volume of 1 lb of dry air plus water vapor such that the relative humidity is 50 per cent, we take the specific volume of the pound of dry air alone, as shown in the preceding paragraph (13.34 cu ft per lb). In the preceding paragraph we saw that moisture sufficient to saturate this air *fully* would increase the specific volume by 0.34 cu ft. For air at 50 per cent relative humidity, we simply add one-half of this increase with the results shown below:

	Cubic Feet per Pound
Specific volume of dry air at 70°F	13.34
Volume added by water vapor enough to cause 50% R.H.	0.17
Specific volume of 1 lb of dry air at 70°F, plus its vapor, such that the R.H. is 50%	13.51

Hence you may set up a general procedure for calculating the specific volume of air that is partly saturated with moisture. This procedure is as follows:

1. For any temperature air which you have, the specific volume of dry air and the specific volume of fully saturated air are given in columns (8) and (9) Table V.

2. Take the difference in these two values, which represents the added volume caused by moisture sufficient to saturate the air.

3. Calculate the relative humidity, or determine it in some other way.

4. Multiply the difference obtained in (2) by the relative humidity of the air expressed as a decimal. This gives the added volume caused by the moisture actually present in your partly saturated sample of air.

5. Add the result obtained in (4) to the specific volume of dry air, and the result is the specific volume of the partly saturated air.

Look now at columns (10), (11), and (12). The headings of these three columns are as follows:

Enthalpy per Pound

Dry air 0°F datum (10)	Vapor 32°F datum (11)	Dry air with vapor to saturate it (12)

For the present we shall not consider column (11). If you check with Table I of the Steam Tables you will see that column (11) merely reproduces the h_g column from the Steam Tables; *i.e.*, it shows the total heat of dry saturated steam at each temperature. The slight disagreement between the Steam Tables and column (11) in the Psychrometric Tables is of the same nature as the disagreement in the values shown for partial pressure in columns (2) and (3). Do not let this slight disagreement concern you at all.

Column (10) shows the total heat content in 1 lb of dry air above 0°F, as indicated by its title. You may wonder why, since we measure total heat above 32°F as our artificial zero in the case of the steam tables, we now proceed to measure the total heat above 0°F in the Psychrometric Tables.

It is simply a matter of convenience. You know that the true zero for total heat is at a temperature of 0°F *absolute,* or −460°F. We select the artificial zero point of 32°F in the case of the steam tables simply because this temperature happens to be the melting point of ice and we never have liquid water below this temperature. Hence this is made a convenient starting place at which we can call total heat 0 in the case of the steam tables. Air, however, does not go through any change of state at any ordinary temperatures, and air as cold as 0°F is frequently encountered in air-conditioning problems. Therefore, in psychrometric tables we select 0°F as a convenient artificial zero point above which to measure total heat.

If you take 1 lb of dry air at 0°F having a specific heat of *approximately* 0.24, how much heat must we add to raise the temperature of this dry air to 70°F? Using our fundamental heat equation from Chap. V, we have

$$\begin{aligned} Q &= cW(\Delta T) \\ Q &= 0.24 \times 1 \times 70 \\ &= 16.80 \text{ Btu per lb} \end{aligned}$$

If you look at the total heat of dry air at 70°F as shown in column (10) in the Psychrometric Tables, you will find that it is shown as 16.79 Btu per lb. Hence the values in column (10) simply show the sensible heat necessary to heat 1 lb of dry air from 0°F to each particular temperature shown in the tables.

However, as we have seen, we never have 1 lb of dry air without having the dry air mixed with a certain amount of water vapor. Taking saturated air at 70° again as an example, 1 lb of dry air contains 0.0157 lb of water vapor. For dry saturated water vapor at 70°F, the Steam Tables show a total heat of the vapor (h_g) of 1092.3 Btu per lb. Column (11) of the Psychrometric Tables (which we have neglected so far) shows the total heat of water vapor as 1090.7 Btu per lb. Let us use the latter value and get the *latent* heat of the 0.0157 lb of water vapor that is present in 1 lb of fully saturated air at 70°F.

$$0.0157 \times 1090.7 = 17.12 \text{ Btu}$$

Since this water vapor is mixed with our pound of dry air, the total heat of the mixture must be the sum of the *heat content of the dry air* plus the *heat content of the water vapor* in the mixture. If you add the heat content of the water vapor as we have just calculated it to the heat content of the dry air at 70° as shown in the table, you get

$$16.80 + 17.12 = 33.92$$

According to this calculation, the total heat of 1 lb of dry air at 70°F. plus water vapor to saturate this air fully, is 33.92 Btu. If you check in column (12) of the Psychrometric Tables, you will see that the strictly correct value is 33.96 Btu. (This apparent difference of 0.04 is probably because we dropped the last figure from the pounds of water vapor per pound of dry air. That is, we changed 0.01574 to 0.0157, dropping the last figure. Also our calculations are made in the engineering manner with a slide rule, whereas the tables were probably calculated on calculating machines.)

Column (12) in the Psychrometric Tables used in conjunction with column (10) becomes our most important single phase of psychrometric data. Suppose we have a sample of air at 70°F and a relative humidity of 50 per cent, what is the total heat content of this partly saturated air? We would take the difference between the total heat of the dry air with vapor to *fully* saturate it, as shown in column (12) of the Psychrometric Tables. From this we would subtract the total heat content in 1 lb of dry air as shown in column (10) in the table. This difference would give us the heat content of a fully saturated mixture which is due to the water vapor, as shown below:

Total heat of 1 lb of dry air plus vapor to fully saturate it, at 70°F	33.96
Total heat of 1 lb of dry air at 70°F	16.79
Difference (latent heat)	17.17

Since the pound of air at 70°F and at a relative humidity of 50 per cent contains only one-half of the moisture that would be required to fully saturate it, such a sample of air could contain only one-half of the latent heat value we have just calculated. The total heat of our partly saturated air is then found as follows:

$$16.79 + \tfrac{1}{2}(17.17) = 25.37 \text{ Btu per lb}$$

Thus far we have not mentioned one of the most important psychrometric properties. This property is known as the "wet-bulb" temperature of air. Ordinary temperature, as we know it, measured *in the shade* by means of an ordinary thermometer whose bulb is not wet, is called the *dry-bulb temperature of air.* We would read "70° D.B.," meaning that the dry-bulb temperature of the air was 70°. This is read simply "70° dry bulb." The wet-bulb temperature of air is measured by means of a thermometer with its bulb enclosed in a cloth mesh bag; the bag must be moistened with clean water, and there must be a wind or air motion past the thermometer bulb until the temperature comes to a steady value. This air motion by the bulb may be achieved in one of three ways. (1) The thermometer may be held in front of an air-discharge grille or inserted into an air duct through which air is moving. (2) The thermometer may be whirled about on the end of a stick or chain. (3) In a special type of instrument, air may be drawn in by suction over the bulb of the wet-bulb thermometer.

The water evaporated from the wet bulb of a thermometer is the same as the water evaporated from wet sidewalks as they dry off or from clothes drying on a line. This evaporation takes place because the air, into which the moisture is evaporating, is not completely saturated with water vapor. In other words, the water vapor originally in the air is superheated. As we have learned, in this case the vapor pressure of the water vapor already in the air is lower than the vapor pressure corresponding to the temperature of the air. Suppose we return to the example of air at 70°F containing 55.1 grains per lb.

This air is, as we have seen, at a relative humidity of 50 per cent. The vapor pressure of the water vapor of this air is approximately 0.376 in. Hg, which corresponds to a saturation temperature of approximately 51°F; thus 51° is the *dew point* of this air. This is calculated as follows: Run your finger along column (7) in the Psychrometric Tables, which gives the grains of water vapor per pound of dry air, until you come to the value nearest 55.1 grains per pound. Without bothering to interpolate, take 55.45 in the table as the nearest value and read to the left to get the vapor pressure and the saturation temperature. Remember that the absolute humidity determines the vapor pressure of the water vapor and the saturation temperature corresponding thereto.

Water on a sidewalk, in clothes on a line, or around a wet-bulb thermometer that has been standing in air, if the air were *absolutely still*, would be at the same temperature as the air. In our example, we are considering a wet-bulb thermometer past which air is moving rapidly, the properties of the air being as follows:

Dry-bulb temperature	70°F
Wet-bulb temperature	?
Absolute humidity	55.1 grains per lb
Relative humidity	50%
Vapor pressure	0.376 in. Hg
Dew point (saturation temperature)	51°F

If we have dipped the cloth wick of our wet bulb into a bottle of 70° water and then have exposed the bulb to moving air having the properties described above, what will be the vapor pressure of the water in the cloth wick of our wet bulb? The answer to this is easy; since the water is at 70° we look opposite 70° in the Psychrometric Tables and read the vapor pressure to three significant figures as 0.739 in. Hg. The vapor pressure in our air is only 0.376 in. Hg, or approximately half of the vapor pressure of the water in the wick of our wet bulb. Therefore, some of the water will change its state and cease to be liquid water on the wet bulb; molecules will fly off in a gaseous state and mingle with the water vapor originally in the air. Since the air is moving rapidly past the wet bulb, the water that evaporates from the bulb will be carried away and new air at the original vapor pressure of

0.376 in. Hg will always be in contact with the cotton wick of our wet bulb.

But you may say that it requires heat to cause water to change its state, and where do we get this necessary latent heat of vaporization? It is correct to raise this question here because the evaporation of water from a wet bulb *does* require latent heat. The source of this heat is what makes the wet bulb the valuable instrument it is.

Suppose you cool some water from 70 to 60°F—to do this you must remove Btu from the water. In other words, if you take some water at 70°F and remove heat from it, the temperature of the water will drop. In the case of our wet bulb, evaporation starts because of the vapor-pressure difference between liquid water in our wet-bulb wick and the vapor pressure in the air blowing past the wick. Just after exposing a freshly dipped wet bulb to moving air, the Btu required to vaporize some water from the wick are taken *right out of the water on the wick itself.* This causes the temperature of our cloth wick to drop, since the cloth is soaked with water and the temperature of the water is dropping. The wet-bulb thermometer shows this temperature drop vividly. Probably you have observed this phenomenon in the laboratory—if you have not, make a point of doing so at your earliest opportunity. You can easily make a wet bulb by getting a shoe lace, which is like a hollow cloth tape, and slipping it over the bulb of an ordinary thermometer at home.

How far will the temperature of a wet-bulb thermometer drop? What makes it stop dropping? Is it true that water continues to evaporate from the wick as long as the wick is wet, even after the water in the wick is as cold as it is going to become? Under these conditions, where does the heat come from to provide the required latent heat of vaporization? The correct answers to these questions complete the discussion of the theory of a wet-bulb thermometer.

In the example we are considering (see the table on page 119 if you have forgotten it), let us assume for the present that the temperature of our wet-bulb wick drops to 58.5°F and then does not drop any further. If the bulb is still wet, the wet-bulb thermometer will continue to read 58.5°F, even though air at a dry-bulb temperature of 70° is blowing past, and water will continue to evaporate from the wick. We want to find out where the latent heat is coming from to provide this continued evaporation of water.

Think back to Chap. V regarding the fundamental laws of heat and also regarding heat transfer by conduction. There we stated that if two bodies at different temperatures were placed in contact it was a fundamental thermodynamic law that heat would flow from the hotter to the colder body by conduction. Our moving stream of air is touching our wet bulb; the air is at 70°, and the wet bulb is at 58.5°. Therefore, heat flows continuously from the hotter body to the colder body; *i.e.*, heat flows from air to the colder wet-bulb wick. This heat, which flows from the warmer air to the cooler wet-bulb wick, is sensible heat. As soon as this sensible heat gets into the water of the wet-bulb wick, it is used as latent heat to evaporate some water. (The heat of a gas flame under a kettle of boiling water is sensible heat, but as soon as it gets into the water it is used as latent heat.) Thus you see the source of the heat that is used as latent heat of vaporization to continue the evaporation of water from the wet-bulb wick.

We have raised the question of how far a wet-bulb temperature drops and what makes it stop dropping. We can now answer this question. The farther the wet-bulb temperature drops below the dry-bulb temperature, the faster heat can flow from the air to the wick. When a wet-bulb thermometer has dropped only 1 or 2°, very little heat flows from the air to the wick, because the temperature difference is small. If the rate of evaporation of water from the wick is so rapid that the heat flowing to the wick from the air is not sufficient to provide the required latent heat of vaporization, some of this latent heat will be taken out of the water itself and the temperature will drop further. After the temperature of the wet-bulb wick has dropped lower, heat will flow to the wick from the air at a faster rate. If this heat is still insufficient to provide the required latent heat, the temperature of the wet bulb will drop still further. *In short, the temperature of the wet-bulb wick will continue to drop until it becomes sufficiently cooler than the air so that the flow of sensible heat from the air to the bulb is just sufficient to provide the required latent heat that is being used to evaporate water from the bulb.*

This is one of the many examples in the science of equilibrium, or balance. In this case it is a heat balance, or thermodynamic equilibrium. Sensible heat (the h_f) in the water on the wick is used to evaporate some of this water. So long as this sensible heat is being taken out of this water faster than heat flows into the water from the air, the temperature of the wick continues to drop. When it has dropped low enough so that the latent-heat requirements are exactly supplied by the heat flowing into the wick from the warmer air, the temperature indicated by the wet bulb ceases to drop, or comes to rest.

The lower the relative humidity of the air flowing from the bulb, the faster water will evaporate. Hence the *lower the relative humidity* of the air, the *greater the need for latent heat* to evaporate water from the wet bulb. For air at a certain dry-bulb temperature, the lower the relative humidity of the air, the lower the latent heat of the air.

Read the last paragraph over and be sure you understand it thoroughly. From this you can see that the lower the latent heat of air (at a given dry-bulb temperature), the more latent heat is required to evaporate water from a wet bulb in this air. As we have seen, the *more latent heat required* to evaporate water from the wet bulb, the *lower the temperature of the wet bulb must be in order to obtain this heat as sensible heat* by heat flow due to conduction.

Let us now summarize our discussion regarding wet-bulb temperature and its significance. The wet bulb requires latent heat to evaporate moisture. It requires more of this latent heat if the latent heat of the air is low. Sensible heat from the air flows to the wet-bulb wick because of the temperature difference, thus providing this latent heat which is needed. Thus the wet-bulb thermometer measures a peculiar new kind of temperature entirely different from ordinary dry-bulb temperature. The wet-bulb thermometer really measures a new temperature that reflects both the *sensible heat* and the *latent heat* of the air being measured.

If air is at 100 per cent relative humidity, no water at all will evaporate from the wick of a wet bulb because the air is already 100 per cent saturated with moisture. Therefore, the wet-bulb temperature will be exactly the same as the dry-bulb temperature since no evaporation takes place to use up heat. The lower the relative humidity of air at a given dry-bulb temperature, the lower the wet-bulb temperature will be, as we have seen from our discussion. We hope that it is now clear to you that the reading we get by means of a wet-bulb thermometer depends upon both the dry-bulb temperature of the air and the absolute humidity of the air. You know that the sensible heat of air is determined only by its dry-bulb temperature. You also know that the latent heat of air is determined only by its absolute humidity. Therefore, the wet-bulb temperature of air is affected by both the sensible heat and the latent heat of the air.

We must now ask you to accept a statement without mathematical proof. It is, however, a statement that you can understand as a result of our foregoing discussion. Since this statement is one of the most important basic principles of psychrometry, we shall state it in two ways.

For any mixtures of air having the same total heat, regardless of the proportions of this total heat due to sensible heat of the dry air and latent heat of the water vapor, the wet-bulb temperatures are the same.

Different samples of air having the same wet-bulb temperature have the same *total heat,* even though the sensible heat and latent heat components may be widely different for the different samples.

The second statement could be rewritten, using slightly different words as follows:

Different samples of air having the same wet-bulb temperature have the same total heat regardless of their dry-bulb temperatures and relative humidities.

The psychrometric principle that we have just given you is now known to be slightly inexact. However, the inaccuracy is only a small fraction of 1 per cent. Until a few years ago, however, air-conditioning engineers believed this relationship to be absolutely exact according to a mathematical proof given by Willis H. Carrier.[1] A few years ago Carrier and several others discovered this minute inaccuracy and published papers explaining it. According to the 1939 A.S.H.V.E. "Heating, Ventilating, and Air Conditioning Guide," this error is so slight that it is neglected in ordinary psychrometric tables and psychrometric charts.

Table 1 shows various properties of different samples of air at a different dry-bulb temperature and, hence, at different sensible heats. Although the tables do not show it, the dew points (dew points fix absolute humidity—the absolute humidity fixes latent heat) are different and, hence, the latent heats are different. Note that the total heat is, however, the same in each case.

We can have the following different samples of air all at a wet-bulb temperature of 58.5°F.

TABLE 1

Dry bulb	R.H. %	Wet bulb	Sensible heat, Btu per lb	Latent heat, Btu per lb	Total heat, Btu per lb
58.5	100	58.5	14.03	11.35	25.38
60.5	90	58.5	14.51	10.87	25.38
62.5	80	58.5	14.99	10.39	25.38
64.5	70	58.5	15.47	9.91	25.38
67	60	58.5	16.07	9.31	25.38
70	50	58.5	16.79	8.59	25.38
73.5	40	58.5	17.63	7.75	25.38
77.5	30	58.5	18.59	6.79	25.38
83	20	58.5	19.91	5.47	25.38
92	10	58.5	21.83	3.55	25.38

The values given in Table 1 will not check exactly with the Psychrometric Tables because it was made up from a psychrometric chart. We want to call your attention to the fact that the temperatures in column (1) of the Psychrometric Tables may be really used as any one of three different temperatures. Remember that the Psychrometric Tables are based upon 100 per cent saturated air. For fully saturated air, the *dry-bulb temperature*, the *wet-bulb temperature*, and the *dew point temperature* are the same. You already know that if you have some air at a dew point temperature of 50°, you will find its absolute humidity opposite 50° in the Psychrometric Tables, *regardless of the dry-bulb temperature of your air*. Similarly, if you have some air at a wet-bulb temperature of 70°, it makes no difference

[1] CARRIER, W. H., Rational Psychrometric Formulae, *A.S.M.E. Trans.*, vol. 33, 1005, 1911.

whether the dry-bulb temperature is 70, 80, or 90°, you still find the total heat of this air opposite the temperature of 70° in the Psychrometric Tables. The table gives the total heat for air that is fully saturated at 70°, but such air would have a wet-bulb temperature of 70°. As we have seen, its total heat is the same as any other having a wet-bulb temperature of 70°.

Because of the ease of measuring the dry-bulb temperature of air, this is one of the properties of air that we can almost always measure and be sure of. In most cases, we can obtain the wet-bulb temperature of the air in addition to the dry-bulb temperature; if this is not possible we can usually read the relative humidity of the air directly from an instrument calibrated in per cent relative humidity. In practical problems, therefore, you can ordinarily count on knowing one or two pairs of data regarding the air. You would know either the dry-bulb and wet-bulb temperatures, or the dry-bulb temperature and relative humidity. Shortly we shall give you several problems in which you will calculate seven other properties of air from two known properties. Before doing this however, we wish to summarize our knowledge of psychrometry gained thus far.

In the Psychrometric Tables we have studied the following properties of air:

Dry-bulb temperature
Wet-bulb temperature
Relative humidity
Dew point
Grains per pound (absolute humidity)
Total heat
Sensible heat
Latent heat
Specific volume

Although the Psychrometric Tables present data for fully saturated air, we can read many properties of partly saturated air directly from the tables with no additional calculations. The following paragraph gives the data regarding partly saturated air, which can be read directly from the tables. Do not memorize this material, but study it so that you understand it and can reproduce it with an intelligent explanation.

If you know any one of the following three properties of air, you can read the other two directly from the table. This holds true regardless of whether the air is partly or fully saturated.

Dry-bulb temperature
Sensible heat
The specific volume of the dry air only

If you know either of the following two properties of air, you can read the other one directly from the table even if you know nothing else at all about the air.

Wet-bulb temperature
Total heat of the air

If you know any one of the following three properties, you can read the other ones directly from the table.

Dew-point temperature
Vapor pressure of the water vapor
Absolute humidity (absolute humidity may be expressed as the weight of water per pound of dry air, or the weight of water per cubic foot of mixture. The weight of the moisture may be expressed either as grains or pounds)

Knowing either the dew point or the absolute humidity, you can also, by a simple calculation, obtain the latent heat of the moisture in the air. This calculation is shown in Eq. (2).

We want to give you four basic equations that you will use frequently in making calculations regarding partly-saturated air, based upon data from our tables.

$$\text{Sensible heat} + \text{latent heat} = \text{total heat} \tag{1}$$

$$\text{Latent heat} = \text{total heat} - \text{sensible heat} \tag{2}$$

These equations are the same equation written in different forms. Relative humidity has been defined earlier in this chapter. The equation below shows two fractions that express relative humidity as a decimal, both of which are frequently used.

$$\frac{\%\ \text{R.H.}}{100} = \frac{\text{actual grains per pound}}{\text{grains per pound if saturated}} = \frac{\text{actual latent heat}}{\text{latent heat if saturated}} \tag{3}$$

Whenever you wish to find the specific volume of partly-saturated air, you must take the relative humidity (or either of the two fractions that are equivalent to it) and use this value as indicated in Eq. (4) below:

$$\text{Specific volume} = \text{specific volume dry air} + \frac{\%\ \text{R.H.}}{100} \tag{4}$$

In order to determine how well you understand psychrometry and the use of the Psychrometric Tables, let us now undertake the solution of two practical problems.

***Example* 1. Suppose that you have air for which you have measured dry-bulb temperature and wet-bulb temperature, these temperatures being 95 and 75°, respectively; find all the other properties of this air.**

Solution. In order to know what we have to find, let us rewrite the list of properties given on page 124, showing values for the two properties we already know.

Dry-bulb temperature	95°
Wet-bulb temperature	75°
Relative humidity	?
Dew point	?
Grains per pound (absolute humidity)	?
Total heat	?
Sensible heat	?
Latent heat	?
Specific volume	?

From Eq. (3), we have

$$\% \text{ R.H.} = \frac{\text{actual latent heat}}{\text{latent heat if saturated}} (100) \tag{5}$$

Therefore, the per cent relative humidity of our air is as follows:

$$\% \text{ R.H.} = \frac{15.66}{40.25} (100)$$

$$= 38.9\%$$

If our air were fully saturated at 95°, column (7) of the Psychrometric Tables tells us that the absolute humidity would be 255.6 grains per lb. Our air contains only 38.9 per cent of this amount of water vapor so we have

$$0.389 \ (255.6) = 99.2 \text{ grains of moisture per pound of dry air}$$

Look in column (7) of the Psychrometric Tables until you find a value close to 99.2. You may discover that 99.2 is between values given in the table, in which event we will have to interpolate. Corresponding to a saturation temperature or dew point of 67° you find an absolute humidity of 99.19 grains per lb, which taken to three significant figures is 99.2. Therefore, the dew point of our air is 67°.

We now have to find only the specific-volume increase caused by the water vapor. For air at 95°, take the difference between the specific volumes of bone-dry air and of fully saturated air, as shown in columns (8) and (9) of the table. Thus,

$$14.79 - 13.97 = 0.82$$

This difference represents the volume increase that would be caused by sufficient water vapor to saturate the air fully at a temperature of 95°. Our air contains only 38.9 per cent as much moisture as would be required to saturate it. Therefore, the volume increase due to the water vapor actually in our air is as follows:

$$0.389(0.82) = 0.32^*$$

Then

$$13.97 + 0.32 = 14.29 \text{ cu ft per lb}$$

This is the specific volume of 1 lb of dry air plus the actual amount of water vapor that our partly saturated air is carrying.

Our problem is now fully solved, and the answers may be tabulated as follows:

Dry-bulb temperature	95°
Wet-bulb temperature	75°
Relative humidity	38.9%
Dew point	67°
Grains per pound (absolute humidity)	99.2 grains of moisture per pound of dry air
Total heat	38.46 Btu per lb
Sensible heat	22.80 Btu per lb
Latent heat	15.66 Btu per lb
Specific volume	14.29 cu ft per lb

Example 2. Suppose that we have 10,000 cu ft of air that we know is at 80° dry bulb and 50 per cent relative humidity. We want to find all the properties of this air, including the vapor pressure of the water vapor in inches of mercury. In addition, we wish to find how many pounds of air and how many pounds of vapor we have.

Solution. Opposite 80° in the Psychrometric Tables we read the following information regarding *fully saturated* air:

Vapor pressure	1.032 in. Hg
Absolute humidity	155.5 grains per lb
Specific volume of dry air	13.59 cu ft
Specific volume of 1 lb of dry air plus vapor to fully saturate it	14.08 cu ft
Sensible heat	19.19 Btu per lb
Total heat	43.51 Btu per lb

We are now through reading from the Psychrometric Tables for a while, as we have several calculations to make using the above values. Keeping in mind that our relative humidity is 50 per cent, we can write the following equations:

$$43.51 - 19.19 = 24.32 \text{ Btu per lb latent heat of fully saturated air at } 80°$$
$$0.5(24.32) = 12.16 \text{ Btu per lb actual latent heat of our air}$$

* We realize that this value is not exactly 0.3200; instead it is nearer 0.3185. However this is to be added to 13.97, which has only two significant figures to the right of the decimal point. Figures beyond two figures to the right of the decimal point would be meaningless in such an addition, so we "round off" the number that we are going to add to simply 0.32.

$$0.5(155.5) = 77.8 \text{ grains per lb actual absolute humidity of our air}$$
$$0.5(14.08 - 13.59) = 0.25 \text{ cu ft (specific volume increase due to water vapor in our air)}$$
$$14.08 + 0.25 = 14.33 \text{ cu ft per lb specific volume of our partly saturated air}$$
$$19.19 + 12.16 = 31.35 \text{ Btu per lb actual total heat of our air}$$

Let us now tabulate what we have learned before turning back to the Psychrometric Tables to complete our solution.

Dry bulb	80°
Wet bulb	?
Relative humidity	50%
Dew point	?
Grains per pound	77.8
Total heat	31.35 Btu per lb
Sensible heat	19.19 Btu per lb
Latent heat	12.16 Btu per lb
Specific volume	14.33 cu ft per lb
Vapor pressure	?

From this list you see that we now have three unknown quantities. To find the wet bulb, look in the total heat column [column (12)] for a value close to 31.35 Btu per lb. In the table you will see that the values 30.73 Btu per lb and 31.51 Btu per lb are the nearest we can get, and these correspond to 66 and 67°, respectively. Interpolating, we see that our total heat value is $\frac{62}{78}$ of the way in between the two values shown in the table. Since $\frac{62}{78}$ is approximately 0.8, our air has a wet-bulb temperature of 66.8°.

To find the dew point and the vapor pressure of our air we repeat this process, except that we now look in column (7) of the table and try to find the value of 77.8 grains per lb. The two nearest values are 77.21 grains per lb at 60° and 80.08 grains per lb at 61°. Interpolating, our dew point is found to be 60.2°. Similarly, vapor pressure is 0.525 in. Hg. This completes the list of properties that we wish to find. We must now find how many pounds of dry air and how many pounds of water vapor are in our mixture of 10,000 cu ft of moist air.

We have found the specific volume of our air to be 14.33 cu ft per lb. Therefore,

$$\frac{10{,}000}{14.33} = 697 \text{ lb}$$

This gives us the weight of the dry air plus the water vapor, or the weight of our mixture. To find how much of the mixture is dry air and how much is water vapor *by weight*, it is simply necessary to use the method that we have already learned on page 112.

$$\frac{7{,}000}{7{,}077.8}(697) = 689.3 \text{ lb of dry air}$$

$$\frac{77.8}{7{,}077.8} \qquad 7.7 \text{ lb of water vapor}$$

In the preceding problem, suppose that we had been asked to get the proportions *by volume* instead of by weight, *i.e.*, suppose we had wanted the number of cubic feet of dry air and the number of cubic feet of water vapor. In this case we would have had to multiply by different fractions, being careful to use fractions based upon the *volume* relationship, rather than on a weight relationship as was done at the end of the preceding paragraph. In the preceding problem we obtained just the volume relationships that we needed, either by calculations or reading from the table. We found that the specific volume of bone-dry air at 80° was 14.08 cu ft and that the specific volume of our actual air was 14.33 cu ft. Using these values, we can get the number of cubic feet of dry air and the number of cubic feet of water vapor contained in our 10,000 cu ft of moist air.

$$\frac{14.08}{14.33}(10{,}000) = 9{,}825 \text{ cu ft of dry air}$$

$$\frac{0.25}{14.33}(10{,}000) = 175 \text{ cu ft of water vapor}$$

If you thoroughly understand the solution of the two problems just given, you can handle any psychrometric problems of the types which are solved by using psychrometric tables. Try some of the problems at the end of this chapter to be sure that you can handle this type of work properly. We now wish to return to the subject of graphs insofar as graphs relate to psychrometry. As you probably know, complete graphs of the Psychrometric Tables are available, showing not only all the quantities given in our tables, but even some additional quantities. Our workplotting graphs so far has been limited to simpler graphs of simpler tables. Let us, therefore, take out of the Psychrometric Tables certain specific data that we shall tabulate and then graph.

At the start, suppose that we list dry-bulb temperatures from 0 to 100° and that we tabulate the grains of water vapor per pound of dry air required to produce saturation at these various temperatures. Let us also tabulate values for one-half of the absolute humidity at each temperature, thus getting the grains per pound contained in air at 50 per cent relative humidity at each temperature. The results would be as shown in Table 2.

Figure 1 shows a graph on which the horizontal axis shows dry-bulb temperatures according to the Table 2 and the vertical axis shows absolute humidity in grains per pound. Curve *A* on Fig. 1 gives the absolute humidity for 100 per cent saturated air. Curve *B* gives the absolute humidity for 50 per cent saturated air or air having a relative humidity of 50 per cent.

Figure 1 is an extremely simple psychrometric chart. Its usefulness is, of course, very limited in the form shown, because it shows nothing about vapor pressure, specific volume, heat content, nor does it show absolute humidity in grains per cubic foot. Its most serious deficiencies are that it does not show dew point, wet-bulb temperature, or total heat. Let us see if we cannot add to this graph, as shown in Fig. 1, some additional lines that will remove its deficiencies. While we are doing this, we shall also indicate a few lines for other relative humidities opposite 50 and 100 per cent.

TABLE 2

Dry-bulb temperature, °F	Absolute humidity, grains per lb	
	To saturate fully	To saturate 50%
0	5.5	2.75
10	9.2	4.6
20	15.0	7.5
30	24.0	12.
40	36.4	18.2
50	53.4	26.7
60	77.2	38.6
65	92.4	46.2
70	110.2	55.1
75	131.1	65.6
80	155.5	77.8
90	217.	108.5
100	300.5	150.25

In order to make these changes, we must tabulate some more information from the Psychrometric Tables. As we learned in Chap. II, a graph can be plotted only after we have some tabulated data. We can rewrite the temperature column from Table 2 and the absolute humidity column for 100 per cent saturation from this same table. So long as we use the absolute humidities for the 100 per cent saturated condition, these two columns may be rewritten with different headings as shown in columns (1) and (2) in Table 3. As we have seen, *for* 100 *per cent saturated air,* the wet-bulb temperature is identical with the dew-point temperature and the dry-bulb temperature. Therefore, we can add the wet-bulb temperature to Table 3, the actual temperatures being just the same as the dew-point temperatures. Corresponding to each wet-bulb temperature, we can show the total heat in Btu per pound.

Table 3 tells us that we can label the points where our dry-bulb temperature lines meet the 100 per cent relative-humidity lines as dew-point temperatures or wet-bulb temperatures, in addition dry-bulb temperatures, as shown in Fig. 1. Table 3 also tells us that there is a relationship between total heat and wet-bulb temperature. Furthermore, we know that the dew point is the same, provided the absolute humidity is the same. From Fig. 1, you can see that our graph is constructed with the absolute-humidity lines horizontal. Therefore, the dew-point lines are drawn horizontal from the 100 per cent saturation line. This would give us a graph (not illustrated) from which we could read dry-bulb temperature, dew-point temperatures, grains per pound, and per cent relative humidity. However, we do not know in which direction our wet-bulb lines must run. Suppose that we try to draw our line for about a 60° wet bulb. We know that this

TABLE 3

(1) Dew point	(2) Grains per lb	(3) Total heat	(4) Wet bulb
0	5.5	0.83	0
10	9.2	3.80	10
20	15.0	7.09	20
30	24.0	10.89	30
40	36.4	15.19	40
50	53.4	20.3	50
60	77.2	26.4	60
65	92.4	30.0	65
70	110.2	34.0	70
75	131.1	38.5	75
80	155.5	43.5	80
90	217.0	55.7	90
100	300.5	71.4	100

line starts at the 100 per cent saturation line where the 60° dew-point and 60° dry-bulb lines meet the saturation curve. To find the direction taken by this wet-bulb line, we obviously must locate *one* other point on the line, if

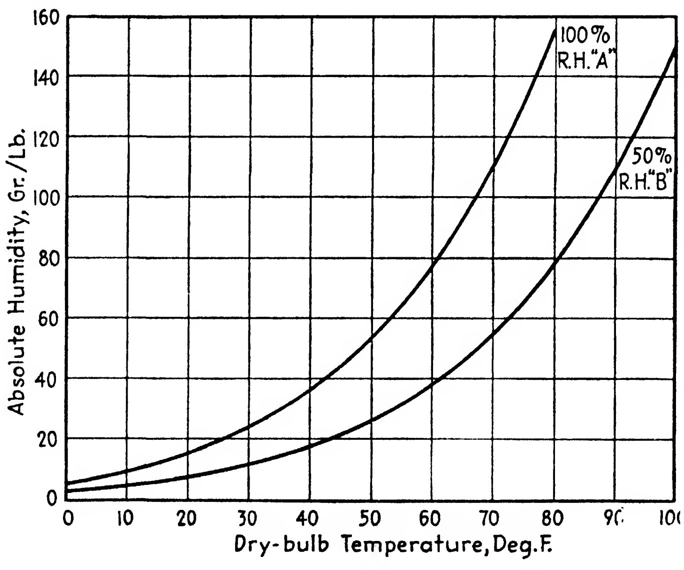


FIG. 1.—Basic psychrometric chart for 100 per cent and 50 per cent saturation.

it is to be a straight line. We would have to locate numerous other points if our wet-bulb line were to be a curve.

Fortunately, we can state as a fact that our wet-bulb line will be a straight line. Let us now locate the points where our wet-bulb lines cross the 50 per cent relative-humidity curve. This will enable us to draw this wet-bulb line. We must make a table such as Table 4, which shows wet-bulb temperatures from 0 to 100° and the corresponding dry-bulb temperature such that the relative humidity is 50 per cent. In Table 4, most data are given only to two or three significant figures.

The values shown in columns (7) and (8) in Table 4 are the values that we wish to graph. Taking, for example, the value where the dry-bulb temperature is 90° and the wet-bulb temperature is 75.2°, let us see the meaning of these two columns. Using these specific values, the table tells us that a line on our chart such as Fig. 1, representing a wet bulb of 75.2°, will start from the 100 per cent saturation curve at a temperature of 75.2° and that this line will intersect the 50 per cent saturation curve at a point where the dry-bulb temperature is 90°F. In this manner, wet-bulb lines are drawn on a chart identical to Fig. 1. All the wet-bulb lines developed by

TABLE 4

For 100% saturated air					For 50% saturated air		
Dry bulb (1)	Total heat (2)	Sensible heat (3)	Latent heat (4)	One-half Latent heat (5)	Column (3) + column (5) (6)	Wet bulb (7)	Dry bulb (8)
0	0.83	0	0.83	0.42	0.42	−1.5	0
10	3.8	2.4	1.4	0.7	3.1	7.7	10
20	7.1	4.8	2.3	1.2	6.0	17.0	20
30	10.9	7.2	3.7	1.9	9.1	25.5	30
40	15.2	9.6	5.6	2.8	12.4	33.6	40
50	20.3	12.0	8.3	4.2	16.2	42.1	50
60	26.4	14.4	12.0	6.0	20.4	50.3	60
70	34.0	16.8	17.2	8.6	25.4	58.5	70
80	43.5	19.2	24.3	12.2	31.4	67.1	80
90	55.7	21.6	34.1	17.1	38.7	75.2	90
100	71.4	24.0	47.4	23.7	47.7	83.8	100

columns (7) and (8) in Table 4, which will fit upon the chart, are drawn. This chart is shown in Fig. 2.

The wet-bulb lines that we have calculated in Table 4 and illustrated in Fig. 2 are unfortunately for *odd*-numbered wet-bulb temperatures and fractions of a degree in addition. In the case of an actual psychrometric chart, "even" wet-bulb temperature lines—30, 35, 40, 45, etc.—would be drawn. We have mentioned that a psychrometric chart should also show the relationship between total heat and wet-bulb temperature. Figure 3 shows the psychrometric chart complete as published by Peerless of America, Inc., for use by their employees and customers. Notice the sloping straight lines above and to the left of the 100 per cent saturation curve. These lines show the relationship between total heat and wet-bulb temperature. You may also note that on the Peerless chart there are lines that show the specific volume in cubic feet per pound of dry air.

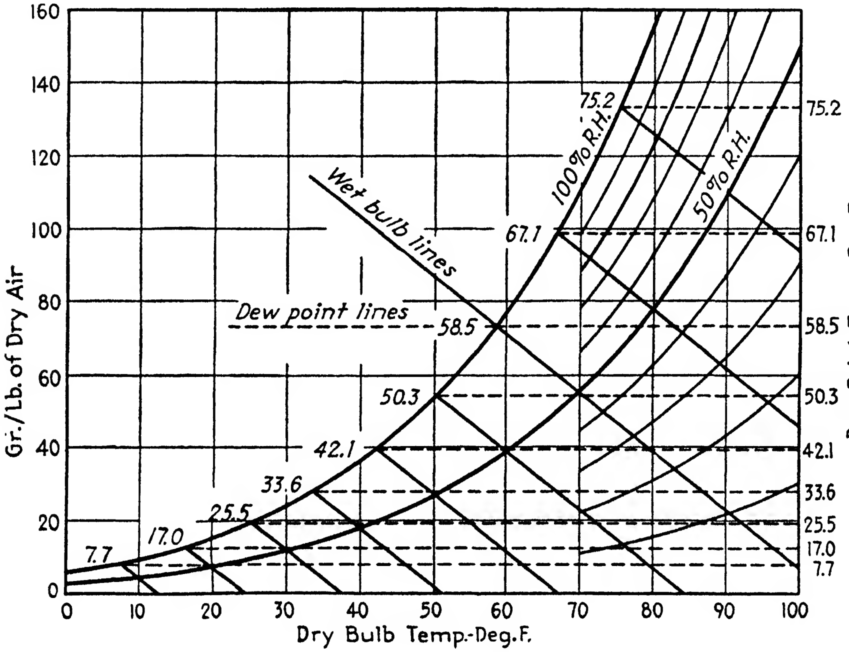


FIG. 2.—Development of psychrometric chart.

For the present we do not intend that you become fully familiar with the psychrometric chart. At this time in concluding our study of psychrometry and psychrometric tables, we simply want you to understand two things. (1) You should appreciate the enormous amount of work involved in compiling and drawing a psychrometric chart. (2) From Tables 3 and 4 and Figs. 1, 2, and 3, we want you to gain a fundamental understanding of *how* psychrometric charts are drawn from the original tables of psychrometric data. Remember that psychrometric charts are merely graphs that give a great deal of information. Later chapters will explain the use of different psychrometric charts in practical problems.

PROBLEMS

1. We have 10,000 cfm of outdoor air at 0°F, 55 per cent relative humidity, which is brought into an air-conditioning system, first passing over heating coils that heat the air to 110°F. Find all the properties of the air before it passes over the heating coils and

after it has passed over the heating coils. Also find the Btu per minute added to the air by the heating coils. (The heating coils add sensible heat only of the air.)

2. Suppose that the air from Prob. 1 after being heated is passed through a spray-type air washer; in the air washer the total heat of the air remains unchanged, but its absolute humidity is increased to 60 grains per lb. Find the dry bulb, the dew point, and

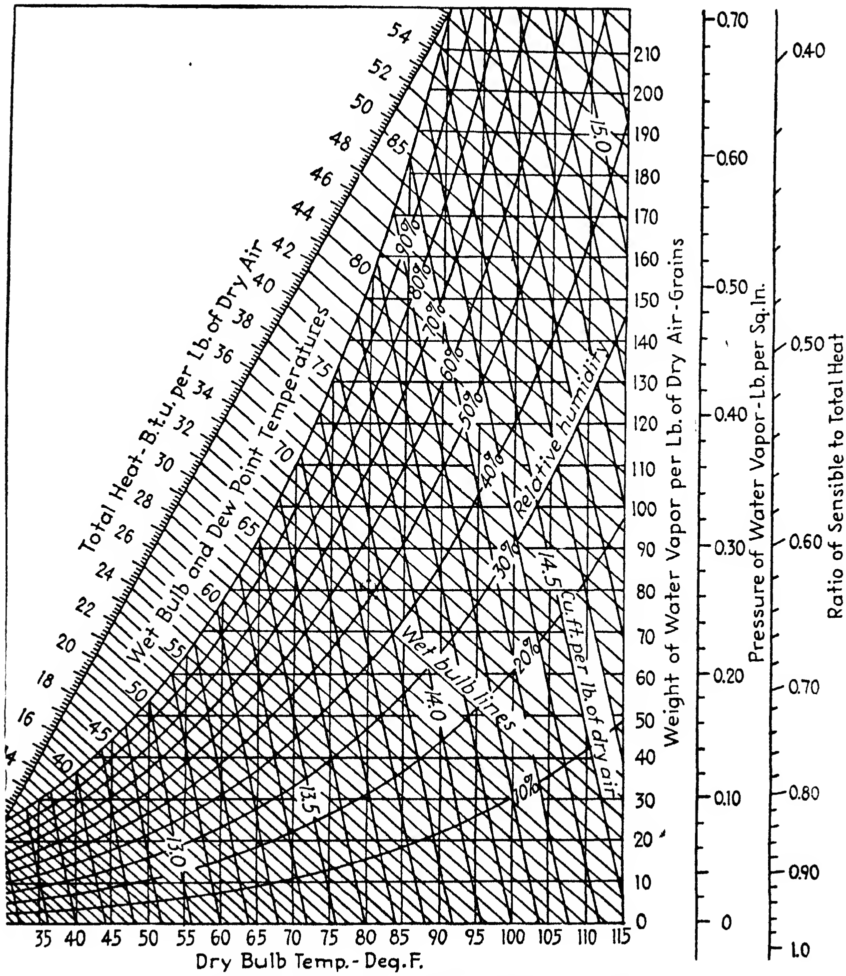


FIG. 3.—Psychrometric chart. (*Peerless Company of America.*)

the relative humidity after it leaves the air washer. Find the Btu per minute of heat (the latent heat) added by the air washer. Where does this latent heat come from, bearing in mind that the *total heat* does not change in passing through the air washer? Find also how many pounds of water per minute are added to the air from the air-washer sprays.

3. If we pass air at 95° dry bulb and 70° wet bulb over a cooling coil that cools the air to 60°F, what properties of the air remain constant (do not change) as the air passes over the cooling coil? How much moisture condenses out of the air? (You probably will have to find almost all the properties of the air before and after the cooling coil in order to be sure which ones change and which do not).

4. You want to cool 1,000 cfm of air from 95° dry bulb, 78° wet bulb to 55° dry bulb, 53° wet bulb; find the following: (*a*) The relative humidity of the air before and after cooling. (*b*) The Btu per minute you must remove. (The total heat you remove.) (*c*) The percentage of this total heat that is sensible and the percentage that is latent. (*d*) The number of pounds per minute of water that is removed as condensation.

5. Express the absolute pressure equivalents of each of the pressures listed in all the following units: inches of mercury, millimeters of mercury, pounds per square inch.

A vacuum of 28 in. Hg
One pound per square inch absolute
One pound per square inch gauge

(These data should be presented in a neatly tabulated form.)

6. Suppose that you have a tank (or room) containing 40,000 cu ft of air at 80° dry bulb, 67° wet bulb. What is the relative humidity of the air? What is the specific volume of the air?

7. How many pounds of dry air are contained in the tank described in Prob. 6? How many pounds of water vapor?

8. Explain briefly (about 100 words) your understanding of why the wet-bulb temperature of air indicates the *total* heat of the air.

9. With a pencil and straightedge extend the wet-bulb temperature lines in well to the left of the saturation curve. Using the data in Table 4, draw a straight line along the ends of your extended wet-bulb lines and plot the total heat values corresponding to the various wet-bulb temperatures. (See the Peerless psychrometric chart on page 134 for a sample, but do not expect your numerical values to agree exactly with those shown on the total heat scale of the Peerless chart.)

10. A winter ventilating system takes outdoor air at 0° dry bulb, 100 per cent relative humidity and delivers it to the ventilated rooms at 70° dry bulb, 40 per cent relative humidity. If the air quantity is 10,000 cfm, how many pounds per minute of dry air are being used? Find also the total heat added per minute to the air and the number of pounds of water added per minute to the air. Show a sketch of the conditioner with two duct connections.

CHAPTER VIII

WATER-VAPOR REFRIGERATION

We are now ready to study the various means of producing refrigeration artificially by man-made means. When we speak of refrigerating effect or refrigeration, we mean producing a lower temperature than the prevailing outdoor or room temperature, within an enclosed space. The early efforts in the direction of refrigeration were confined to the use of naturally frozen ice, which was cut in the wintertime and stored until the following summer.

As you know, when ice melts a change of state is involved and Btu must be added to the ice to change it into liquid water. The melting temperature of ice and, hence, the temperature of the liquid water produced from the meltage is 32°F. Therefore, it is possible to obtain temperatures as low as 35°F, in space refrigerated by ice. Ordinarily, however, 40°F is the lowest temperature that can be obtained practically.

Obviously, it takes more ice to cool a large "walk-in" refrigerator in a butcher store than would be required to cool a domestic refrigerator. Similarly, a great deal more ice would be required if a building such as a theater were to be cooled by the melting of ice. We have to have a unit in which to measure refrigerating effect when it is on a large scale. Since man's earliest efforts at refrigeration were based on the meltage of natural ice, the unit we now use to measure artificial refrigeration is based upon the meltage of ice.

Meaning of "One Ton of Refrigeration Capacity." The latent heat required to melt 1 lb of ice at 32°F to 1 lb of water at 32°F is 144 Btu. Hence to melt 1 ton of ice (which is 2,000 lb), requires 288,000 Btu. Suppose that a certain huge refrigerator melted exactly 1 ton of ice in one 24-hr day; this we define as a *refrigerating capacity of* 1 *ton.* If a small theater required 1 ton of ice each hour for cooling, for 24 such cooling hours, it would require 24 tons of ice, so that the capacity of its cooling plant would be 24 tons. We can now show 1 ton of refrigeration by any one of the following expressions:

1 ton of refrigeration = 288,000 Btu removed per 24 hr
1 ton of refrigeration = 12,000 Btu removed per hour
1 ton of refrigeration = 200 Btu removed per min

To get the number of Btu per hour equivalent to 1 ton of refrigeration, we divide 288,000 by 24. Similarly, we divide 12,000 Btu by 60 to get the number of Btu per minute that is the equivalent of 1 ton of refrigeration.

All large-sized artificial-refrigeration-producing machinery has its capacity stated as tons of refrigeration. This is true regardless of whether the refrigeration machinery is of the two water-vapor types that we are going to study in this chapter, or one of the many mechanical types that we are going to study in Chap. X, or one of the other nonmechanical types that are built like the Electrolux gas refrigerator. It is for this reason that we bring up this unit in which refrigerating capacity is measured before studying refrigeration machinery.

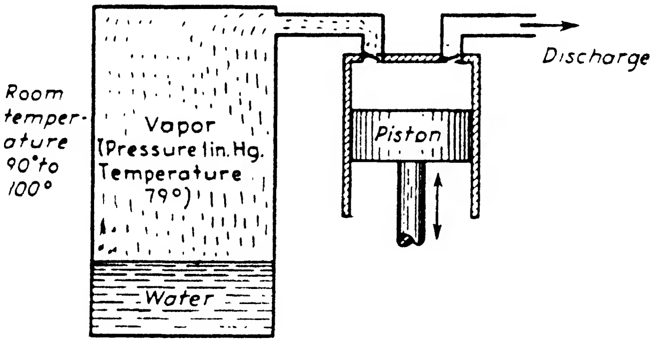


FIG. 1.—Method of reducing pressure over liquid.

Boiling Point at Low Temperatures and Low Pressures. From our study of the Steam Tables and the Psychrometric Tables you have learned that the lower the absolute pressure, the lower the temperature at which water boils. Suppose that we could, in a tank, obtain a pressure of 1.0 in. Hg. From the Steam Tables you can see that the corresponding boiling temperature of water would be a fraction of a degree above 79°F. If this tank were simply a closed tank and if we were heating the water by room air at 101°F outside the tank, the Steam Tables show that the pressure in the tank would increase to 2 in. Hg, under which the boiling of the water could cease unless it were heated higher than 101°F. The tank, discussed above, is assumed to contain no air whatsoever, but only water vapor above the surface of the liquid water.

Let us suppose that we have a tank as shown in Fig. 1. As you can see this tank is exactly like the tank discussed in the preceding paragraph, except that we have a piston type of pump connected so that it will remove the water vapor from the tank as fast as the water vapor is formed. By this action we can keep the absolute pressure in the tank always at 1.0 in. Hg. Of course the pump would have to be carefully regulated so that it would remove the water vapor at exactly the same rate that the vapor was formed. This is necessary to be sure that the pressure will remain constant at an absolute pressure of 1.0 in. Hg, but we shall assume that this is possible. Referring to Fig. 1, let us suppose that the liquid water was originally at room temperature before the pump was operated. The first water to change its state would get its latent heat of vaporization from the body of warm water. The temperature of this water would drop until it read 79°F, the phenomenon being, in general, the same as the dropping of a wet-bulb thermometer. Once the water reached 79°F, the only heat available to continue boiling would be the sensible heat that would flow from the warm air through the tank wall and into the water.

This crude machine illustrated in Fig. 1 is a simple refrigerating machine, it makes it possible for us to "refrigerate" the space within the tank, since we are keeping the inside of the tank at 79° in a room that might be between 90 and 100°. To come within our definition of refrigeration, given in the first paragraph of this chapter, it is not necessary for us to produce extreme cold, it is only necessary that we reduce the temperature within an enclosed space below the temperature of the surroundings.

In order to produce lower temperatures by means of a refrigerating machine, as shown in Fig. 1, it is only necessary to build a better, tighter machine with an extremely good pump that will maintain absolute pressures below 0.5 in. Hg. From the Steam Tables you can see that an absolute pressure of 0.5 in. Hg would result in a boiling temperature of 58.8°. An absolute pressure of 0.25 in. Hg would result in a boiling temperature of 40.2°. As we shall see, it is possible commercially to produce pressures as low as 0.25 in. Hg and cause water to boil at about 40°F. When refrigeration machines of this type are used in air conditioning, we are not interested in simply producing a low temperature inside the tank, instead we want to produce chilled water for use in an air-conditioning unit. Suppose that an air-conditioning unit requires 40 gpm of chilled water at 40.2° and that this water is heated 6° in passing through the conditioner; a water-vapor refrigeration machine would then be required to take this 40 gpm of water at a temperature of 46.2° and cool the water back to 40.2°. We can calculate the tonnage of this machine by using our fundamental heat equation

$$\begin{aligned} Q &= cW(T) \\ Q &= (1)\ (40)\ (8.33)\ (6) \\ &= 2{,}000 \text{ Btu per min} \end{aligned}$$

In this equation, we multiply by 8.33 since each one of the 40 gal of water weighs 8.33 lb. We know that 1 ton of refrigeration has a heat-removal rate of 200 Btu per min. Hence,

$$\frac{2{,}000}{200} = 10 \text{ tons}$$

Thus we can see that our water-vapor refrigeration machine would have to be of 10 tons capacity. In Fig. 2, a refrigeration machine is shown, the same in principle as the machine in Fig. 1, that might be used to do this 10-ton water-chilling job we have just discussed.

Let us consider the changes that we have had to make in the refrigerating

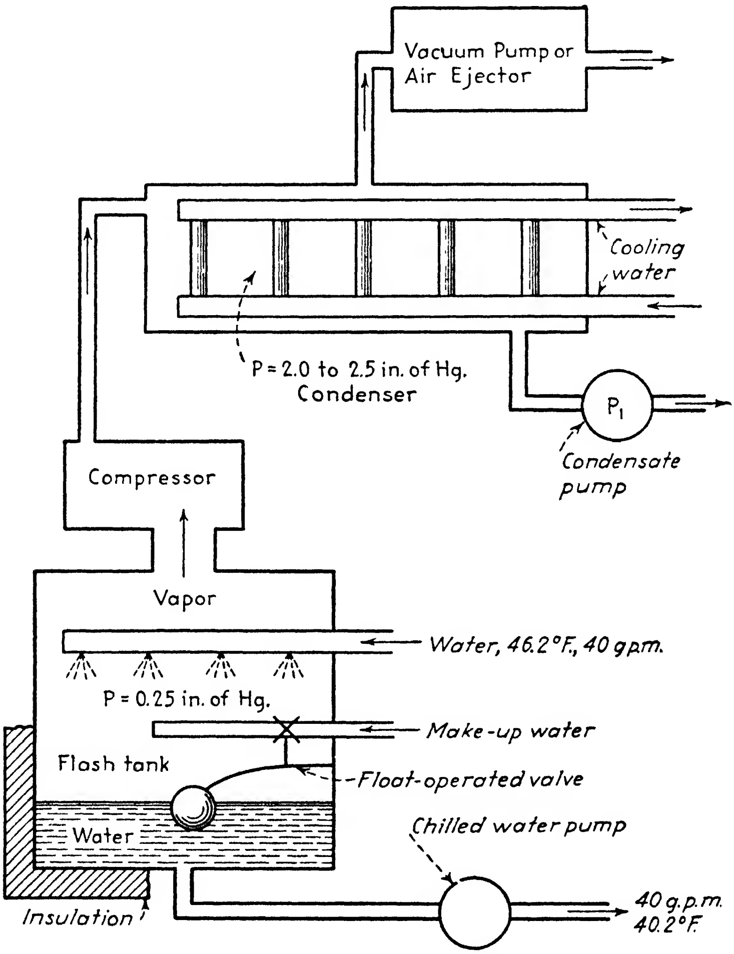


FIG. 2.—Diagram of simple water-vapor refrigeration machine.

machine between the crude device shown in Fig. 1 and the improved device in Fig. 2. We shall consider the several changes individually.

1. Look at the tank in Fig. 2. This is called a flash tank or an evaporator, in an actual machine. Note that the tank is now insulated to prevent heat from leaking into the tank from the room. This is done because any heat that leaked in from the room would be an additional burden on the

pump that is removing vapor from the flash tank. We want the pump to handle only the vapor absolutely necessary for our 10-ton refrigeration load. Note also that we now have two water pipes connected to our flash tank. The water at 46.2° enters the top of the flash tank and is sprayed through nozzles that break the water up into tiny droplets. Some water from the surface of each droplet changes state, taking the required Btu from the remainder of the droplet; thus all the Btu required to evaporate water are taken from the water itself. In this manner we refrigerate the water, which was our original intention in this type of machine. (Evaporation also takes place from the surface of the pool of water at the bottom of the flash tank.)

2. The pump shown in Fig. 1 is really a compressor; it draws in vapor at an absolute pressure of 1.0 in. Hg and compresses or squeezes the vapor so that this pressure becomes 29.92 in. Hg. The compressed vapor would then be discharged to atmosphere. This crude type of compressor, as illustrated in Fig. 1, would not work commercially. It would have to increase the pressure of the water vapor almost thirty times; *i.e.*, P_2 would have to be thirty times P_1. It is difficult to operate a compressor with such a high "compression ratio." There would also be other complications, which we need not discuss at this time.

The pump or compressor in Fig. 2 discharges into a closed tank. Inside this tank is a coil of pipe through which flows cooling water at a temperature of 85 to 90°. This cooling coil keeps the temperature within the enclosed tank from rising much above 100°F. This enclosed tank into which the compressor discharges its vapor is called the "condenser." Inside the condenser we maintain a low pressure of 2 to 2.5 in. Hg by means of the small vacuum pump shown. The vapor discharged by the compressor condenses back to liquid water inside the condenser, and the liquid water is removed by the condensate pump. The absolute pressure that is maintained within the condenser depends upon how cool the condenser is kept by the cooling water. If the condensing temperature is held at 101°, the absolute pressure in the condenser will be about 2 in. Hg. Under this arrangement the compressor raises the pressure of the water vapor from 0.25 in. Hg to 2 in. Hg. This is an increase in pressure of eight times the original pressure, or a compression ration of 8 to 1. Such a pressure increase is feasible in commercial operation.

3. The compressor itself cannot be of the piston type, which we originally showed in Fig. 1. Figure 2 does not explain anything about the nature or design of the compressor. The compressor is of the centrifugal type. A centrifugal compressor is in principle like a series of centrifugal fans or centrifugal pumps arranged in series. The first discharges into the inlet of the second, and so on. By means of a centrifugal compressor of several

stages, water-vapor refrigeration is commercially possible. Figure 3 illustrates more nearly the actual layout of a commercial system of this type.

Problems Involved in Water-vapor Refrigeration. Thus far we have merely tried to make clear the *principles* of operation of water-vapor refrigeration. We are now going to discuss some of the difficulties and problems that were involved before this type of refrigeration was made practical to use. It is obvious that the low absolute pressures required in order to obtain refrigeration by the direct vaporization of water would cause complications. It is difficult to produce such low pressures at all, and it is difficult to build a machine that will operate with such low internal pressures without having objectionable air leakage into the machine.

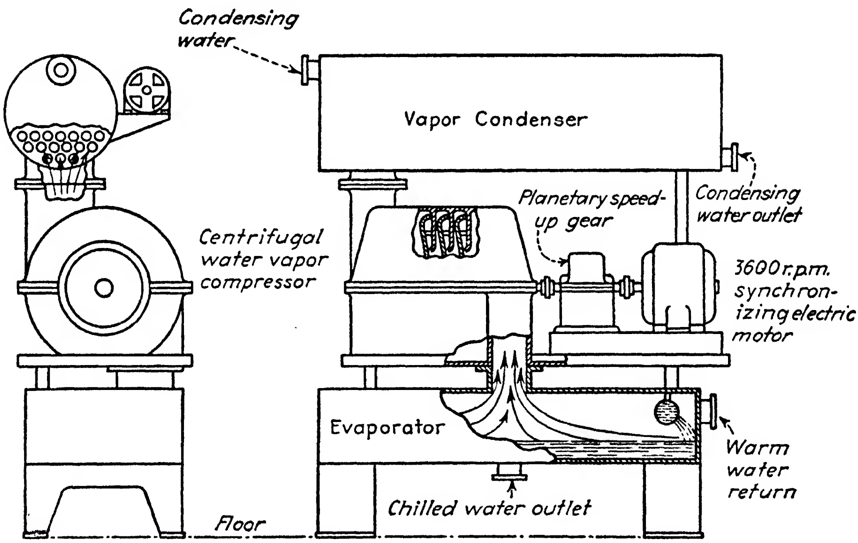


Fig. 3.—Mechanical layout of centrifugal refrigeration condensing unit.

After all, the air is at 14.7 psia, whereas within the water-vapor refrigeration machine a pressure as low as 0.2 psia is required. This means that the pressure of the atmosphere is seventy-five times as great as the pressure within the machine. The problem of leakage depends upon the pressure difference. The tendency for air to leak into a water-vapor refrigeration machine is about the same as the tendency for a compressed gas at 1,000 psi to leak outward from a tank.

Water on being admitted to a flash tank or evaporator, as shown in Fig. 2, boils or flashes to steam with great violence. Means must be provided to prevent the centrifugal compressor from sucking in droplets of liquid water, which are sprayed about owing to the violence of the boiling. With a flash-tank temperature of 50° corresponding to an absolute pressure of 0.36 in. Hg, the volume per pound of steam is approximately 1,700 cu ft. If you check the latent heat of steam (h_{fg}) at 50°, you will see that a 10-ton machine, such as we have been discussing, would require the remova' of about 3,500 cu ft of vapor *per minute.*

This large volume of water vapor must be compressed to a higher pressure, perhaps 2 in. Hg, so that the vapor can be condensed by cooling water at temperatures as high as 85°F. With this entire cycle taking place at extremely low atmospheric pressures, the problems involved in getting a commercial practical machine were much greater than those that confronted the designers of the early conventional refrigeration machines. We are considering water-vapor refrigeration first, not because it was the first commercial man-made refrigeration, but because we can understand it easily after having just studied the Steam Tables and Psychrometric Tables. As a matter of fact, commercial ammonia compressors were available generally before commercial water-vapor refrigeration was in general use. Figure 3 illustrates roughly the general layout on which a commercial water-vapor refrigeration machine might be built. (There is a mechanical refrigeration system that uses a machinery layout similar to this and that operates at pressures well below atmospheric pressure. Keep this entire discussion of water-vapor refrigeration in mind when we study another centrifugal refrigeration compressor in Chap. X.)

Let us consider a complete water-vapor refrigeration machine, as illustrated in Fig. 2. The discussion that follows applies to water-vapor mechanical refrigeration as well as to all other types of mechanical refrigeration that we shall study in Chap. X.

As we have seen in Chap. V, heat cannot flow from a colder body to a warmer body; instead it *always* flows from the warmer of two bodies to the colder body. Suppose we wish to cool a small room. We can put the evaporator of a refrigeration machine within the room. Since we can make the evaporator quite cold—40 to 50°—it is clear that heat will flow from the warm air of the room into the evaporator, and the temperature of the air in the room will be reduced. Where has the heat gone? Where are the Btu that flowed into the evaporator? As we have seen from the preceding discussion in this chapter, the sensible heat that flows into the evaporator is used to boil the refrigerant within the evaporator. (This refrigerant may be water or some other refrigerant.) The Btu that have flowed into the evaporator are now part of the heat content (h_g) of the gaseous refrigerant; we must remove this gaseous refrigerant from the evaporator as fast as it forms.

If our compressor can draw this vapor out of the evaporator and compress it to atmospheric pressure, by discharging the vapor into the atmosphere, we can get rid of the Btu that the vapor contains. With a water-vapor refrigeration machine the cost of the water thus drawn away would be negligible. However, we cannot compress the vapor to a pressure as high as atmospheric pressure because this would be a pressure increase of 120 times an evaporator pressure 0.25 in. Hg. How can we make the Btu leave the water vapor and flow to some other substance by means of which we can discard this heat? The solution of this problem is easy, and it happens to make it possible for us to save our refrigerant instead of discarding it.

On entering the compressor, the refrigerant vapor, in the case of our water-vapor machine, might be at a temperature of about 40°, if our evaporator pressure (absolute) were 0.25 in. Hg. If we are going to have the Btu content in this vapor flow away to a cooler body by normal heat flow, we must know of some way to increase the temperature of the gaseous refrigerant. Furthermore, since the Btu are contained in the refrigerant as latent heat instead of sensible heat, we must make the refrigerant change its state back to a liquid. The centrifugal compressor raises the absolute pressure of the vapor to, say, 2 in. Hg: the boiling point of water corresponding to this pressure is about 101°F. In other words, if we cool the vapor leaving the compressor by means of water at a temperature below 100°, the refrigerant will change state, becoming water at 101° and giving up its latent heat. If our cooling water were really at a temperature of just 100°, it would only be 1° cooler than the warm liquid formed from the condensing refrigerant. With this small temperature difference, we would have to use a tremendous amount of water for cooling because we can let the temperature of the cooling water rise only a fraction of a degree.

In actual practice, if we want a condensing pressure of 2 in. Hg abs, with a corresponding condensing temperature of 101°, we would use cooling water at a temperature below 90°. We would use a sufficient quantity of this cooling water so that it could carry away heat from the condensing water without rising above 93 or 94°. This would give us a temperature difference of 7 or 8° to cause heat to flow from the condensing refrigerant into the cooling water. The cooling water would then be thrown away or else sent to a device known as a cooling tower, which would recool the condensing water to its original temperature of, say, 85 to 88°F.

Let us summarize this discussion of the function of a condenser by tracing briefly the flow of the heat that we have just analyzed in detail. Let us start with warm air in a room that we wish to cool; we shall take heat out of this air, and then we shall trace what happens to this heat until we throw it away. Remember the whole problem of refrigeration is simply the removing of heat from a space where it is not wanted and arranging to discard it in a suitable manner.

1. Heat flows from the warm air of a room through the metal walls of an evaporator.
2. This sensible heat is used to boil the refrigerant.

3. The heat, which is now part of the heat content of a vapor, is removed from the evaporator because the vapor itself is drawn out of the evaporator.

4. The vapor is compressed to an absolute pressure several times as high as the evaporator pressure and is discharged into a vessel called the condenser. The heat we are following is still part of the heat content of this vapor.

5. Additional heat is put into the vapor by the mechanical work of compression. Each foot-pound of work of compression is converted into sensible heat (superheat) of the vapor.

6. Inside the condenser, the vapor encounters a mass of tubes containing water colder than the condensing temperature of the vapors. The vapor condenses to water, and its latent heat is converted back to sensible heat.

7. We now have the heat we are following in the form of sensible heat in a liquid whose temperature would be at about 100°. If the cooling water were shut off, the temperature in the condenser would rise because vapor is condensing rapidly and its latent heat is being converted back to sensible heat.

8. In normal operation, the cooling water is flowing and, since it is cooler than the condensing refrigerant, the heat that we are tracing flows into this cooling water and is discarded. (In the case of mechanical refrigeration machines using refrigerants other than water, which are rather expensive, the condensed refrigerant is piped right back to the evaporator.)

Water-vapor Refrigeration—Steam-jet Type. It is possible to produce the very high vacuum needed for water-vapor refrigeration without using a centrifugal compressor. By means of a carefully designed arrangement of steam nozzles, it is possible to have a jet of steam produce the high vacuum needed in a water-vapor refrigeration flash tank. This jet of steam also draws away the low-pressure cold steam that has been produced by the vaporizing of some water in the flash tank. A schematic diagram or flow sheet illustrating how a steam-jet refrigeration machine functions is shown in Fig. 4. Referring to Fig. 4 you find the flash tank and chilled water lines, with which you became acquainted in Fig. 2. Above the flash tank are the nozzles of the steam jet shooting steam into a passage labeled "booster." The shape of the walls of the booster must be carefully designed, or the machine will not operate properly. The booster is also called the "thermalcompressor," because it actually compresses steam vapors from absolute pressures of perhaps 0.35 in. Hg at the evaporator to a pressure of perhaps 2 in. Hg in the condenser.

The hot high-pressure steam shooting from the nozzle of the steam jet *all* travels through the booster and does not get down into the evaporator. Look at the upper right portion of Fig. 4. This device comprising two small steam jets is called an "air ejector." Turn back to Fig. 3 and you will find a vacuum pump or air ejector, shown at upper left, connected to the con-

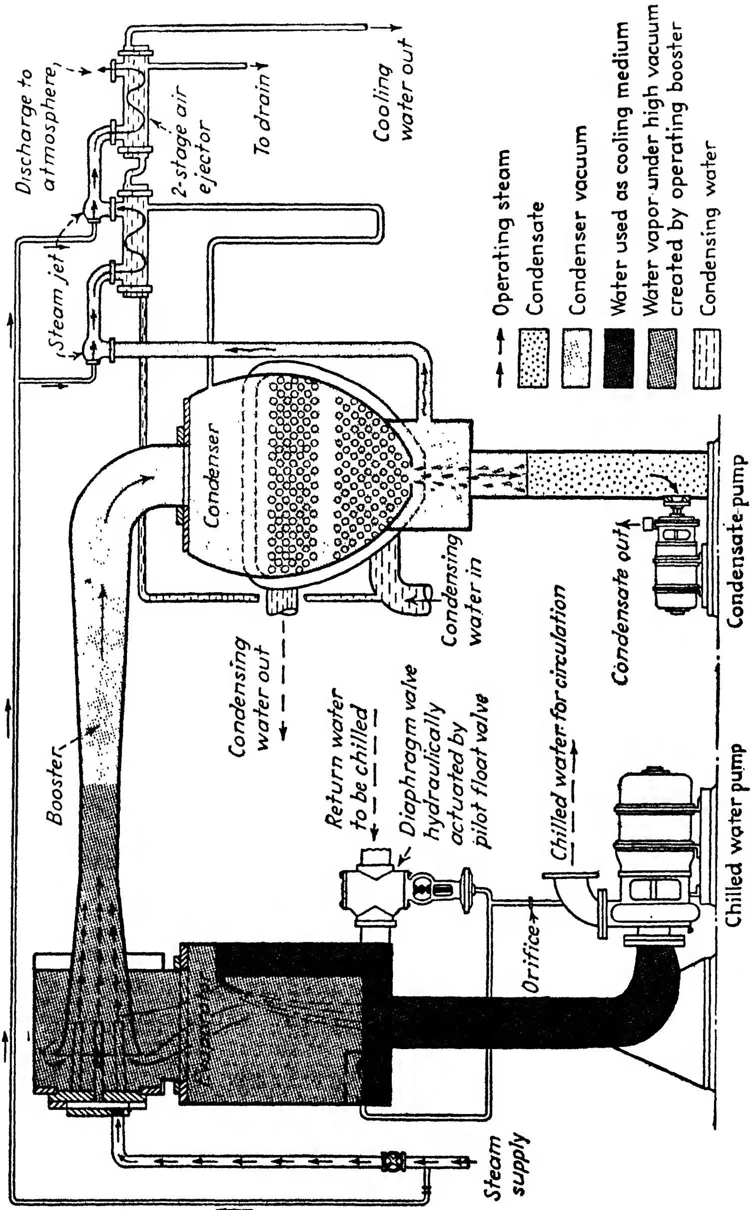


FIG. 4.—Steam-jet refrigeration machine. (*Ingersoll-Rand Co., Inc.*)

denser. In our steam-jet refrigeration machine in the laboratory, we use a water-jet air ejector. Some smaller size commercial steam-jet refrigeration machines use electrically driven mechanical vacuum pumps. However, if steam is available, a steam-jet air ejector is cheaper and simpler for machines of 10 tons capacity and upward. Have you determined why these air ejectors or vacuum pumps are required?

Surely you have had the experience of drawing a glass of water from a faucet and finding it milky in appearance. On allowing the glass of water to stand for 5 min, the cloudiness clears up completely. This cloudiness is caused by small particles of air that were dissolved in the water when it was under pressure. Upon releasing the water from, say, 65 psia to atmospheric pressure of about 15 psia, the air is released from the solution in the form of small bubbles. Similarly, the water in a high-pressure steam boiler is almost certain to contain some dissolved air. Though very small in quantity, this air travels along a steam pipe with the steam and crosses the booster, thus entering the condenser. If means were not provided for continually removing this air, what would happen? The air could not be condensed in the condenser; therefore the condenser would rapidly fill with an unwanted gas, *viz.*, air. This would cause the absolute pressure within the condenser to rise, and the thermalcompressor could not operate against the higher pressure.

If this condition occurred, the jet of high-temperature steam would no longer rush through the booster. The steam would instead spill out of the entrance to the booster and would fill the evaporator. The evaporator pressure would, of course, rise, and instead of chilling water in the evaporator the water would be heated by the hot steam. When this undesirable occurrence is permitted to happen, we say that the "jet breaks." We say this because the smooth flow of steam from the jet through the booster is broken.

In the case of a centrifugal water-vapor refrigeration machine, some air is bound to leak into the evaporator. Also if the chilled water is used in an air washer, some air will dissolve in the water. Dissolved air in the chilled water is released as a dry gas in the evaporator, and this air will not condense in the condenser. Therefore we need an air ejector or a "noncondensible ejector" on the condenser of any water-vapor refrigeration machine.

When steam-jet air ejectors are used, we have two steam jets in series, as shown in Fig. 4. The reason for this is that the steam jet in acting as a thermalcompressor has the power to compress to a final absolute pressure that is about five or six times the initial absolute pressure. The main steam jet or booster compresses vapor from perhaps 0.35 to 2 in. Hg. The first stage of the air ejector compresses the air it is removing from 2 in. Hg (condensing pressure) to perhaps 10 or 12 in. Hg at the entrance to the second stage of the air ejector. The second stage of the air ejector then compresses the air being removed from 10 to 30 in. Hg (atmospheric pressure).

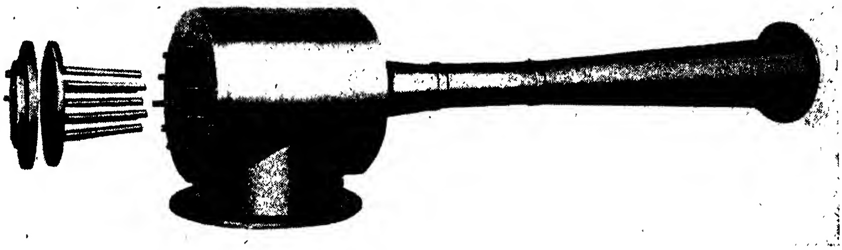


FIG. 5.—Steam-jet booster with the nozzles removed. (*Ingersoll-Rand Co.*)

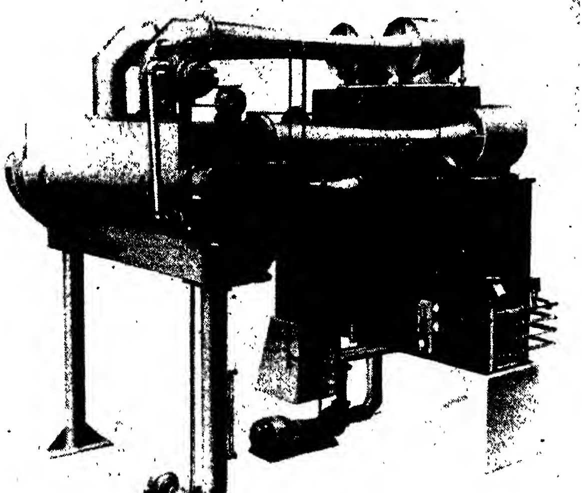


FIG. 6.—Completely assembled steam-jet refrigeration unit. (*Ingersoll-Rand Co.*)

The booster of certain steam jets is about the size of the air-ejector steam jets on a large steam-jet refrigeration machine. For this reason, a two-stage steam-jet ejector cannot be built. Therefore, a single water-jet air ejector that can compress the air from about 2 in. Hg to atmospheric pressure is used.

Figure 5 shows a steam-jet booster with the nozzles removed. This booster is of Ingersoll-Rand manufacture. Figure 6 shows a completely assembled steam-jet refrigeration machine also of Ingersoll-Rand manufacture. The machine in Fig. 6 has three main steam-jet boosters with a two-stage air ejector shown at the upper left. Note the thermometer built into the end of the condenser (extreme left) to record the temperature of the condensing water.

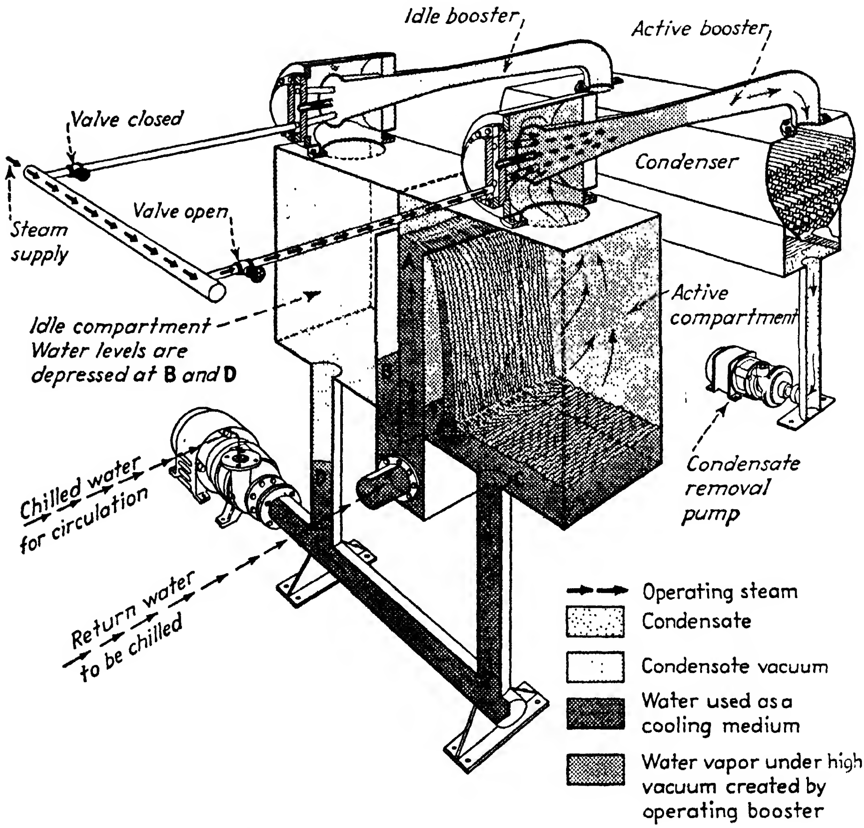


FIG. 7.—Diagrammatic sketch of a compartment-type evaporator. (*Ingersoll-Rand Co.*)

In some cases it is desirable to subdivide the evaporator into separate sections with one booster connected to each section. In the case of a 400-ton machine with four 100-ton boosters, impressive steam economies are possible. A single booster would be operated at full capacity, the other boosters being shut down. Like almost any other machine, a steam-jet booster is most efficient at or near full load. A diagrammatic sketch of an Ingersoll-Rand compartment-type evaporator is shown in Fig. 7.

Steam-jet refrigeration has been used widely for many years in *industries* requiring refrigeration. In the past 5 or 6 years it has become more and more popular for producing chilled water in air-conditioning installations. Therefore, we shall investigate its advantages and disadvantages compared with water-vapor refrigeration using the centrifugal compressor. In the first place, a centrifugal water-vapor refrigeration machine could be used where electric power is cheaper than steam. A steam-jet refrigeration machine obviously requires an abundant supply of steam at an energy cost preferably lower than the cost of electric energy for the same refrigeration tonnage. A steam-jet refrigeration machine requires more condensing water for cooling the condenser. Table 1 shows the heat that must be removed from the condensers of the two types of water-vapor refrigeration machines.

TABLE 1.—HEAT REMOVED FROM CONDENSER, BTU PER MINUTE PER TON OF REFRIGERATION

	Centrifugal water vapor	Steam-jet water vapor
Vapor from evaporator	200	200
Steam from jet	...	500*
Heat of compression	50*	...
Total	250	700

* Approximate—plus or minus about 10 to 20 per cent.

The heat of compression represents the temperature increase caused by the sudden compression of the vapor. You will recall that, in Chap. III, we said that when a gas was suddenly compressed the speed of motion of its molecules was increased and its temperature raised. In the case of any type of mechanical refrigeration, including the centrifugal water-vapor machine, the heat of compression is the Btu equivalent of the work in foot-pounds done by the electric motor. For instance, a 1-hp electric motor would do 33,000 ft-lb of work each minute, all of which would appear as heat in the gas or vapor being compressed. This heat would be

$$\frac{33,000}{778} = 42.5 \text{ Btu per min}$$

Neither steam-jet nor centrifugal water-vapor refrigeration is desirable when water temperatures much below 45°F are desired. Figure 8 shows how the capacity, or tonnage, of a steam-jet refrigeration machine decreases as the chilled-water temperature is dropped. From Fig. 8 you can see that a steam-jet machine that would have 100 tons capacity when producing 50° chilled water would have only 70 tons capacity when producing 40° chilled water and only 60 tons capacity at about a 37° chilled-water temperature.

From Fig. 9 you can see that the characteristics of a centrifugal water-vapor machine are similar to those of a steam-jet refrigeration machine of the same size. Figure 9 shows typical hypothetical data and is not intended to represent test data on an actual machine.

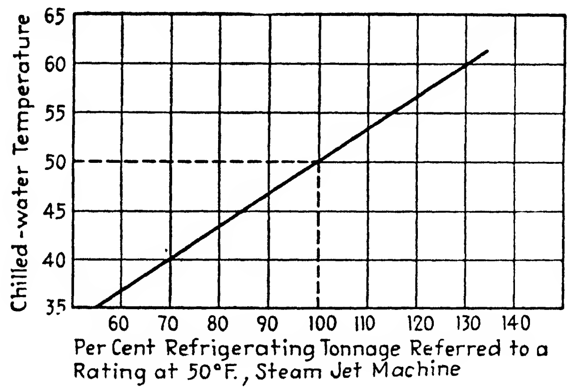


FIG. 8.—Steam-jet refrigeration machine performance curve.

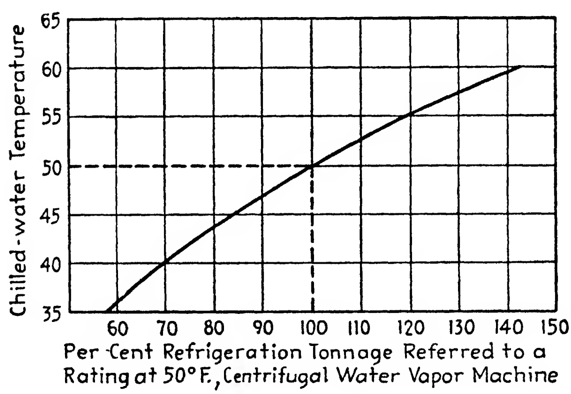


FIG. 9.—Centrifugal water vapor machine performance curve.

The characteristics illustrated in Figs. 8 and 9 are not all on the disadvantage side. An increase of 5° in chilled-water temperature increases the tonnage of one of these units almost 20 per cent. This represents a much greater tonnage increase than could be obtained with ordinary refrigerating equipment by a similar increase in the chilled-water temperature.

There is another advantage in the use of water-vapor refrigeration on large-size air-conditioning jobs, which you can easily appreciate. In large systems, air-conditioning refrigeration usually must produce chilled water that is piped to the various air-conditioning units. If water-vapor refrigeration is used, an evaporator temperature of 48° produces 48° chilled water for use in the systems. If ordinary mechanical refrigeration is used, the evaporator of the mechanical systems must be at 38 to 40° in order to chill water to 48°. This 8 or 10° temperature difference is necessary so that heat will flow from the water to the refrigerant.

From the foregoing you may have wondered why all air-conditioning refrigeration is not of the water-vapor type. As a matter of fact, the majority of air-conditioning refrigeration is standard mechanical refrigeration of the type we shall discuss in Chap. X. There are several reasons for this. In the first place, steam-jet refrigeration equipment costs more to install than ordinary mechanical refrigeration equipment. Its cost is reasonable only in the case of large installations. Furthermore, if city water is used to cool the condenser, the water bill will be roughly doubled for a steam-jet installation. If a cooling tower or other means is used to eliminate the need for city water, the cost of this water-saving equipment is almost double for the steam-jet plan.

Finally, steam-jet refrigeration, to be economical for air conditioning, requires an abundant supply of inexpensive steam, which is available in summer. In a very few cases, buildings generating their own electric power may have high amounts of steam at 5 psig to throw away during the summer. If such buildings have a sufficient supply of condensing water, the steam-jet installation is desirable. The lower the pressure, the more steam the steam jet uses and, hence, more heat is removed from the condenser. For example, by changing the nozzle in a certain steam jet, it will operate with steam at 13 psig instead of 75 to 80 lb per in. At the lower pressure it requires roughly 50 per cent more steam per minute than at the higher pressure. Therefore, in the case of most buildings, where condensing water would involve a definite cost, steam-jet refrigeration is possible only with steam at 100 psig, or thereabouts. Steam at such pressures is expensive, and frequently it is cheaper to purchase electric energy with which to drive other types of refrigeration equipment.

Centrifugal water-vapor refrigeration, because of the many problems that we have discussed, is not being offered by many manufacturers. Although there are a number of successful installations, the cost of installation of the equipment is high. It has proved difficult to convince the majority of the air-conditioning purchasers that this type of refrigeration offers advantages that are worth the increased cost.

You should now have an understanding of the fundamentals of refrigeration, using water as the refrigerant. We have presented this type of refrigeration first because water is a familiar substance and because we have studied its properties as to pressure and boiling temperatures in the preceding two chapters.

PROBLEMS

1. We know that it is possible to cool water, by evaporation of a portion of the body of water, within a chamber maintained at a very low absolute pressure. Explain how this can be done, citing the Steam Tables for illustration if you desire.

2. What types of pumps (compressors or evacuators) are desirable for maintaining the very low absolute pressure necessary in a chamber such as you have described in Prob. 1? Why is an ordinary reciprocating compressor unsuitable for this service?

3. A steam-jet flash tank cools water at the rate of 1,000 lb per min from 70 to 60°F. A portion of the water is, of course, evaporated to provide the cooling effect. What vacuum must be maintained? *Approximately* how many pounds of water per minute will be evaporated? (Use h_{fg} value at *about* 65°, a fair average of the temperature at which the evaporation occurs.)

4. A water-vapor refrigeration machine evaporates water at a temperature of 40°F. From the Steam Tables, find the specific volume of dry saturated steam (water vapor) at a temperature of 40°F. If 2 lb per min of water vapor is evaporated, how many cubic feet per minute of water vapor must be removed from the evaporator or flash tank?

5. If a 10-ton water-vapor refrigeration machine makes chilled water at 50°F, find (*a*) the latent heat of vaporization (h_{fg}) of water (take the value at 50°F), (*b*) the specific volume of the water vapor leaving the evaporator, (*c*) how many pounds of water per minute must be evaporated to produce the desired 10 tons of refrigeration, (*d*) how many cubic feet per minute of vapor must be removed from the evaporator.

6. Referring to Fig. 2, describe the purpose of the vacuum pump or air ejector shown connected to the condenser.

7. Draw from memory a simplified sketch of a steam-jet refrigeration machine. Your sketch may illustrate the type of steam-jet arrangement illustrated in Fig. 4. Mark typical evaporator and condenser temperatures and pressures on your sketch.

8. Water-vapor refrigeration equipment, whether of the steam-jet or centrifugal-compressor type, exhibits the same characteristics of sharp reduction in capacity as the chilled-water temperature is dropped. Assuming a machine to operate at 100 per cent of rated capacity at a chilled-water temperature of 50°, tabulate the per cent capacity that might be expected at chilled-water temperatures from 35 to 60°.

9. From your knowledge of water-vapor refrigeration, as described in this chapter, would you select either type of water-vapor refrigeration for a 12 hr per day theater in a New York City building where no roof space was available for a cooling tower. Give two or three positive reasons for your answer.

10. A certain steam-jet machine normally uses 2.0 lb of main jet steam for each pound of vapor drawn out of the flash tank. Taking h_{fg} in the evaporator as 1,065 Btu per lb, and in the condenser as 1,040 Btu per lb, find the condenser heat release for a 10-ton machine. What is the condenser temperature?

CHAPTER IX

PROPERTIES OF REFRIGERANTS FOR MECHANICAL SYSTEMS

In the previous chapter we discussed the use of water as a refrigerant. In that case the compressor was of a centrifugal type and the water throughout the refrigeration cycle was far below atmospheric pressure. There are a variety of so-called chemical refrigerants that vary from simple common chemicals to complex chemicals. These chemical refrigerants, like water when it is used as a refrigerant, are in a liquid state at certain points in the refrigeration cycle and in a gaseous state at other points of the cycle. We shall discuss the properties of some of these refrigerants and the relation of these properties to the use of these refrigerants in compression-refrigeration systems. We shall find that data regarding the refrigerants we are going to discuss are available in tables similar to the Steam Tables. There is a separate table for each refrigerant, and for each refrigerant there is a table for the dry saturated vapor and a table for the superheated vapor.

Thermodynamic Properties of an Ideal Refrigerant. In the preceding chapter we discussed refrigerating effect and the unit (ton of refrigeration) used to measure refrigerating effect. As we learned in the case of water-vapor refrigeration, refrigerating effect is produced by causing the refrigerant to boil in a chamber called the "evaporator." When water vapor was used as the refrigerant, this boiling removed heat from the water itself, chilling the water. In the case of the chemical refrigerants, or the so-called "standard gas refrigerants," the latent heat to boil the refrigerant comes from outside the evaporator. In air-conditioning applications, we ordinarily do one of two things. Sometimes the evaporator itself is a finned coil through which warm air is blown. In this case, heat flows from the warm air into the refrigerant which is boiling inside the evaporator. In other cases, the evaporator is a tank inside of which is a coil made of pipe or tubing. Warm water flows through this tubing, and heat flows from the water to the boiling refrigerant; thus the water is refrigerated.

One ton of refrigeration means the extraction of 200 Btu per min. If we had a refrigerant having a latent heat of, say, 3,000 Btu per lb, only 1 lb of refrigerant per minute would have to be boiled in a 15-ton refrigeration machine. From this you can see that the larger the latent heat of a refrigerant, the better the refrigerant when considered from this standpoint alone.

In the case of water-vapor refrigeration, one disadvantage we stated was the very low absolute pressure that had to be maintained in the evaporator in order to get the desired low temperatures. It would be more desirable to have a refrigerant that would boil at 40°F under a pressure somewhat greater than atmospheric pressure. This would avoid any possibility of air leaking into the evaporator. If a leak did occur, the refrigerant would leak outward. Such a leak is much easier to find than a leak where air is seeping inward. Therefore, if a refrigerant boils at a temperature of 30 to 40° at pressures somewhat below atmospheric pressure (but not *extremely* low), we have better evaporator conditions than we had in the case of water. Therefore, let us say that our second thermodynamic property of a refrigerant should be "a 'reasonable' evaporator pressure corresponding to the evaporator temperatures we want."

In our study of water-vapor refrigeration we saw that the water vapor had to be compressed to a condensing pressure (absolute) several times greater than the absolute pressure in the evaporator. This condensing pressure must correspond to a boiling temperature for a refrigerant 15 or 20° higher than the temperature of the warmest condenser cooling water that is likely to be available. Using a cooling tower, it is not economical to get cooling water lower than 83 to 85°F on a hot summer day in New York City. In many Southern and Middle Western cities, the temperature of the city water reaches 85° or higher during the month of August. Therefore, a useful refrigerant must have, at a condensing temperature of 100 to 110°, a corresponding condensing pressure not too many times greater than the evaporator pressure. Summarizing the three thermodynamic properties we require of an ideal refrigerant, we have the following:

1. Large latent heat
2. A reasonable evaporator pressure corresponding to the evaporator temperatures we want
3. At condensing temperatures of roughly 110°, a corresponding condensing pressure not too many times greater than the evaporator pressures.

Other Physical and Chemical Properties of an Ideal Refrigerant. The physical and chemical properties of refrigerants are closely related, and we shall not endeavor to separate them in our discussion. In this day when physical chemistry is an accepted scientific field, it is hard to say where the boundary line lies between chemistry and physics. In connection with refrigerants, most of the properties that we are going to discuss are the results of the application of a little common sense to the problem. There is little of a technical nature involved.

For air-conditioning applications, we may easily draw up a list of the desirable general properties of an ideal refrigerant. Since leaks do occur in connection with refrigeration machinery, it is important that an ideal refrigerant be nonpoisonous and not have any bad effect on the respiratory organs of human beings. In order to help locate leaks it would be helpful to have a refrigerant that had a slight but noticeable and not too unpleasant odor. Since refrigeration machinery is often located near open appliances (gas heaters, etc.), a refrigerant should be noninflammable so as to avoid the risks of fires or explosions.

Ordinary refrigeration compressors resemble automobile engines considerably in their mechanical construction. Refrigeration compressors are made of cast iron or some other iron-alloy casting. Steel is also used in the construction of compressors. Of course, compressors require lubrication just as any automobile engine does. From this we can conclude that a refrigerant should be noncorrosive; *i.e.*, it should not attack or eat iron, iron alloys, copper, or brass. Furthermore, refrigerants should not react chemically (form a chemical combination) with lubricating oil. Lubricating oil sometimes combines chemically to form a sludgy lardlike compound having no lubricating value whatsoever. If this were to occur in a compressor, it would be just as dangerous as driving an automobile with no oil in the engine.

The desirable general characteristics for an ideal refrigerant are as follows:

1. No obnoxious odor
2. Nontoxic
3. Noncombustible and nonpoisonous
4. Noncorrosive on steel, iron, brass, or copper
5. Mix readily with oil physically but not combine chemically
6. Slight pleasant odor, if possible.

All these characteristics can be obtained in an actual refrigerant to a greater or lesser extent. Sulphur dioxide (SO_2) is a commonly used refrigerant in domestic refrigeration and commercial low-temperature refrigeration. It meets all the above qualifications except that it has an extremely unpleasant odor and in sufficiently strong concentration it has an irritating effect upon the nose, throat, and lungs. Extremely strong doses of sulphur dioxide can cause serious harm to the eyes and can even cause death.

Carrene (methylene chloride, CH_2Cl_2) meets all the requirements listed above, except that it does not have any noticeable odor. As a matter of fact, there is no refrigerant that has a slight but noticeable and pleasant odor. Refrigerants all seem to have either no odor at all or a rather unpleasant and irritating effect on the nostrils. Carrene has been widely used in air conditioning, but we shall see that it requires a special type of centrifugal compressor similar to the compressors used with water vapor.

In air-conditioning work we are limited almost exclusively to two refrigerants. Methyl chloride (CH_3Cl) is used considerably in places where its use is not prohibited by law. Methyl chloride meets most of our list of desirable properties. Liquid methyl chloride may be spilled on the floor without harm to the occupants in the room, provided they leave before the concentration of methyl chloride gas becomes too high. The odor of methyl chloride is not irritating or unpleasant. However, methyl chloride is definitely inflammable, and when mixed in the gaseous state with just the right amount of air it can cause an explosion. Nevertheless, the record in communities where methyl chloride can be used does not indicate that it is a dangerous refrigerant because of its slight defects. (A good many older domestic refrigerators use methyl chloride.)

In New York City and in many other larger municipalities, air-conditioning refrigeration is restricted to the use of so-called "Freon gases." The most common of these is called Freon 12 for short. Chemically it is dichlorodifluoromethane (CCl_2F_2). Freon meets all the properties for an ideal refrigerant which we have listed above, except that it has no odor at all. When Freon is heated to high temperatures so that it decompresses (breaks up chemically), the new products formed are poisonous. An exposure of 2 hr to a mixture containing 30 per cent decomposed products from Freon 12 causes death or serious injury to guinea pigs. Fortunately, however, these decomposed products are irritating when inhaled in slight quantities. Therefore, people would not remain exposed long enough to do any harm. These decomposed products are not inflammable. However, Freon rarely can encounter temperatures to break it down in this manner in the case of an air-conditioning installation. Therefore, we must conclude from the laws that have been passed that the Freon refrigerants are the safest and come closest to being harmless to humans.

There are certain other properties of refrigerants that affect their desirability for use in mechanical refrigeration machines. Obviously, a refrigerant must have a low freezing point, because we often wish to have evaporator temperatures as low as 35° in ordinary air-conditioning work and below 0° in some industrial work. On the other hand, the critical point, which is the same as the critical point we studied in connection with steam, must be rather high. One of the worst properties of carbon dioxide as a refrigerant is the fact that it has a critical temperature of about 87°. Remember that this means that the refrigerant cannot be liquefied at a temperature above 87° no matter how high the pressure. In Table I, we list all the properties desired in refrigerants, as given in the "1939 Refrigerating Data Book," published by the American Society of Refrigerating Engineers.

We have discussed all the properties listed in Table 1 before, with the exception of Nos. 8, 12, 13, and 14. Properties 12, 13, and 14 are beyond the scope of our discussion of refrigerants. The refrigerants commonly used in air conditioning are satisfactory regarding these properties, and we do not have to decide how well a newly discovered refrigerant meets them.

What is meant by the phrase "low piston displacement" mentioned as item 8 in Table 1? You will recall that in Chap. VIII, we said that a 10-ton water-vapor refrigeration machine at an evaporator temperature of 50° would require the removal from the evaporator of 3,500 cu ft per

TABLE 1.—PROPERTIES DESIRED IN REFRIGERANTS

1. Suitable evaporator pressure
2. Suitable condensing pressure
3. High critical point; low freezing point
4. Low price
5. High coefficient of performance
6. Low density of vapor and liquid
7. High latent heat, high specific heat of vapor, and low specific heat of liquid.
8. Low piston displacement
9. Inertness
10. Stability
11. Noncorrosiveness
12. High electrical resistance
13. Low viscosity, gas and liquid
14. High conductivity
15. Suitable properties with respect to oil
16. No toxic properties
17. No explosive properties
18. No irritating properties
19. No properties injurious to goods, foods, etc., depending upon application
20. Detectable leaks; low leakage loss

min of vapor. If this vapor were being removed by a reciprocating or piston-type compressor, the piston displacement per unit of time would have to be 3,500 cfm. It is obvious that the fewer cubic feet of refrigerant vapor that must be removed from the evaporator *each minute*, for *each ton* of refrigeration, the greater the capacity of a given refrigeration compressor operating at a fixed speed. Suppose we have a refrigeration compressor with one cylinder having a cross-sectional area of ¼ sq ft, a piston stroke of 1 ft, operating at 400 rpm. For a single stroke, this piston would have a displacement of ¼ cu. ft.; thus at 400 rpm. this compressor could withdraw 100 cu ft of refrigerant vapor from an evaporator each minute.

Table 2 shows five properties of refrigerants. This table shows data only for seven of the most common refrigerants, but these data are available for all commercial refrigerants.

We have discussed refrigerants with regard to their toxic or poisonous effect. Table VI[1] gives the toxic properties of various refrigerants including several that are not encountered ordinarily in air-conditioning work. Table VII gives data regarding those refrigerants which are readily combustible. Methyl chloride is the only one of the refrigerants listed in Table VII which is ordinarily encountered in air-conditioning work. Ammonia, although it is the most common refrigerant in ice manufacture and cold-storage plant refrigeration, is almost never encountered in air conditioning. Table VIII gives data regarding the explosive properties of certain refrigerants.

[1] Tables with roman numerals are given in the Appendix.

Table IX compares various refrigerants as to the displacement per ton, latent heat, and various other factors that affect the efficiency and the size of a refrigeration machine. Item 16 in Table IX, "Carnot cycle," simply means the theoretical 100 per cent efficiency ideal cycle.

Note: All refrigerants require more horsepower per ton than the Carnot cycle [column (7)]; also all refrigerants have a lower coefficient of performance than the Carnot cycle [column (8)]. Hence the efficiency [column (9)] is lower for all actual refrigerants than the theoretical efficiency of the Carnot cycle.

In the titles of columns (3), (4), and (5) of Table IX, you will see the figures 5 and 86° mentioned. These temperatures are the Refrigerating Engineers Society standard temperatures for the evaporator and condenser temperatures of a refrigeration cycle. In air conditioning our evaporator temperature is rarely lower than 35°F; our condensing temperature, on the other hand, is ordinarily 15 or 20° higher than 86°F.

TABLE 2.—SUMMARY OF PHYSICAL-THERMAL REFRIGERANT PROPERTIES

Refrigerant	Boiling point, °F	Freezing point, °F	Critical point		Specific heat of liquid, 5–86° avg	Specific gravity of liquid, p = atm
			Pressure, psi	Temperature, °F		
Ammonia, NH_3	−28.0	−107.86	1,651.0	271.	1.12	0.684
Carbon dioxide, CO_2	−109.3	−69.9	1,069.9	87.8	0.77	1.56
Dichlorodifluoromethane, CCl_2F_2	−21 7	−247.0	582.0	232.7	0.23	1.48
Methyl chloride, CH_3Cl	−10.6	−144 0	969.2	289.6	0.38	1.002
Methylene chloride, CH_2Cl_2	103.7	−142.0	640.0	421.0	0.33	1.291
Sulfur dioxide, SO_2	13.6	−98 9	1,141.5	314.8	0.34	1.357
Water, H_2O	212.0	32.0	3,226.0	706.1	1 0	.....

It would be possible to make a table such as Table IX based upon an evaporator temperature of 40°F and the condensing temperature of 105°F. However, Table IX is still of value to us because we can see how one refrigerant compares with another, the properties of all of them being given for the same standard conditions.

Table X illustrates the suction and head pressures (the evaporator and condenser pressures), the difference between them, and the ratio of one to the other. Note that Table X gives this information not only for the standard refrigeration cycle of 5° evaporator temperature and 86° condenser temperature, but also for more usual air-conditioning temperatures of 40° suction and 100° condenser.

In the Appendix, tables are presented showing the properties of the following refrigerants both as dry saturated vapors and as superheated vapors:[1]

Table XI	Saturated ammonia vapor
Table XII	Superheated ammonia
Table XIII	Solid and saturated carbon dioxide
Table XIV	Saturated Freon 12
Table XV	Superheated Freon 12
Table XVI	Saturated methyl chloride
Table XVII	Saturated sulphur dioxide
Table XVIII	Superheated sulphur dioxide

The use of the tables giving the properties of refrigerants is similar to the use of the Steam Tables, with which you are already familiar. We have given you the tables regarding ammonia not because ammonia is widely used in air conditioning, but because ammonia is the standard industrial refrigerant and the tables regarding it are very complete. You may find them of value for future reference.

Problems in the use of these tables will be deferred until after Chap. X, at which time you will have a complete understanding of the mechanical refrigeration cycle.

PROBLEMS

1. For dry saturated ammonia vapor at 40°F, find the following properties: (*a*) corresponding saturation pressure, in pounds per square inch gauge, (*b*) latent heat of vaporization, (*c*) specific volume of vapor.

Find the same properties for saturated ammonia vapor at 100°F. (You may use 15 instead of 14.7 if you wish.)

2. Repeat Prob. 1, using the refrigerant carbon dioxide. Can carbon dioxide exist as as a vapor at 100°F? In other words, what is its critical temperature?

3. Repeat Prob. 1 using the refrigerant Freon 12.

4. Repeat Prob. 1, obtaining the information about methyl chloride and also sulphur dioxide.

5. Present the information that you have obtained in Probs. 1 to 4 in a table that will present a comparison of the five refrigerants.

6. Find the specific volume of the vapor and the total heat contained per pound of Freon 12, at 125 psig and a temperature of 130°F. For Freon at the same pressure but at 120°F, obtain the same properties by means of interpolation in the table. (See Table XV.)

7. What are the three most important thermodynamic properties of an ideal refrigerant? Which refrigerant for which you have data has a critical point so low that it is rarely of value in air conditioning?

[1] For those desiring a complete discussion of the properties of refrigerants including graphs of the properties of ammonia and Freon 12, we suggest that you refer to the "1939 Refrigeration Data Book," Part II, Secs. 9–10, pp. 97–134, American Society of Refrigerating Engineers.

8. With reference to the thermodynamic properties and to other general desirable properties, why is Freon considered such an ideal refrigerant? In giving your answer, compare Freon 12 with methyl chloride.

9. List the three most important thermodynamic properties and five or six desirable general properties of an ideal refrigerant. Then using your list as a standard of excellence, compare ammonia, Freon, water, and air as refrigerants. From your comparison table, what is the main disadvantage of water as a refrigerant, and why is air an extremely poor refrigerant?

10. Suppose that you have a compressor that can pump a certain number of cubic feet per minute of refrigerant vapor, the speed of the compressor being fixed. Assume that the compressor has an adequately large motor. Assume 5°F evaporator and 86°F condenser temperatures. From the general tables on properties of refrigerants, or from the condensed data in Tables IX and X, estimate roughly whether Freon 12, ammonia, or methyl chloride would produce the greatest refrigerating capacity.

CHAPTER X

MECHANICAL REFRIGERATION: THERMODYNAMIC CYCLE

We have already become familiar with the basic principles of any mechanical refrigeration cycle in our study of water-vapor refrigeration in Chap. VIII. However, when studying water-vapor refrigeration, we did not discuss some of the important details that are found in connection with conventional mechanical refrigeration using standard chemical refrigerants. You will recall that in the case of water-vapor refrigeration we were not concerned about getting the condensed refrigerant back to the evaporator because water is not costly or scarce. In the case of mechanical refrigeration using the expensive chemical refrigerants, we must always return our condensed liquid refrigerant to the evaporator for reuse.

Let us look at the evaporator alone as it would normally be constructed in a mechanical refrigeration system. For the moment, we shall forget all about the compressor, condenser, and other component parts of the system. This will mean that our evaporator will be exactly the same in principle as the evaporators that we studied in connection with water-vapor refrigeration, with one exception. The exception is that we must have tubes carrying the water to be chilled, these tubes being built within the evaporator. This arrangement is unnecessary when making chilled water with water as the refrigerant.

Thermodynamics of the Evaporator. Figure 1 illustrates a mechanical-refrigeration evaporator using Freon as the refrigerant. The function of the evaporator illustrated is to cool chilled water from 55 to 48°. In Fig. 1 note that we have a "liquid line" bringing liquid Freon under high pressure up to the evaporator. However, we want our Freon boiling within the evaporator to be at a pressure of only 50 psia. Obviously then, we need some type of needle valve or pressure-reducing valve in our liquid line just before it enters the evaporator. Without such a valve it would be impossible to maintain the necessary pressure difference between the high-pressure liquid and the low pressure we need in the evaporator. (Refer to Table XV[1] to check the pressure, temperature, and total heat data given in connection with Fig. 1.)

The evaporator in Fig. 1, known as the "flooded" type, is the type commonly used where the evaporator functions to chill water. It is called a "flooded" evaporator because a large body of liquid Freon is in the

[1] Tables with roman numerals are given in the Appendix.

evaporator at all times during normal operation. The level of this liquid is controlled and maintained automatically by the valve we discussed above. This valve opens and closes its small orifice under some type of automatic control to admit Freon to the evaporator as required to maintain the desired liquid level. The outside of this type of evaporator would be heavily insulated so that heat could not leak into the evaporator from the warm air of the room. All the heat that is absorbed by the boiling of our Freon would be removed from the water flowing through the coil inside the evaporator.

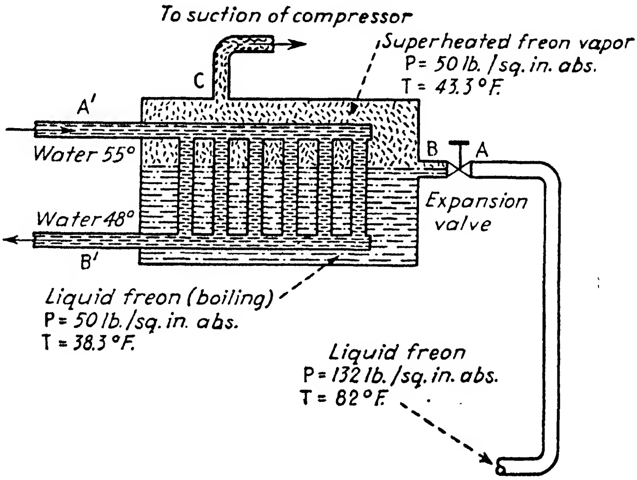


FIG. 1.—Diagram of mechanical refrigeration evaporator

One of the most important calculations to be made in connection with mechanical refrigeration is the determination of the number of pounds of Freon that must be circulated per minute for each ton of refrigeration desired. Suppose that we take the latent heat (h_{fg}) of Freon at a temperature of 38.3° and at a pressure of 50 psia. To get this from your tables, we must use an equation that has been studied in connection with the Steam Tables, *viz.*,

$$h_{fg} = h_g - h_f$$

Using this equation and interpolating in the table, the latent heat of Freon, at our evaporator condition, is found to be 65.73 Btu per lb. Does this mean that each pound of Freon circulated will remove 65.73 Btu from the water that is to be cooled? In other words, does this mean that the useful refrigerating effect from each pound of Freon circulated is 65.73 Btu? The answer to both these questions is definitely *no*. We must be careful about this as it is one of the most common mistakes in thinking about mechanical refrigeration. In the first place, the high-pressure liquid Freon is at a temperature of 80°F in the liquid line before it enters the evaporator. (We shall explain shortly where this liquid Freon comes from and why it is at the temperature and pressure conditions given.) Obviously, if we have Freon boiling in the evaporator at a temperature of 38.3°, any warm liquid that enters at 80° must be cooled to 38.3° before it can be of any service in removing Btu from the water we are trying to cool. Also note that the gaseous Freon leaving the evaporator is at a temperature of 43.3°F. Since it is 5° warmer than the boiling temperature within the evaporator, the gas leaving the evaporator is superheated 5°. The heat required to superheat this gas is taken from the water being cooled. Hence we have two reasons why we cannot take the latent heat value of a refrigerant at evaporator conditions and call this value our refrigerating effect per pound of refrigerant circulated. These two reasons are as follows:

1. The liquid entering the evaporator is warm and must be cooled to the evaporator temperature before it can do any good.

2. The refrigerant vapor is heated a few degrees after it leaves the surface of the liquid, and hence the refrigerant takes Btu from the water after it is in the vapor state.

Of course, it is possible to calculate the amount of cooling that must be done in order to cool the warm liquid entering the evaporator. This can then be subtracted from the latent heat of the refrigerant. We can then calculate the extra heat taken out of the water to superheat the refrigerant vapor. This quantity would then be added to the quantity just obtained by the subtraction described above. Thus we would finally be able to get the net useful refrigerating effect per pound of refrigerant circulated. However, this process is too cumbersome as it involves calculating two quantities and then subtracting one and adding the other to the latent heat of the refrigerant.

Instead, the evaporator should be considered as a boiler, for that is what it really is. You will recall that in Chap. VI we had a problem involving a steam boiler. Water entered the steam boiler with a certain heat content and was driven out of the boiler as steam, which also had a known heat content. The difference between these two heat values gave us the amount of heat that the boiler added to each pound of water. Similarly, in a refrigerant evaporator we have liquid at a certain total heat in the liquid line at point *A* (see Fig. 1) before it enters the evaporator. Also we have gaseous refrigerant at point *C* leaving the evaporator. We can easily find the heat content of the refrigerant at points *A* and *C*, thus giving us the heat content of the refrigerant before entering and after leaving the evaporator. Just as in the case of our steam-boiler problem, we can take the difference between these two quantities and obtain the increase in *heat* *content per pound of refrigerant passing through the evaporator.* This over-all increase in heat content gives us immediately the true refrigerating effect per pound of refrigerant circulated. To be sure that you understand this important principle, let us determine the refrigerating effect per pound of refrigerant circulated for an evaporator with conditions as shown in Fig. 1.

We want to find the heat content of liquid Freon at point A, which is in the liquid line. This value may be obtained sufficiently accurately by taking the total heat of liquid Freon at 80°, even though our actual refrigerant is at a somewhat higher pressure. From the table we find h_f (at A) = 26.28 Btu per lb. Now turn to the superheated Freon tables. Note that the column heading of the upper right-hand column gives saturation temperatures and pressure conditions similar to those which we have in our evaporator in Fig. 1. The table then shows specific volume, heat content, and entropy for temperatures from 50° up to 240°. Our superheated Freon at point C is at 43.3°, however; so we must extrapolate in the table. In this case we are *extrapolating* instead of *interpolating* because we are not finding a new value *between* two values given in the table. Instead, we are finding a new value *lower than the lowest value given* in the table. This extrapolation is performed in a manner similar to ordinary interpolations.

Temperature, °F	H
40	?
50	84.24
60	85.72

The difference in the heat content at 50° and at 60° is 1.48 Btu per lb. We know enough about the characteristics of superheated Freon vapor so that we are justified in *assuming* that there would be the same difference between the heat content at 40° and at 50°. Therefore, since our superheated vapor is at 43.3°F we have the following:

$$50° - 43.3° = 6.7°$$

$$\frac{6.7}{10}(1.48) = 0.98°$$

$$84.24° - 0.98° = 83.26°$$

In our study of our evaporator we now have the following:

$$h_f \text{ (at } A) = 26.28 \text{ Btu per lb}$$
$$H \text{ (at } C) = 83.26 \text{ Btu per lb}$$

As we have said, the difference between these two quantities equals our refrigerating effect per pound of refrigerant circulated. Therefore,

$$\begin{aligned}\text{Refrigerating effect} &= H \text{ (at } C) - h_f \text{ (at } A) \\ &= 83.26 - 26.28 \\ &= 56.98 \text{ Btu per lb}\end{aligned}$$

For the particular evaporator we are considering in Fig. 1, we must then set up the following expression to obtain the number of pounds of refrigerant that we must circulate *per minute per ton* of refrigeration capacity.

$$\frac{200}{56.98} = 3.51 \text{ lb of Freon per ton of refrigeration}$$

We wish to emphasize that this expression is good only when the evaporator conditions are exactly as shown in Fig. 1. If we change any of the following, a new expression must be set up for the new conditions:

1. The temperature of the warm liquid Freon in the liquid line
2. The absolute pressure (and hence the boiling temperature) in the evaporator
3. The temperature of the superheated gas drawn off from the evaporator.

We now want to look at the remaining parts of a complete refrigeration machine. In Fig. 1, we had only the evaporator and the expansion valve, with pipe connections for the liquid and suction Freon connections. We now want to see what happens to the superheated refrigerant vapor when it is drawn off from the evaporator and trace its path through the system until it returns to the expansion valve once more as warm liquid Freon. These other parts of the refrigeration machine are shown (*without the evaporator*) in Fig. 2.

Figure 2 shows the Freon vapor at point C entering the suction of the compressor. The pressure at this point is the same as the pressure in the evaporator, 50 psia. The compressor increases the pressure of this gas and does work upon the gas in compressing it. The mechanical work done by the compressor, as we saw in our study of water-vapor refrigeration, is converted into sensible heat, and the temperature of the gas, therefore, rises considerably. We have shown the gas at point D leaving the compressor at a pressure of 132 psia and at a temperature of 120°F. These conditions would vary, depending upon the size of the condenser and the temperature of the cooling water supplied to the condenser. Also the compressor itself must be cooled to prevent overheating due to this work of compression just mentioned. Small compressors have cooling fins cast in the outside of the cylinder and head castings. Larger compressors have the cylinder and head cooled by a water jacket similar to that which cools the head and cylinders of an automobile engine. The temperature of the gas leaving the compressor depends upon how much cooling is done to the compressor directly.

The pipe line carrying this hot compressed gas to the condenser is not insulated. By leaving this pipe bare, we help make it possible for this hot gas to lose some of its superheat to the cooler room air surrounding the pipe. On entering the condenser, the hot gas first loses all this superheat. The saturation or condensing temperature corresponding to an absolute pressure of 132 psi is seen from the saturated Freon table to be approximately 100°F. Therefore, the hot gas in Fig. 2 is superheated about 20°. On entering the condenser this gas would first cool to 100°, then as it continues to pass through the condenser, being cooled by the cooling water, the gas would condense to liquid Freon at a temperature of 100°. The latent heat liberated by the condensing gas would flow through the condenser tubes

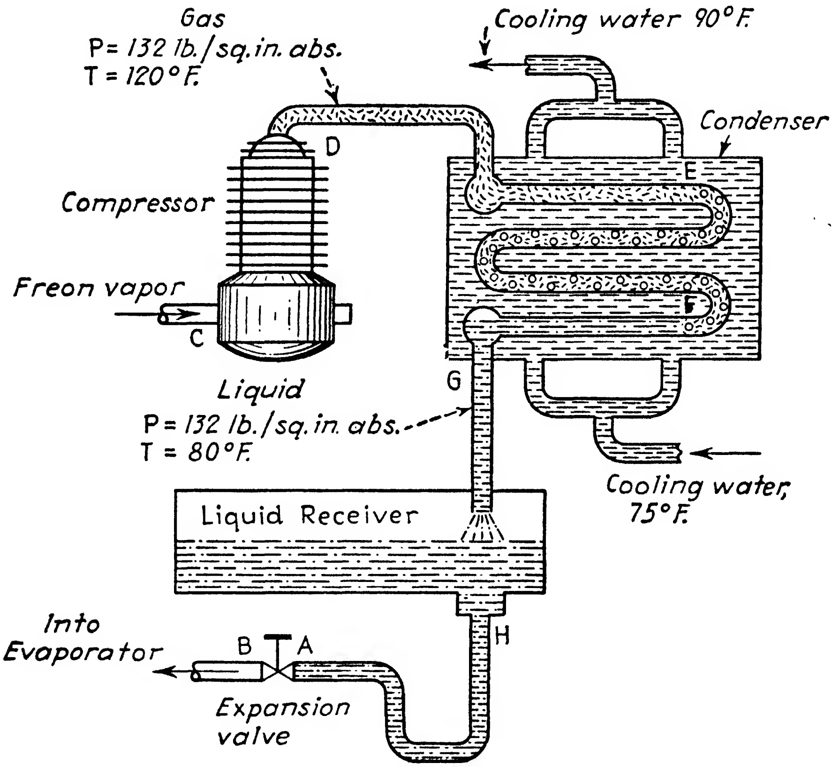


Fig. 2.—High side of mechanical refrigeration system.

and be carried away by the cooling water. Toward the end of the condenser, all the gas would be condensed back to a liquid. This liquid would then be cooled somewhat, depending upon how long a tube length remained before leaving the condenser. As shown in Fig. 2, the cooling water would enter the condenser at a temperature of 75°, *entering the end of the condenser at which the condensed Freon leaves.* The cooling water would then *leave* the condenser at a temperature of 90°, *leaving* at the end of the condenser at which the hot gas *enters.* We have shown the liquid Freon leaving the condenser at a temperature of 80°F. The condensing temperatures and conditions shown in Fig. 2 are reasonable for a condenser having plenty of capacity, using city water for cooling. The heat-flow arrangement shown, having the coolest cooling water enter and contact the coolest refrigerant, and having the warmest cooling water leave where the hottest refrigerant enters, is called "the countercurrent flow." Almost all shell-and-tube type heat exchangers operate on this principle of flow.

The liquid receiver shown in Fig. 2 is simply a reserve tank. The capacity of a liquid receiver should be great enough so that it can hold all the Freon of the system in a liquid state plus a reasonable safety margin. In a system such as illustrated in Figs. 1 and 2, the liquid receiver would contain but little Freon during normal operation, because most of the refrigerant would be over in the flooded evaporator. However, some types of evaporators do not operate flooded but, instead, have their expansion valve arranged to admit Freon as required without building up a great quantity of liquid Freon in the evaporator. Machines of the latter type do not require such large liquid receivers as do the flooded type.

Superheating Refrigerant Vapor in Evaporator. Most refrigeration installations are based upon superheating the refrigerant 5 to 10° in the evaporator. If we could be sure of getting dry saturated vapor or vapor superheated perhaps 1°, this would represent ideal operating conditions. Remember our temperature-volume gas law. The more we superheat the vapor in the evaporator, the greater its specific volume. This means that to draw away 1 lb of vapor the compressor must handle more cubic feet of vapor if the vapor is highly superheated. This would mean that the compressor would have to have a greater displacement than would be necessary if the gas were to be only slightly superheated. In other words, we must superheat the gas somewhat to be sure that we do not get any liquid refrigerant at all back to the compressor. It is undesirable to get liquid back to the compressor through the suction line for two reasons. (1) This liquid should have boiled in the evaporator to produce useful refrigerating effect. If it leaves the evaporator as a liquid, it has not produced the refrigerating effect of which it is capable. (2) In many compressor designs a small amount of liquid reaching the inlet to the compressor can do serious harm. If a large amount of liquid reached the compressor inlet, harm would be caused to any type of compressor. This injury could easily be as serious as ruining the valves and smashing either the head or the piston of the machine.

If an evaporator is too small or if we admit too much liquid Freon to a given evaporator, it will be impossible for enough heat to flow into the refrigerant to vaporize it completely. This must be avoided as it will result in damage to the compressor, because of liquid slugs, as outlined above. On the other hand, if an evaporator is too big compared with the other parts of the system or if the temperature of the warm substance that the evaporator is cooling is too high, the excess heat will flow into the evaporator. In this case, the liquid Freon admitted to the evaporator would be completely vaporized and then might be superheated 15 to 20°. Both of these extremes are undesirable. Evaporators must be carefully selected so that they will operate at the capacity and the operating conditions desired, superheating the refrigerant vapor 5 to 8°.

Thermostatic Expansion Valve. This type of expansion valve is used a great deal in air-conditioning work. It s arranged with a diaphragm or bellows which can register pressures from both sides. One side of the diaphragm registers evaporator pressures. The other side of the diaphragm is connected to a small thermostatic bulb and registers a pressure that depends upon the temperature of this thermostatic bulb. A spring with a tension adjustment screw also acts on the valve shaft in addition to the diaphragm. By means of this adjustment screw, such a valve can be arranged to open when the temperature at the thermostatic bulb is 5°, 7°, or any desired amount above the evaporator temperature corresponding to a particular evaporator pressure. Since different refrigerants have different evaporator pressures corresponding to a certain evaporator temperature, such expansion valves must be purchased for use with one specific refrigerant. For instance, Freon has an evaporator pressure of 51.7 psia corresponding to 40°F; methyl chloride has an evaporator pressure of 42.6 psia corresponding to this same temperature of 40°. In each case this evaporator pressure would be acting on one side of the diaphragm of the expansion valve. The pressure within the thermostatic bulb would be acting against the other side of the diaphragm so that you can see that a different type of thermostatic bulb would be required for Freon and for methyl chloride refrigerants. This bulb is fastened adjacent to the suction line, near the evaporator.

Subcooling of Refrigerants. We have mentioned that the warm refrigerant entering the evaporator must be cooled to the evaporator temperature before it can produce useful refrigerating effect. Actually, this happens almost instantly when the refrigerant passes the expansion valve. Referring to Fig. 1, note that the Freon at 80°F passes through the expansion valve, after which the pressure is only 50 psia. The saturation temperature corresponding to 50 psia is 38.3°. On the other hand, the saturation pressure corresponding to 80°F is 98.76 psia. This means that the vapor pressure of the Freon liquid at 80°F is much greater than the absolute pressure within the evaporator. Hence when a drop, a cupful, or a pound of liquid Freon enters the evaporator, a portion of it flashes immediately to vapor. The required latent heat is taken from the liquid Freon, thus reducing its temperature and its vapor pressure. Just a sufficient quantity of Freon flashes to a vapor to cool the remaining Freon liquid to a temperature corresponding to the absolute pressure within the evaporator.

(This is similar to our discussion regarding the drop in temperature of a wet-bulb thermometer in Chap. VII.)

The warmer the liquid refrigerant when it enters the evaporator, the more liquid must flash to a vapor in order to cool the remaining body of liquid. Refrigerant that evaporates thus does not produce useful refrigerating effect. On page 164, we had the formula

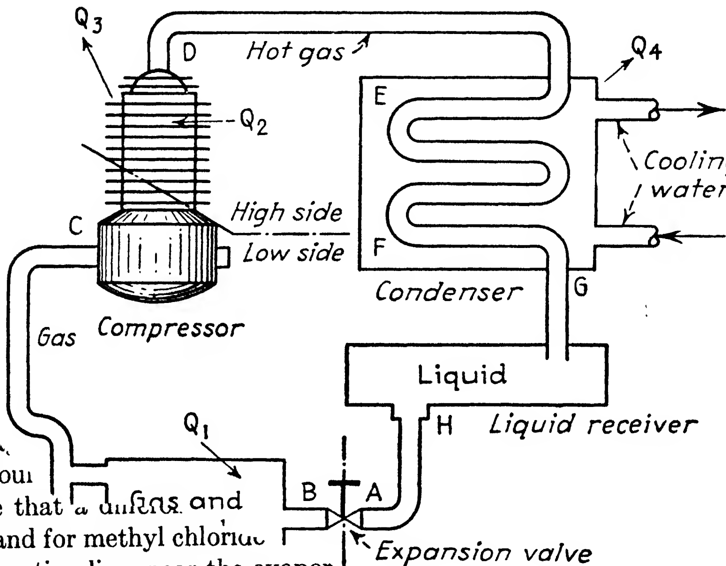
$$\text{Refrigerating effect} = H \text{ (at } C) - h_f \text{ (at } A)$$

From this equation you can see that the smaller the heat of the liquid that enters the evaporator, the greater the useful refrigerating effect. In other words, the lower the temperature of the liquid entering the evaporator, the more useful refrigerating effect obtained from each pound of refrigerant circulated. This means, of course, that the cooler the liquid entering the evaporator, the less pounds of refrigerant must be circulated per minute per ton of refrigeration capacity. This is one important reason for providing condensers sufficiently large, not only completely to condense the gaseous refrigerant, but also to subcool the liquid refrigerant before it leaves the condenser.

Subcooling of the liquid is important from another standpoint. It may not be economical to have a condenser very much larger than is barely needed to condense the refrigerant under conditions of maximum condens- be load. By maximum condenser load, we mean the combination of maxir- ype is tonnage output being required of the machine, at the same time th- it water condenser cooling medium is at its warmest temperature. A c- ressor will must be large enough to condense the refrigerant under these ... automobile conditions. If this were not the case, bubbles of ... from an operating stand- expansion valve and enter the evaporat- ...sign and build, compressors must be than these conditions of maximum ...ing either by air or by water. As stated correct size will deliver liq- ...ssors cannot be properly cooled by air alone and upon the amount of ... ket cooling.
the particular opera- ...cles as a Graph. Thus far we have not mentioned the subcooling are obt- ...y" in the Steam Tables; we have also omitted this same being a saving ir- ...is shown in the tables giving the properties of refrigerants. than maximum ...ations or any other practical value are concerned, we have no

Later in this ...y within the scope of this course. However, there are certain cally which w- ...re made using temperature as the vertical axis and entropy as trate superhe- ...al axis. These graphs are called "temperature-entropy charts"

Complete ..., and they can be drawn for steam and water or for the vapor- looked at tw- ...onship of any of our refrigerants. These charts present a vivid have chosen ...what happens to the refrigerant itself as it travels around through 2 because ...ical-refrigeration system. Temperature-entropy charts for steam There is a- ...in analyzing the boilers and engines in steam power plants, and e also of great importance in designing and building diesel engines, that in Fig. 1, beyond the expansion valve through the evaporator on to the inlet of the compressor, we have an absolute pressure of 50 psi. Note also that in Fig. 2, from point *D* at the compressor discharge through the hot gas line, condenser, liquid receiver, and liquid line to the expansion valve, we have throughout an absolute pressure of 132 psi. In other words, all of Fig. 1 is the low-pressure portion of the cycle, and all of Fig. 2 is the high-pressure portion of the cycle. For convenience, we call these the low side and the high side, respectively. Are the expansion valve and the



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Subcooling of Refrigerants. We have me
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valve, after which the pressure is only 50 psia. The saturaxpansion valve is
corresponding to 50 psia is 38.3°. On the other hand, the until refrigerant
sure corresponding to 80°F is 98.76 psia. This means tgh pressure and
pressure of the Freon liquid at 80°F is much greater thann as it passes
pressure within the evaporator. Hence when a drop, a cupfu the system.
of liquid Freon enters the evaporator, a portion of it flashes the high and
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tity of Freon flashes to a vapor to cool the remaining Freon d and dis-
temperature corresponding to the absolute pressure within the ev
parts we

have studied in detail in Figs. 1 and 2. In Fig. 3 we have shown a broken line through the compressor and through the expansion valve to indicate the division between the high side and the low side of the system.

In investigating the thermodynamics of a mechanical refrigeration cycle, we have spoken of heat *leaving* or *entering* the refrigerant itself, and we have spoken of this heat as four different heat quantities. The four heat quantities, as shown in Fig. 2, are Q_1, Q_2, Q_3, Q_4. The heat represented by each letter is described at the bottom of Fig. 3, and the points of heat input or removal are indicated on the figure by an arrow for each of the four quantities. If we had a theoretical refrigeration compressor, Q_3 would be zero. The theoretical or ideal compressor assumes that there is no heat loss—no heat flow—from the gas during compression. (Such an ideal compression is called an "adiabatic" compression, meaning a compression where no heat flows into or out of the gas from or to an external source. Remember this definition of adiabatic, as we shall meet the word frequently in later chapters.) Theoretically, an adiabatic compression would be slightly more efficient than our actual compression cycle. However, it would be impossible to insulate a compressor well enough to prevent heat flow to the cooler room air, even if we wanted to. In the second place, we do not want refrigeration compressors to operate on a truly adiabatic cycle, because such a compressor would become extremely hot and lubrication would be too great a problem. If an ordinary compressor of the water-jacket type is operated with no cooling water supply, it is insulated by the stagnant water lying in the water jackets. Such operation of an ordinary compressor will cause damage as serious as the damage caused by operating an automobile engine without cooling water. To be practical from an operating standpoint and to be reasonably simple to design and build, compressors must be operated with fairly good cooling either by air or by water. As stated before, the larger compressors cannot be properly cooled by air alone and must have water-jacket cooling.

Refrigeration Cycles as a Graph. Thus far we have not mentioned the quantity "entropy" in the Steam Tables; we have also omitted this same quantity which is shown in the tables giving the properties of refrigerants. As far as calculations or any other practical value are concerned, we have no use for entropy within the scope of this course. However, there are certain graphs that are made using temperature as the vertical axis and entropy as the horizontal axis. These graphs are called "temperature-entropy charts" or diagrams, and they can be drawn for steam and water or for the vapor-liquid relationship of any of our refrigerants. These charts present a vivid picture of what happens to the refrigerant itself as it travels around through a mechanical-refrigeration system. Temperature-entropy charts for steam are used in analyzing the boilers and engines in steam power plants, and they are also of great importance in designing and building diesel engines, gasoline engines, air compressors, as well as refrigeration compressors. Since the design of such machines is not within the scope of this book, we shall study temperature-entropy charts only so far as such charts are helpful to us in getting a graphic picture of a refrigeration cycle.

Figure 4 shows a temperature-entropy chart in blank form; no refrigeration cycle is drawn on this chart. Of the two heavy curves in Fig. 4, the curve to the left represents liquid at its boiling temperature and is really a graph of the entropy values at various temperatures for a liquid. We must emphasize that this represents a liquid at its boiling temperature

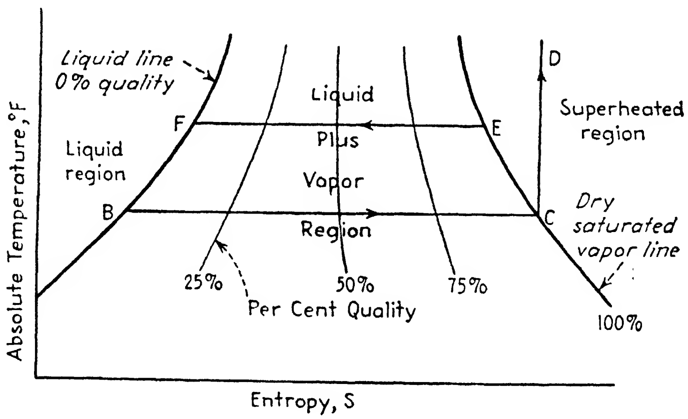


FIG. 4.—Skeleton T-S chart.

such that, if any further heat is added, some of the liquid will become a vapor. The heavy line to the right represents dry saturated vapor at its boiling temperature. In other words, vapor at point *E* is at the same temperature as liquid at point *F*; the difference is that at point *E* the substance contains all its latent heat of vaporization—it is 100 per cent dry saturated vapor, or 100 per cent quality—whereas at point *F* the substance is merely a liquid at its boiling temperature. Lines are sketched in Fig. 4 indicating the substance as a mixture of liquid and vapor at 25, 50, and 75 per cent quality.

Remember that the chart in Fig. 4 does not represent any particular substance—it is a typical chart such as might be drawn for water, Freon, ammonia, etc. This chart gives us the following general picture. The space to the left of the heavy curve marked "liquid line" represents nothing at all for our purpose. The area between the liquid line and the dry saturated-vapor line represents mixtures of liquid and vapor at various qualities. The area to the right of the dry saturated-vapor line represents superheated vapor. Thus the horizontal line *BC* represents complete boiling of a liquid at its boiling point into a dry saturated vapor at the same temperature. Starting at *C* and going back to the left to *B*, the line *CB* represents the condensing of a dry saturated vapor back to a liquid at the same temperature. Line *FE* (or, reading backward, line *EF*) represents this same boiling or condensing process at a higher temperature. Since for every boiling point there is one and only one corresponding pressure, lines *BC* and *FE* are "lines of constant pressure" as well as "lines of constant temperature."

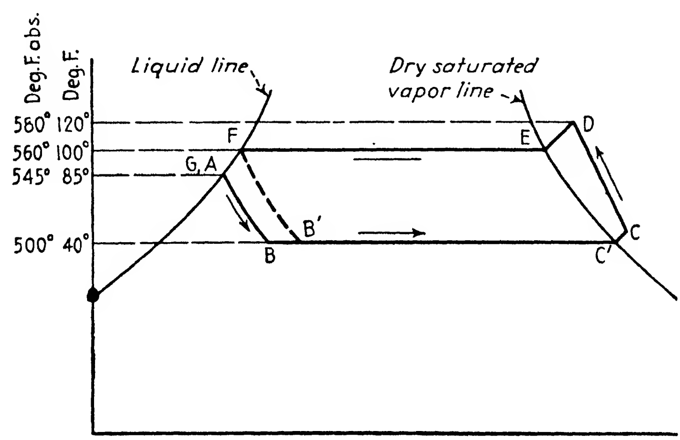


Fig. 5.—Mechanical refrigeration cycle on T-S diagram.

Any vertical lines in the superheated region, such as line *CD*, represent an adiabatic compression, as was described in connection with an insulated compressor cylinder earlier in this chapter. Actually, we always remove heat from our compressor so that our actual compression line is not a vertical line such as the line *CD*. If Fig. 4 is clear to you and if you understand the picture it represents, look at Fig. 5, which shows a refrigeration cycle drawn upon a temperature-entropy chart. Refer to Fig. 3 while studying Fig. 5. The letters indicating various points of the cycle are the same in Figs. 3 and 5.

Start at point *A* (which is the same as point *G*), and trace the refrigerant around the cycle. Line *AB* on the temperature-entropy chart represents a flashing of a portion of the refrigerant into a vapor after it has passed through the expansion valve. In other words, at point *A* you have liquid refrigerant (on the liquid line), while at point *B* perhaps 20 per cent of the refrigerant has flashed to a vapor, the other 80 per cent remaining a liquid. Line *BC* represents the boiling at constant temperature of the refrigerant in the evaporator. The short line *C′C* represents the superheating (a few degrees) of the vapor in the evaporator. Line *CD* represents the actual compression cycle. (Note the difference from line CD in Fig. 4.) At D we then have the refrigerant as a hot gas in the pipe from the compressor to the condenser. Our chart line DE represents the removal of superheat from the refrigerant after the hot gas has entered the condenser. Line EF represents the condensing at constant temperature of the refrigerant throughout the major portion of the condenser. Line FG represents the subcooling of the liquefied refrigerant just before it leaves the condenser. Point G is the same as our point A, so that we have traced the refrigerant all the way around the cycle. We now have the condensed refrigerant subcooled and ready to pass through the expansion valve and start around the cycle once more.

If our condenser were not sufficiently large to provide any subcooling, point A would coincide with point F. Instead of line AB in our cycle, we would then have the line FB. You can see that at point B' more of the refrigerant has evaporated than at point B. In other words, less of the refrigerant *remains to evaporate usefully* in providing refrigerating effect. The latent heat available for useful refrigerating effect would be represented by line BC in the first case or by line $B'C'$. Figure 5 shows very vividly the advantage of some subcooling because there is increased latent heat available for useful refrigerating effect in the amount represented by line BB'.

The vertical distance between the line BC and line FE is a rough qualitative measure of the amount of work the refrigeration machine is being required to do. If the condensing temperature is raised because of warmer cooling water, the line FE would move upward correspondingly. If the evaporator temperature is reduced for any reason, line BC will move downward. Either of these things (raising the condensing temperature or lowering the evaporator temperature) makes it more difficult for the refrigeration machine to produce a given refrigeration capacity. This means one of two things. (1) A reduction in capacity might be suffered if the refrigeration compressor cannot be speeded up and possibly equipped with a larger motor. (2) Even if it is possible to operate the compressor at a higher speed with a more powerful electric motor, the *horsepower required per ton of* refrigeration will be greater.

The effect of condenser and evaporator temperatures on refrigeration machines cannot be overemphasized, both as to machine capacities and power requirements. It is possible to purchase a standard 100-hp factory-assembled refrigeration machine complete except for the low side (*i.e.*, complete except for expansion valve and evaporator). Operating at an evaporating temperature of 55°F with a condensing temperature within the condenser of 95°F, such a machine can produce a refrigerating effect of 140 to 150 tons. This gives a *brake horsepower consumption per ton of refrigeration* of 0.67 to 0.71. On the other hand, the *same refrigeration machine* could be operated at an evaporator temperature of 0°F to produce 15° brine for a fur-storage vault system; if an evaporative condenser were used, the condensing temperature might easily be 115°F in extreme summer weather. Under these conditions, such a machine might produce only a refrigerating effect of 60 tons, which is at a horsepower requirement of 1.67 bhp per ton of refrigeration. Thus you see that *the evaporator temperature and the condensing temperature at which a refrigeration machine is to operate are of vital importance.*

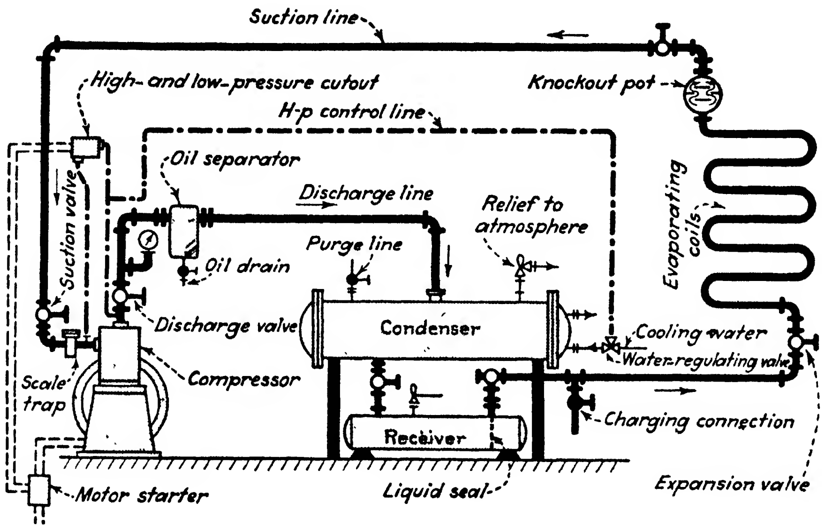


FIG. 6.—Layout of mechanical refrigeration hookup. (*Courtesy of Power.*)

Choice of Refrigeration Machines for Air Conditioning. Most manufacturers complete and publish data concerning what are known as "condensing units." By condensing unit is meant the frame, refrigeration compressor, V-belt drive, electric motor, electric-motor controller, and condenser. In other words, a condesning unit consists of the entire high side of the refrigeration cycle along with the electric motor, drive, and motor controller. Data are published by the manufacturers which show the tonnage or the Btu per hour of refrigerating effect, based upon evaporator temperatures from about 20 to 50°. As to condensing temperatures, the manufacturer's data usually states both the various condensing temperatures and also the quantities of cooling water in gallons per minute at various temperatures from 70 to 90°. Some manufacturers' data give only the cooling water requirement and not the actual condensing temperature that is produced within the condenser.

Evaporators are ordinarily purchased separately from the condensing units, even though they may be purchased from the same manufacturer. If the evaporator is furnished by the same manufacturer who furnishes the condensing unit, the air-conditioning purchaser can buy them both on the same purchase order and thus make one manufacturer clearly responsible for the performance of the two units when operated together. In this case, the manufacturer's representatives will check up on the choice of both the condensing unit and the evaporator. However, when evaporators are purchased from a different supplier, it is necessary that the air-conditioning purchaser select an evaporator that is the proper size for his refrigeration load and that will handle this refrigeration load when supplied with liquid Freon at the maximum temperature at which liquid will be delivered by the condensing unit. It is important not to buy an evaporator that is too big, for this is liable to result in excessive superheat in the suction gas the compressor must draw away from the evaporator.

Figure 6 shows the component parts of a simple refrigeration unit complete with compressor, condenser, receiver, evaporator, piping, and controls.

PROBLEMS

1. Define refrigeration. By what units do we measure refrigeration? What is the Btu equivalent to the unit of refrigeration?

2. If an indirect-expansion (water-chilling) air-conditioning refrigeration installation must cool 300 gpm of water from 54 to 48°, what tonnage must be installed? (One gallon of water weighs 8.33 lb.)

3. The refrigeration compressor performs a vital function in a mechanical refrigeration system. State what it does, dividing your answer in two parts.

4. What is the function of an expansion valve? How does a thermostatic expansion valve operate?

5. A direct-expansion air-conditioning installation is to operate at a 45° evaporator temperature (assume no superheat); its condensing temperature is 95°, and the liquid is subcooled 10°. If Freon is used, how many pounds of refrigerant must be circulated per minute per ton of capacity?

6. We have two Freon evaporators both of 5 tons capacity on air-conditioning units, both operating at a suction temperature of 38°F. (This corresponds to an evaporator pressure of 35.2 psig.) One evaporator operates at 7° of superheat and the other at 17° of superheat; find the difference in the specific volume of the Freon in the suction line from the two evaporators. (Consider 35.2 psig equal to 50 psia.)

7. A flooded evaporator has a volume of ¼ cu ft and is to operate seven-tenths full of liquid Freon, the evaporator temperature being 50°F. The liquid receiver will ordinarily contain liquid at a temperature of 90°F. Applying a 20 per cent safety margin, what must the volume of the liquid receiver be, if it must be able to contain all the refrigerant in the system? (Safety margin allows for pipes, etc.)

8. Assume that you have four 10-ton refrigeration machines using different refrigerants, each operating at 40°F evaporator temperature and 100°F condensing temperature. The refrigerants are Freon 12, ammonia, methyl chloride, and sulphur dioxide. Find for each machine the *cubic feet* per minute (cfm) of refrigerant in the suction line. (To do this you must first find the pounds of refrigerant per minute; assume no superheat or subcooling.) (Tables are used as in Probs. 1 to 4 of Chap. IX.)

9. If the suction-line velocity may be 1,500 ft per min., and the liquid-line velocity is permitted to be 200 ft per min., what is the minimum inside diameter (*in inches*) of the suction and liquid lines of the 10-ton Freon machine in Prob. 9? Find the pipe area first in square *feet*, then change to square inches. These formulas are of value:

$$\text{Area of circle} = A = \frac{\pi d^2}{4} \qquad d = \text{diameter of circle}$$

$$\text{cfm} = Q = AV \qquad V = \text{velocity}$$

10. From your work in Probs. 8 and 9, which of the four machines in Prob. 8 would need the largest suction line? Which would need the largest compressor displacement?

CHAPTER XI

MECHANICAL REFRIGERATION EQUIPMENT

We are now familiar with refrigeration machines in general and with the refrigeration cycle, regardless of the refrigerant used. In previous chapters we have completed this study, commencing with the study of refrigeration machinery using ordinary water as the refrigerant. In this chapter we wish to discuss briefly certain phases of domestic refrigeration, different types of expansion valves, absorption refrigeration, and finally the Electrolux refrigeration cycle. The latter is a type of absorption refrigeration, and although it is not used in air conditioning, it is an interesting application of Dalton's law, and all air-conditioning men should understand it.

Domestic Refrigeration Compressors. Literally millions of domestic refrigeration compressors have been manufactured, and the majority of them are still in service today. These machines represent the smallest practical refrigeration machines that are manufactured and used. However, they were not made available commercially until many years after large ammonia ice machines had gained wide acceptance and use.

The early domestic refrigeration machines, and many of those manufactured today, use reciprocating compressors. As you know, this type of compressor consists of a cylinder containing a piston that moves back and forth inside the cylinder. Naturally there are many design modifications, including different piston designs and valve arrangements, as well as different bearings, seals, crankshafts, means of cooling, methods of lubricating, etc. Fundamentally, however, all reciprocating compressors are the same, and mechanically they all resemble an automobile engine or air compressor.

In air-conditioning refrigeration, all but a few of the small portable unit coolers use reciprocating compressors. The tendency in this field is toward lighter, higher speed multicylinder compressors as contrasted with the one- and two-cylinder heavy-duty slow-speed machines that were orginally used. One disadvantage of any reciprocating machine is the unavoidable vibration and wear and tear due to the fact that relatively heavy metal parts are flying *back and forth* (or have a "reciprocating" motion). Electric motors and steam turbines *spin in pure rotation.* For these reasons, it has been possible to design high-speed efficient and light equipment, in all sizes from fractional horsepower up to thousands of horsepower, in the turbine field.

In domestic refrigeration we have two types of compressors that are of the positive-displacement type and that are nevertheless *rotary* rather than *reciprocating*. It is possible that air-conditioning compressors may come to use some type of rotary positive-displacement compressor instead of the reciprocating machines now in general use, for those refrigerants requiring positive-displacement compressors. (We have already seen that water vapor and Carrene may be used as refrigerants with a centrifugal com-

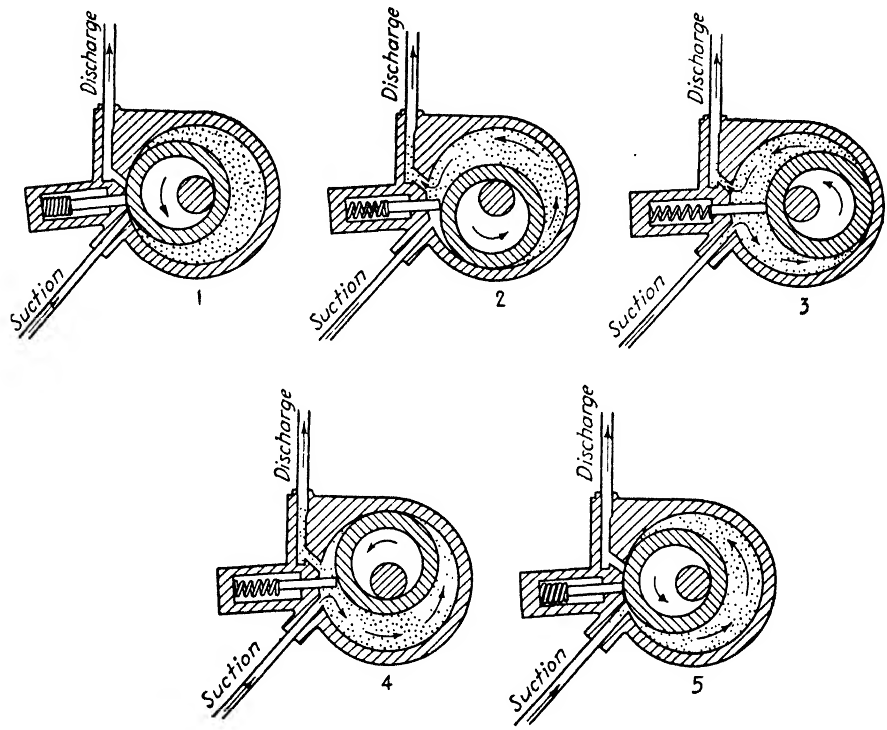


FIG. 1.—Cycles of Norge Rollator compressor. (*Courtesy of Norge.*)

pressor that is not of the positive-displacement type.) It is advisable, therefore, that we examine briefly and become familiar with one or two types of rotary compressors that have been successfully used in domestic refrigeration.

Figure 1 illustrates the cycle of the Norge Rollator compressor. Note that all figures presented in this chapter (except those regarding Electrolux refrigerator) are simply sketches intended to show "*how* it works." These figures do not illustrate exactly the manner in which the various devices are mechanically constructed and assembled. Looking at Fig. 1, we see five numbered sketches; the first shows a Rollator compressor about to begin a stroke. Note that we do not say whether this is an intake (suction) stroke or a compression (discharge) stroke. The reason is that the Norge compressor, in all other positions except position 1, is engaged in *both* suction and discharge strokes at the same time. The sliding wedge, shown in all sketches and indicated in sketch 1, divides the high-pressure and low-pressure operations within the compressor.

In sketch 2, note the open space below a sliding wedge and between the thick outer portion of the compressor and the rotating inner member. This small space connects directly with the suction line. In sketch 2 note the much larger crescent space above the sliding wedge and between the thick outer portion of the compressor and the inner rotating member. This space connects with the discharge line from the compressor to the condenser. Sketch 3 shows the compressor after the rotating member has turned a little over one-quarter of a turn beyond the position shown in sketch 2. The suction chamber is greatly increased in volume, as shown in sketch 3, thus drawing gas in from the evaporator. The area above the sliding wedge is greatly *decreased* in volume, which means that the gas in this chamber has been squeezed or compressed. Sketch 4 shows the compressor after the rotating part is turned a little over one-quarter of a turn further. The volume above the sliding wedge has become almost zero, which means that the gas in this space has been compressed and most of it has been forced out the discharge line. The suction chamber in sketch 4 is increased until almost the entire space inside the compressor has become filled with low-pressure gas from the evaporator. Let us call the stroke of this compressor, which we have just discussed, "stroke 1." We can then say that sketch 5 shows the compressor about to start a second stroke. During this second stroke the compressor will discharge the gas that was drawn in during stroke 1, and it will draw in a second volume of gas (from the evaporator) which will, in turn, be discharged during stroke 3.

Figure 2 illustrates a type of rotary compressor that has been used as an oil pump, an air compressor, and also as a refrigeration compressor. Such "vane" pumps, as this type of rotary compressor is sometimes described, have been built with one to six and eight vanes. A four-vane compressor, substantially as illustrated in Fig. 2, is used in most Coldspot domestic refrigerators. These machines are manufactured by the Sunbeam Electric Company and retailed by Sears, Roebuck and Company. The compressor design represents clever mechanical engineering and results in one of the best possible rotary compressors from the standpoint of vibrationless operation, simplicity, and freedom from wear.

The Coldspot rotary compressor is similar in one operating characteristic to the Rollator type just discussed—it also draws in and discharges the refrigerant gas *continuously*. In other words, unlike a reciprocating compressor, it does not have separate suction and discharge strokes, but instead is always engaged in *both* of these operations *at the same time*. As illustrated in Fig. 2, this compressor rotates counterclockwise. Each of the four vanes may slide in or out toward or away from the center of the rotor. Some vane pumps or compressors of this general design use light springs behind the vanes so that they are held against the surface of the outer chamber when the compressor is not operating. Other types use no springs at all. In either case, when the machines are operating, the vanes are held out by centrifugal force and create, in the presence of an oil film, a tight seal against the outer metal surface. The center of the rotor is at a different point than the center of the circular chamber in the body of the compressor.

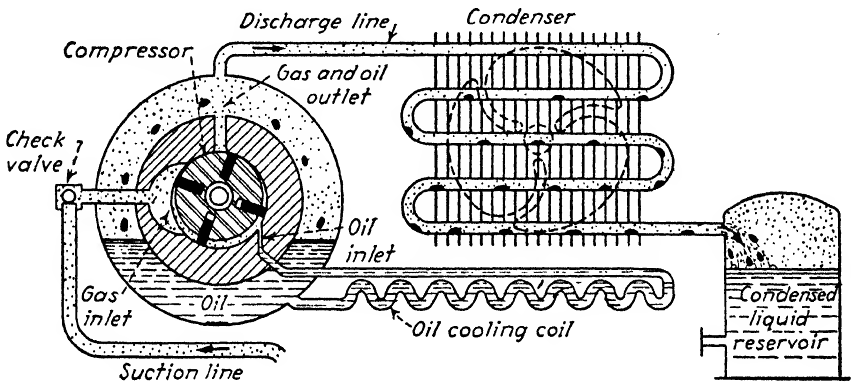


FIG. 2.—Diagram of rotary-type compressor and high side.

As the rotor turns counterclockwise, the space between two vanes, in the chamber between the rotor and the fixed portion, becomes *smaller* as the *vanes travel from the inlet port toward the discharge port.* Hence gas is carried away continuously from the inlet port and squeezed out the discharge port. The check valve is inserted to prevent the leakage of high-side gas back through the compressor to the evaporator when the compressor is at rest.

Expansion Valves. Thus far we have called attention to the fact that some sort of expansion valve must be located between the liquid receiver and the evaporator of any mechanical-refrigeration system. We now know that the expansion valve fulfills several functions. It provides the barrier making possible the pressure drop from the high side to the low side. (In absolute pressure, the high side is ordinarily at a pressure two to four times as high as the low side.) As a part of this function, an expansion valve meters or controls the flow of refrigerant into the evaporator. Obviously if there were no expansion valve, refrigerant would rush into the evaporator until the pressure in the evaporator and the pressure in the condenser become substantially the same. As a further corollary to this function of dividing the high side from the low side, the expansion valve may act to control the pressure that exists in the evaporator. This is readily understood by realizing that if the expansion valve were tightly closed it would be possible for the compressor to create a high vacuum within the evaporator. In fact a hand valve is provided in the liquid line which, when closed, permits this very operation, thus removing all refrigerant from the low side of the system and delivering it to the condenser and liquid receiver. This operation is known as "pumping down" the system.

There are various types of expansion valves available to fulfill the functions discussed above. The types operate differently, have different advantages and disadvantages, and may or may not be adjustable. Because some of these expansion valves become most highly perfected in domestic refrigeration applications, we have deferred their study until this time.

The low-side float-type expansion valve can be used only with flooded or semiflooded evaporators. Such an evaporator is always full or partly full of boiling liquid refrigerant. (See Fig. 1, Chap. X, for an illustration of this type of evaporator, although the expansion valve shown is not a lowside float.) Figure 3 illustrates schematically a low-side float expansion valve. When the liquid level in the evaporator drops because of the boiling away of some of the refrigerant, the float drops and opens the expansion valve to admit additional liquid refirgerant. Just as soon as the level rises sufficiently, the rising float closes the expansion valve. This is all there is to the operation of this simple type of expansion valve Actually, when a refrigeration machine is operating steadily, the float may remain in practically the same position, this position being such that the expansion valve is open slightly, thus admitting liquid refrigerant at the rate at which the refrigerant is being removed from the evaporator as a gas.

From the above discussion you see that a low-side float expansion valve must have its float chamber on the cold low side of the system. This valve acts to maintain a constant level of refrigerant in the evaporator, provided there is a sufficient charge of refrigerant in the system. What would happen if leakage greatly reduced the charge of refrigerant in the system? From the preceding paragraph we saw that the float drops and opens the expansion valve when more liquid refrigerant is needed in the evaporator. If we do not have sufficient liquid refrigerant in the liquid receiver to flow into the evaporator and raise the float, the float will stay down. This means that the expansion valve will stay open. The result is a failure to get the desired refrigerating effect from the machine. If insufficient refrigerant is the cause and the system has a low-side float expansion valve, the head pressure and suction pressure will be close together instead of showing their expected difference. Since the evaporator pressure is higher than usual, the evaporator temperature will be high. This trouble can be corrected by finding and repairing the leak that caused the loss of refrigerant, then replenishing the charge of refrigerant.

There is another float-type expansion valve known as the high-side float expansion valve. This is illustrated in Fig. 4. Compare Fig. 4 with Fig. 3. The most important difference is in the location of the float that operates

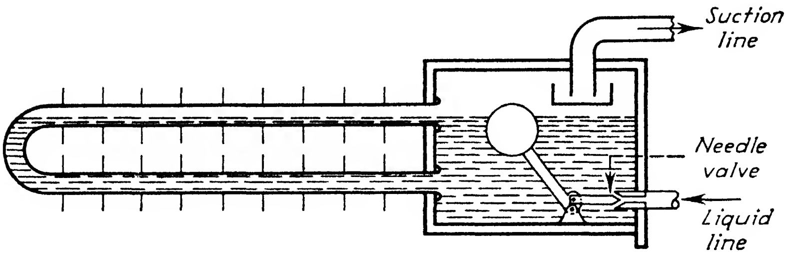


FIG. 3.—Low-side float expansion valve.

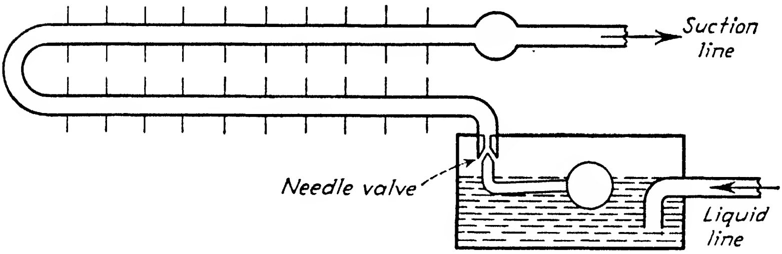


FIG. 4.—High-side float expansion valve.

the needle valve. The low-side float is in the evaporator, or possibly in a small chamber just outside the evaporator—it is definitely on the cold low side of the system. The high-side float, on the other hand, is in a pool of warm liquid refrigerant that has not yet passed through the expansion valve. The difference in these locations brings up an interesting comparison. If a certain evaporator is to operate three-quarters flooded, its capacity of liquid refrigerant at this level being 10 lb, a low-side float can, of course, be used with a liquid receiver having a capacity of 20 lb of liquid refrigerant. It would be possible to charge the system with 15 to 18 lb of refrigerant, which would provide a reserve of 5 to 8 lb of refrigerant to take care of minute leaks such as leakage through the seal. Until leakage had reduced the charge to 10 lb of refrigerant, no operating trouble would be encountered. With a high-side float, on the other hand, any warm liquid refrigerant reaching the float chamber is *immediately admitted* to the evaporator. Therefore, in the case of the evaporator discussed above, it would be necessary to charge the system *carefully* with 10 to 11 lb of refrigerant. In normal operation with a high-side float expansion valve, *all this refrigerant* would be *in the evaporator* in a liquid form, except for the small amount of gas in the suction and discharge lines and in the condenser.

Under these conditions, any leakage means that the evaporator will not be sufficiently flooded and the rated heat transfer will not be obtained. It is impossible to provide a charge appreciably greater than the amount of refrigerant desired in the evaporator, because any additional refrigerant admitted would go at once into the evaporator. For certain refrigeration applications in commercial refrigeration, and some air-conditioning installations where chilled water is produced in a central plant, it is desired to

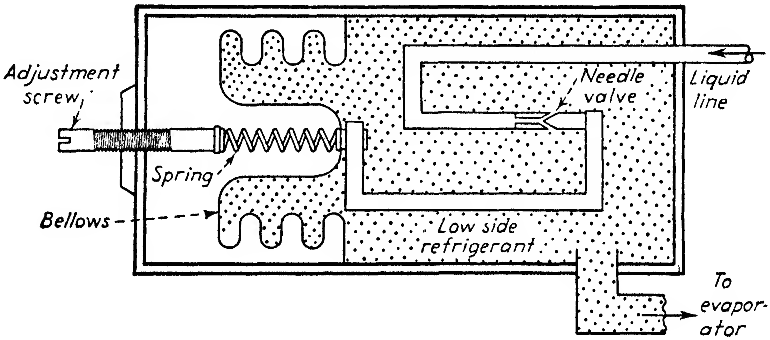


FIG. 5.—Automatic expansion valve.

maintain the evaporator at a constant temperature. This, of course, means maintaining the evaporator at a pressure as nearly constant as possible. The two float-type expansion valves previously discussed do not offer any means of regulating the amount of refrigerant admitted to the evaporator at a rate that will *control the pressure* (and *temperature*) in the evaporator. A valve known as the "automatic expansion valve," illustrated in Fig. 5, will provide a constant evaporator pressure and has advantages so long as the refrigerating load on the machine *does not vary too greatly* from time to time. Referring to Fig. 5, this type of expansion valve consists essentially of the familiar needle valve that is operated by an arm connected to a spring and a pressure bellows. Note that the pressure bellows are filled with low-side refrigerant and hence react to evaporator pressure. An increase in evaporator pressure tends to stretch the bellows, the way camera bellows are stretched when a folding camera is opened. If the bellows stretch in this direction, the arm and valve needle move to the left, tending to close the valve. If the bellows pressure decreases, the spring can push the arm and the needle to the right, thus tending to open the valve.

Let us suppose that this valve is closed and that the machine is in operation; the compressor will draw off some gas from the evaporator, causing a slight decrease in the pressure (and temperature) within the evaporator. This pressure decrease is reflected or "felt" by the bellows in the valve mechanism. The bellows then exert a slightly weaker force to the left, and the spring moves the arm and the valve naturally to the right. This opens the valve, admitting additional refrigerant to the evaporator, increasing the pressure therein. This increasing pressure, acting through the bellows, tends to overcome the force of the spring and move the needle to the left, which tends to close the valve. As in the case of the float-type expansion valve, the automatic expansion valve during steady operation of the machine does not make a complete cycle from fully open to tightly closed. Instead it modulates or "floats" at a partly opened position, closing a little or opening a little under the influence of the spring and pressure bellows. From the foregoing discussion you can see that the automatic expansion valve is responsive to one thing only, *i.e.*, evaporator pressure. By means of a thumbscrew or adjustment dial, the spring tension can be varied manually so that any desired pressure can be maintained in the evaporator. This type of valve is not suitable for use with flooded evaporators in air-conditioning work, because the load varies a great deal and the valve takes no cognizance of the liquid level in the evaporator. It is not used in the domestic field.

Automatic expansion valves are simple and do not get out of order easily, their main advantage being that where loads do not vary too greatly they make it possible to operate with a constant evaporator temperature. Their use becomes disadvantageous when the load fluctuates greatly. Suppose that a water-chilling evaporator designed to cool water from 55 to 45° is subjected to 70° water for appreciable periods of time. With this warm water, increased quantities of heat will flow into the constant-temperature evaporator because of the increased temperature difference. This will tend to boil away the refrigerant more rapidly, which will register an increased evaporator pressure on the bellows of the valve. This pressure causes the valve to move toward the closed position, decreasing the rate at which refrigerant is admitted to the evaporator. In this case, with fewer pounds of refrigerant being admitted per minute to the evaporator, under conditions of high Btu input to the evaporator, the refrigerant vapor leaving the evaporator will be highly superheated. Under the opposite conditions, of a lower-than-rated load, the automatic expansion valve would tend to move toward a more fully opened position because of the slightly lower evaporator pressure. This results in the paradox of admitting increased quantities of refrigerant to the evaporator under conditions of reduced load. The compressor would then operate continuously and waste power unless a separate thermostatically controlled switch were provided to shut off the compressor.

There is a modification of the automatic expansion valve that consists of the same mechanism with the addition of another pair of bellows on the shaft that operates the arm and needle. This type of valve is called a "thermostatic expansion valve," and it has entirely different operating

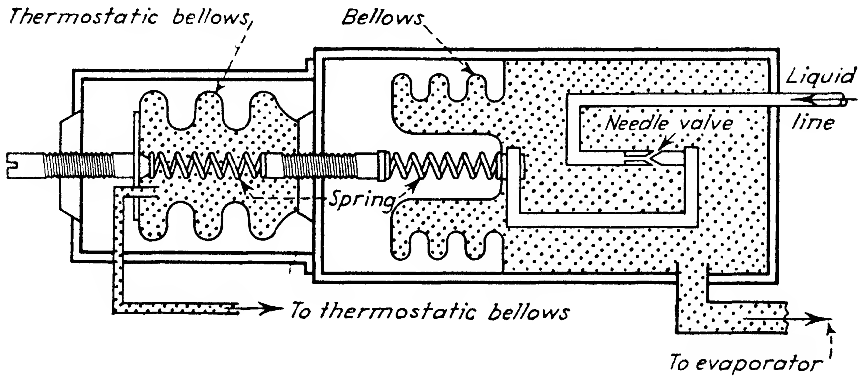


FIG. 6.—Thermostatic expansion valve diagram.

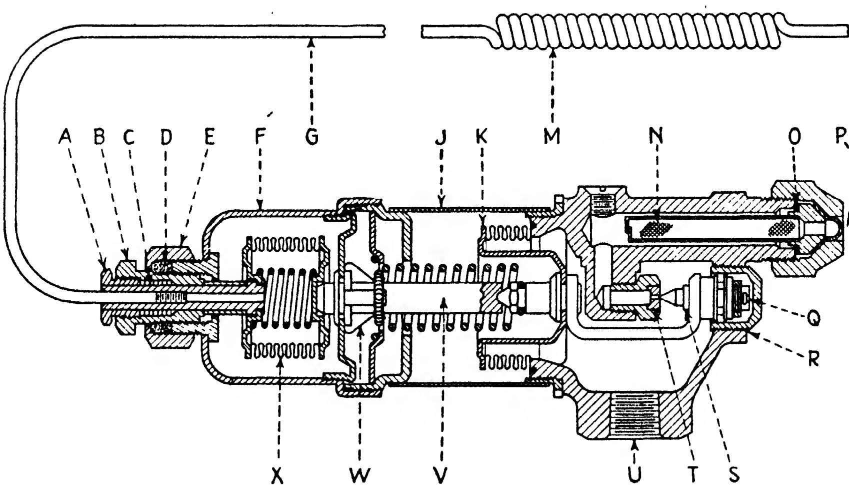


FIG. 7.—Mechanical arrangement of thermostatic expansion valve. *(Detroit Lubricator Co.)*

characteristics from the automatic expansion valve. It is illustrated schematically in Fig. 6, the actual construction being shown in Fig. 7. To help you trace the operation of the thermostatic expansion valve, we suggest that you mark (in Fig. 7) the needle and entire moving arm and shaft in color. This extends from the bellows *X* to the needle *S* and the needle swivel *Q*, including the entire shaft *V*. By shading this, you will thoroughly understand the extent of the moving part within this valve body and case. In this type of valve there is a bellows and spring similar to that found in the automatic expansion valve, the bellows being sensitive to the pressure in the evaporator. In addition, the thermostatic bellows (X in Fig. 7), referred to as the "power-element" bellows, acts upon the moving shaft. In the case of the thermostatic expansion valve, the thermostatic bellows applies the force to the shaft, which gives ultimate control to the motion of the shaft. (The thermostatic expansion valve illustrated in Fig. 7 is of the adjustable type, having an adjustment to regulate the spring tension manually and, hence, the operating characteristics.)

The movement of the shaft is under the control of the spring tension and the bellows that react to the evaporator pressure, the same as in the case of the automatic expansion valve. However, neither of these factors causes the ultimate operation of the valve. Remember that the evaporator pressure is a measure of the evaporator temperature also. The thermostatic element connected to the thermostatic bellows is located adjacent to the suction line so that this element senses the *temperature* of the suction line. The force exerted by the power bellows (or the thermostatic bellows) is such that the valve will be at midway open position *when the temperature of the suction line is a fixed number of degrees above the evaporator temperature corresponding to the existing evaporator pressure.* In other words, evaporator temperature and pressure may fall considerably as the load varies, using this type of expansion valve. Let us consider two concrete examples to understand fully how this valve operates. Suppose that the heat load on an evaporator is such that the evaporator pressure stabilizes at 37 psi (corresponding temperature 40°). After the compressor has been in operation for 5 or 10 min, the expansion valve would maintain an opening such that the vapor in the suction line would remain at a fixed temperature. It is common for this valve to operate at a setting such that the vapor leaving the evaporator is superheated 5°; this would mean that, in the case we are discussing, vapor in the suction line would be at a temperature of 40° + 5°, or 45°. Suppose now that the heat load on the evaporator is greatly increased, the compressor, of course, remaining of constant displacement and, hence, able to remove the same volume of refrigerant vapor per minute from the evaporator. Because of the increased heat input to the evaporator, the refrigerant would boil more rapidly, giving off an increased quantity of vapor. Pressure in the evaporator would rise until the compressor was drawing off gas at the same rate at which the gas was formed in the evaporator.

The valves shown in Fig. 6 are not actually constructed as illustrated in the sketch. The actual construction of a popular valve of this type, as manufactured by the Detroit Lubricator Company, is shown in Fig. 7 by courtesy of the manufacturer.

Let us suppose that conditions balance out at an evaporator pressure of

TABLE 1.—COMPARISON OF EXPANSION VALVES

	Low-side float	High-side float	Automatic	Thermostatic	Capillary tube
1. Sensitive to what?	Liquid *level* in evaporator	Liquid *level* in liquid receiver (or special small chamber beyond receiver)	Evaporator *pressure* (hence evaporator temperature)	Difference between evaporator temperature and suction gas temperature, *i.e.*, the degrees of superheat in the suction gas	*Pressure difference* between high side and low side
2. Reacts how?	Admits more refrigerant on a drop in level	Admits all refrigerant that reaches float chamber. High side stays dry	Admits more refrigerant on a drop in evaporator temperature. Undesirable	Admits more refrigerant on a rise in superheat	Admits slightly more reirigerant on a drop in evaporator pressure or on a rise in condenser pressure
3. Best application	Flooded evaporators, domestic type, to large water chillers in central plant air conditioning	Same as low-side float type. System then must have exactly correct charge with no excess or shortage. Used as oil-return tray	Evaporators where constant temperature is important and where load does not vary much. Little used, as most loads vary greatly	All dry expansion evaporators, all sizes, air conditioning and commercial where load varies. Thus used instead of plain automatic type	Individual small evaporators where load does not change too much. For individual tubes on some direct-expansion finned coils in air conditioning
4. Limitations	Only good for partly or fully flooded evaporators. No control of suction superheat or evaporator pressure	Same as low-side type. Also, excess charge means liquid in suction lines; reduced charge cuts down capacity proportionately and increases suction superheat	Not good for varying loads on one evaporator, or for multiple evaporators with widely differing loads. Rarely used in air conditioning	Most widely used on ordinary air-conditioning installations, and much commercial refrigeration. Delicate mechanism and can be damaged. Requires maintenance and perhaps some adjustment	Only used when supplied with evaporator coil by the manufacturer, and in some package units factory assembled and charged. Equalizes evaporator and condenser pressure at once on shutdown
5. Adjustment	None, once installed	None, once installed	Available with and without manual adjustment	Available with and without manual adjustment	None—tube length, bends, and inside diameter determines flow

46.7 psig (corresponding temperature 50°) and that the valve is so arranged that it would automatically increase its opening, admitting refrigerant at a faster rate such that the temperature of the refrigerant in the suction line would be held at about 55°. In other words, the vapor leaving the evaporator would have a constant superheat, regardless of varying load conditions which would cause varying evaporator pressures. Note carefully how this differs from the operation of the automatic expansion valve, which, as we saw on page 184, decreases the supply of refrigerant with an increase in load and permits the vapor leaving the evaporator to become highly superheated.

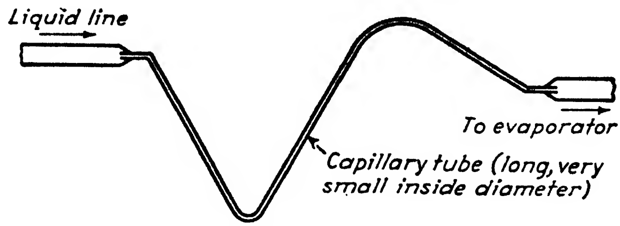


FIG. 8.—Capillary tube.

We wish to consider one further type of expansion valve, which is not really a "valve" at all. This is called a "capillary tube," meaning a tube the inside diameter of which is very small. Capillary tubes have been used considerably in domestic and commercial refrigeration. Although their use for these purposes is not increasing today, the knowledge gained through this application has brought about an increasing use of the capillary tube as an expansion valve in certain special air-conditioning applications. Figure 8 illustrates the use of a capillary tube as an expansion valve in the case of a domestic-refrigeration evaporator.

The capillary tube, because of its small inside diameter, has a great resistance to flow through the tube. By using 5 to 20 ft of such tubing, it is possible to secure any desired rate of refrigerant flow with a certain pressure difference (perhaps an 80-lb difference for Freon) between the high side and low side. The flow will depend upon this pressure difference, the flow being greater if the pressure difference increases, a decrease in flow occurring if the pressure difference decreases. There is no possible adjustment of this type of expansion valve, nor can it be made responsive to thermostatic control in the ordinary sense. However, if the evaporator pressure decreases because of a shortage of refrigerant in the evaporator, the pressure difference between the high and low side *increases*, so that the refrigerant flows to the evaporator at a slightly faster rate.

Difficulty has always been experienced in distributing refrigerant equally to the many tubes of a direct-expansion finned coil. Figure 9 illustrates a means of using capillary tubes to provide the pressure drop between high and low side and at the same time to distribute refrigerant equally to all tubes of a direct-expansion coil. Previous methods have been based upon having the tubes of the coil connected to a header using an automatic, a thermostatic, or even a float-type expansion valve to admit refrigerant to the header. Under these conditions, the entire header is part of the low side. Once the refrigerant enters the header, it naturally takes the easiest course through the coil. In spite of ingenious baffling inside the header, this frequently results in some tubes receiving large quantities of refrigerant and other tubes being virtually starved.

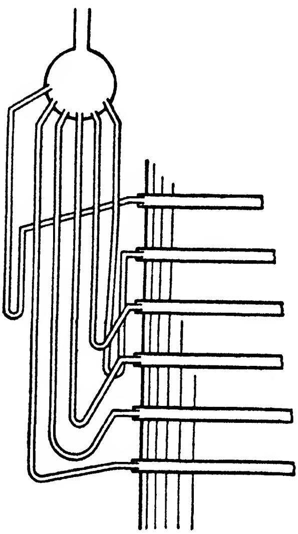


FIG. 9.—Distribution header at evaporator.

Figure 9 illustrates schematically a direct-expansion coil in which six tubes are fed liquid refrigerant by means of capillary tubes. The high-side warm liquid enters a plenum chamber out of which six capillary tubes run, one leading to each tube. Since it is a short distance from the plenum chamber to the top tube, why do we not use a straight length of capillary tube just a few inches long? If this were done, the resistance through this short length capillary tube would not be so great as the resistance through the capillaries leading to the other tubes. Hence, with the same pressure difference between the high side and the low side, a different quantity of refrigerant would flow through each different length of capillary tube, and we should not achieve our objective of equal distribution of refrigerant through all tubes. Instead, *both* the capillary tube *length* and the *shape of the bend* are carefully calculated so that the *resistance to flow* is the same for *all* the capillary tubes. This means that actually the same quantity of refrigerant will flow through each of the capillary tubes.

Some manufacturers have built ordinary direct-expansion air-conditioning coils using capillary tubes in the manner just described, as illustrated schematically in Fig. 9. Such an application, however, although it does achieve the objective of distributing refrigerant equally to all tubes, is also subject to the disadvantages of limitations of capillary-tube expansion valves as summarized in Table 1. As indicated in item 4 of Table 1, capillary-tube expansion valves cause the pressure in the evaporator and condenser to equalize almost instantly when the compressor shuts off. This is undesirable, especially in the case of air-conditioning compressors that may stop and start rather frequently under the influence of automatic controls. Furthermore, Table 1 and our previous discussion shows that capillary-tube expansion valves are not desirable when the load on the evaporator may vary widely.

However, it seems desirable to use modified capillary tubes, *i.e.*, tubes having a larger inside diameter than true capillary tubes. The resistance to flow of these larger diameter tubes is not great enough to provide the needed pressure drop from the high side to the low side. Therefore, a thermostatic expansion valve is placed in the liquid line just before the liquid enters the plenum chamber out of which the tubes lead. With this arrangement, most of the pressure drop occurs across the thermostatic expansion valve, the modified (larger) capillary tubes providing the remaining pressure drop. Capillary tubes must be carefully chosen by the coil manufacturer for both length and shape of bends, under which condition the refrigerant is distributed equally to all tubes of the coil. This arrangement results in ideal conditions for air-conditioning work, providing all the advantages of the thermostatic expansion valve, coupled with positive assurance that no tube will be flooded while other tubes are starved. Figure 11 shows a close-up of the actual construction of the header in a new Trane direct-expansion coil using the modified capillary tubes in the manner just described.

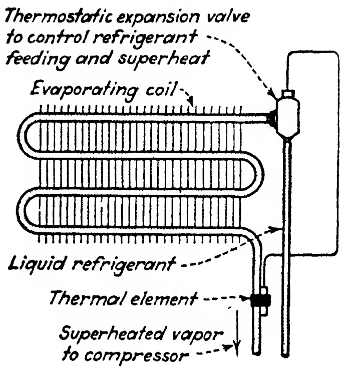


FIG. 10.—Thermostatic valve hookup.

Summary. Expansion Valves. First we discussed the high-side and low-side float-type expansion valve. We then discussed automatic expansion valves, thermostatic expansion valves, and finally capillary-tube expansion valves. We have summarized the high lights of the foregoing discussion in Table 1. Note that none of the expansion valves discussed provides anywhere nearly *all* the control that is required by the mechanical-refrigeration system. Probably a thermostat, humidistat, dual-pressure control, or a combination of these devices may act to stop and start the compressor motor. This function is separate and distinct from the function of an expansion valve. We shall discuss it later when we take up controls in general.

Absorption Refrigeration. As we have previously stated, it is true that absorption refrigeration is rarely, if ever, used in air-conditioning work. Nevertheless absorption refrigeration is widely used industrially, and the Electrolux refrigerator provides a unique example of the importance of absorption refrigeration in small units. Since air-conditioning work involves so much use and knowledge of refrigeration in various forms, an air-conditioning man cannot afford to be ignorant of absorption refrigeration. Therefore, we are going to study absorption refrigeration briefly.

Absorption refrigeration involves producing refrigerating effect in an evaporator without the use of a mechanical compressor. To do this by means of a machine that operates in *intermittant cycles* is easy, and we shall

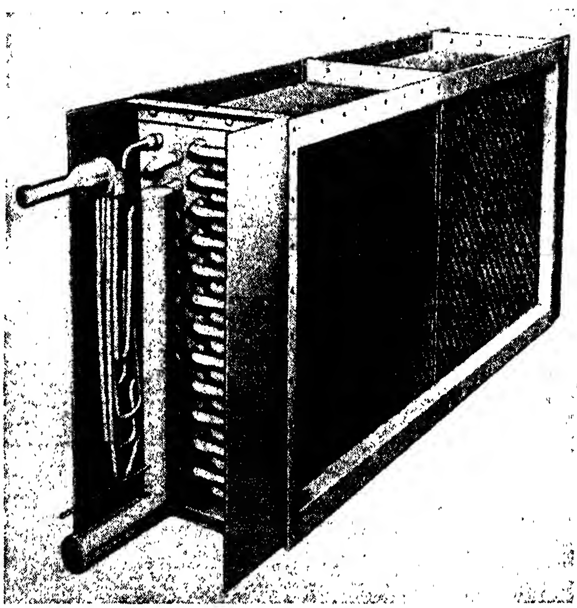


FIG. 11.—Direct expansion evaporator. (*The Trane Co.*)

examine this method first. We shall then consider continuous-cycle absorption refrigeration, of which the Electrolux refrigerator is the most unusual example. The unique feature of the Electrolux is that there are no moving parts whatsoever—not even a fan or circulating pump.

Ammonia is the most commonly used refrigerant in abosrption refrigeration. Ordinary water will abosrb large quantities of ammonia vapor if the water is kept reasonably cool—say, 90 to 100°. In fact water can absorb ammonia vapor so rapidly that it is possible to build a water "absorber" that will draw off ammonia vapor from an evaporator just as effectively as a mechanical compressor draws off this vapor. Suppose that we had a tank of pure liquidified ammonia under high pressure such as might be found in the liquid receiver of a mechanical refrigeration machine or in one of the steel cylinders used to charge refrigeration machines. It would then be possible to feed this high-side liquid ammonia through an expansion valve and into an evaporator. Such an arrangement is illustrated in Fig. 12.

The equipment shown in Fig. 12 would provide refrigeration only until the liquid high-side ammonia became exhausted or until the water in the absorber tank become so saturated that it could absorb no more ammonia, whichever occurred first. By proper design of the systems, the size of the ammonia tank and absorber can be chosen so that the absorber is not quite full when the ammonia tank becomes empty. Refrigeration systems, as described above and illustrated in Fig. 12, have been and are being used commercially to refrigerate trucks. Every day in summer, and every three

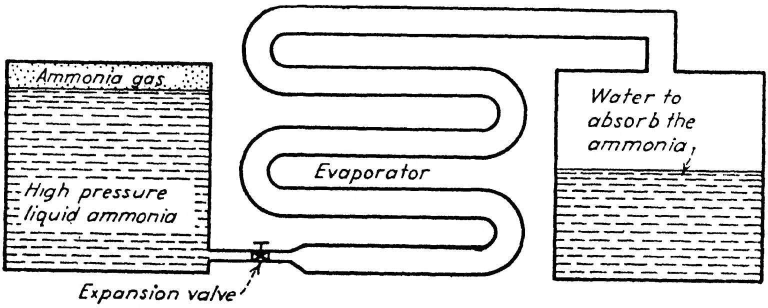


FIG. 12.—Principle of absorption refrigeration system.

or four days in winter, the trucks go to a service station. Here the ammonia tank is filled with ammonia and the absorbers are drained and refilled with pure water. The ammonia contained in the solutions drained from the absorbers is reclaimed and used over again.

Obviously, the absorption system discussed above does not provide *continuous* refrigeration because of the servicing operation. It is possible, however, to build a plant incorporating the device that reclaims the ammonia from the absorber water. Such a plant operates continuously and requires an ordinary condenser, liquid receiver, expansion valve, and evaporator, just like any mechanical system. A continuously operating ammonia absorption system is illustrated in Fig. 13. The absorber, ammonia-solution pump, and generator are shown to the left. It is easy to compare this system to ordinary mechanical systems by noting that the compressor shown at the left in Fig. 13 can replace the generator pump and absorber. The same condenser, receiver, expansion valve, and evaporator can be used.

In systems such as the one illustrated in Fig. 13, the generator removes most of the ammonia from the water, but not quite *all* the ammonia. Therefore, a *weak water solution of ammonia flows* by gravity from the generator to the absorber—this solution might be 98 per cent water and 2 per cent ammonia. In the absorber, ammonia is absorbed into the water until the solution is at a concentration of about 30 per cent ammonia. The pump is required to pump this strong water solution of ammonia up to the generator because the *generator* operates at a *high pressure* while the *absorber operates at* substantially *evaporator* (*low-side*) *pressure*. The weak solution of ammonia is called "weak aqua"; the strong solution is called "strong aqua." The pump is referred to as the "strong aqua pump."

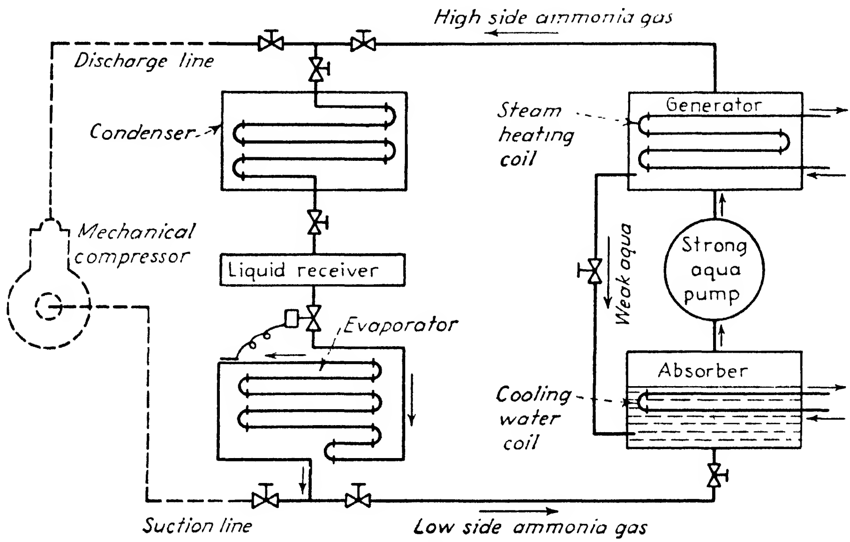


FIG. 13.—**Ammonia absorption system.**

Figures 14 and 15 illustrate the cycle of the Electrolux continuous-absorption refrigeration cycle. Figure 14 shows the piping from the outside, the various lines being marked to show the substances flowing within. Figure 15 shows the same thing in a cut-away view. Let us describe the operating cycle, starting from the generator. Keep in mind the functions of the generator and absorber in connection with absorption refrigeration in general.

The ammonia vapor is driven off from the "strong aqua" in the generator by means of the same principle that lifts coffee to the top of a coffee percolator. The weak aqua and the ammonia vapor rise up the line marked "liquid lift," leading into a "separator." As you can see in Fig. 15, the ammonia vapor travels *upward* from the separator, while the weak aqua drops in the separator, and runs by gravity back to the absorber.

The ammonia vapor traveling to the condenser is similar to the high-pressure gas discharged from a mechanical-refrigeration compressor. The condenser removes the latent heat from this vapor, condensing it. The liquid flows (through a trap fitting) into the evaporator. Here in the evaporator we discover the first difference from the evaporator of a mechanical system or the evaporator of an *ordinary* absorption system. *The total pressure within the evaporator is the same as the total pressure in the condenser. The total pressure throughout this system is the same.*

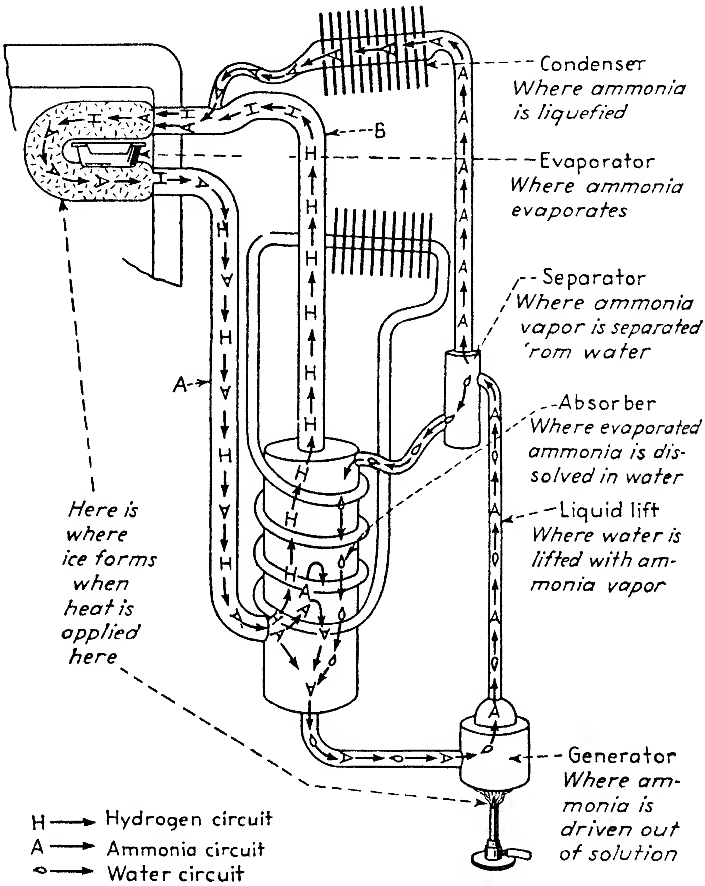


FIG. 14.—Electrolux continuous-absorption refrigeration cycle. *(Servel, Incorporated.)*

Hydrogen gas also enters the evaporator, from pipe *B*. Never mind how this hydrogen gas gets here; you will learn this in the next paragraph. The quantity of hydrogen gas is so proportioned that the partial pressure of the ammonia within the evaporator will be only a small portion of the total pressure. Perhaps the partial pressure of the ammonia might be 53 psig, which would cause the ammonia to have a boiling temperature of 10°F (see Table XI in the Appendix). This is a practical application of Dalton's law. The liquid ammonia entering the evaporator is affected, with regard to boiling temperature, *only by its own partial pressure.* Since the total pressure throughout this system is the same, it is obvious that according to Dalton's law the total pressure within the evaporator is the sum of the partial pressures of the hydrogen and of the ammonia. Thus the liquid ammonia boils in the evaporator just as refrigerant boils in any other evaporator.

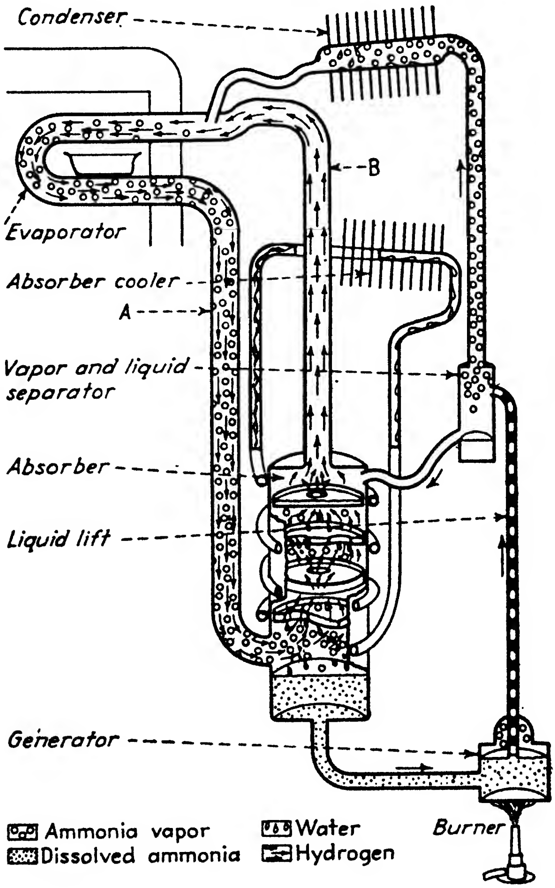


FIG. 15.—Schematic arrangement of Electrolux cycle. (*Servel, Incorporated.*)

The ammonia vapor mixes with the hydrogen gas, and the mixture is much heavier than the extremely light pure hydrogen gas in pipe *B*. Consider pipes *A* and *B*, with the evaporator and absorber, as a closed loop. Since the mixture of ammonia and hydrogen in pipe *A* is heavier than hydrogen in pipe *B*, natural circulation will take place owing to the difference in the weights of the gases in the two legs. (You will recall that convection circulation is similar to this, the lightness in one leg being caused by heating the fluid in the system.)

The hydrogen and ammonia traveling downward in pipe A enter the absorber. Within the absorber the weak aqua, practically pure water, enters the top, as we have seen. It trickles down within the absorber over a series of baffles, or trays. The ammonia gas is absorbed by the water; the hydrogen, however, will not dissolve in the water. Therefore, the hydrogen rises upward in pipe B because of the natural circulation phenomena described in the preceding paragraph. The bottom of the absorber is somewhat higher than the generator; hence, the strong aqua flows by gravity to the generator. We have now completed the tracing of the operating cycle, since we have started this description at the generator.

We have neglected to mention the cooling coil wrapped around the absorber. We urge you to neglect this cooling coil also until you understand the functioning of the Electrolux cycle itself. The cooling coil around the absorber can simply be a copper tube carrying cold city water. In fact, early Electrolux refrigerators used city water to cool both condenser and the absorber. (The absorber cooling system is similar to the cooling system of an automobile engine using no water pump. The warm fluid rises through the left leg of the loop, passes through the finned radiator to be cooled, and returns to the absorber through the right-hand leg of the loop.) Natural circulation is sufficient, and methyl chloride instead of water is used. The methyl chloride then vaporizes in the coil around the absorber and is condensed by the finned radiator.

The foregoing discussion with Figs. 14 and 15 explains the fundamentals of the Electrolux refrigeration cycle. For those interested in a more technical description of the cycle, including the various heat interchangers and mechanical design refinements, the following pages are presented as optional material.[1]

A More Technical Description—Starting at the Generator

The Electrolux unit is made up of a number of steel vessels and pipes welded together to form a hermetically sealed system. All the spaces of the system are in open and unrestricted communication so that all parts are at the same total pressure.

The charge includes an aqua-ammonia solution of a strength of about 30 per cent concentration (ammonia to water by weight) and hydrogen. For a unit of sufficient capacity for a 5 cu ft cabinet, the approximate charge is: 1.5 lbs ammonia; 3.5 lbs water; 0.01 lb hydrogen. The liquid is charged into the unit as solution and then the hydrogen is added.

The elements of the system include a *generator* (1) (sometimes called *boiler* or *still*), an *ammonia condenser* (2), an *evaporator* (3), an *absorber* (4), and a *hydrogen reserve vessel* (5). There are three distinct fluid circuits in the system: An ammonia

[1] The following four pages and Figs. 14 and 15 are reprinted by permission from "The Miracle of Ice from Heat," Copyright, 1936, Servel Inc., Electrolux Refrigerator Sales Division, Evansville, Ind.

circuit including the generator, condenser, evaporator, and absorber; a hydrogen circuit including the evaporator and absorber; and a solution circuit including the generator and absorber.

Starting with the generator, heat is applied by a gas burner or other source of heat to expel ammonia from solution. The ammonia vapor thus generated flows upwardly to the ammonia condenser. In the path of flow of ammonia from the generator to the ammonia condenser are interposed an *analyzer* (6) and a *rectifier* (7). Some water vapor will be carried along with the ammonia vapor from the generator. The analyzer and rectifier serve to remove this water vapor from the ammonia vapor. In the analyzer, the ammonia passes through strong solution which is on its way from the absorber to the generator. This reduces the temperature of the generated vapor somewhat to condense water vapor without condensing ammonia, and the resulting heating of the strong solution expels some ammonia vapor without additional heat input. The ammonia vapor then passes through the rectifier (7) where the residual small amount of water vapor is condensed by atmospheric cooling and drains to the generator (1) by way of the analyzer (6).

The ammonia vapor, which is still warm, passes on to the upper section (2a) of the ammonia condenser (2) where it is liquefied by air cooling. The ammonia condenser is provided with fins for this purpose. The ammonia thus liquefied flows into the *upper evaporator section* (3a). A liquid trap is interposed between the ammonia condenser and the upper evaporator section (3a) to prevent hydrogen from entering the condenser. The upper evaporator section (3a) is provided with fins and directly cools the food space.

Ammonia vapor which does not condense in the upper section passes to the lower section (2b) of the ammonia condenser and is liquefied and flows through another trap into the lower ice freezing evaporator section (3b).

Hydrogen gas enters the lower evaporator section (3b) and, after passing through a precooling pipe part, flows upwardly, in counter-flow to the downwardly flowing liquid ammonia. The effect of the placing of a hydrogen atmosphere above the liquid ammonia in the evaporator is to reduce the partial pressure of the ammonia vapor in accordance with Dalton's law of partial pressures. While the total or gage pressure in the evaporator and the pressure in the condenser are the same, there is substantially pure ammonia in the space where condensation is taking place and consequently the vapor pressure of the ammonia substantially equals the total pressure. Under Dalton's law, the total pressure of a gas mixture is equal to the sum of the partial pressures of the individual gases. Consequently in the evaporator the partial ammonia vapor pressure is less than the total pressure by the value of partial pressure of the hydrogen. The lesser ammonia vapor pressure results in evaporation of the ammonia with consequent absorption of heat from the surroundings of the evaporator and the cooling of the surroundings which are in a well-insulated enclosure.

The cool heavy gas mixture of hydrogen and ammonia vapor formed in the evaporator leaves the top of the evaporator and passes downwardly through the center of the *gas heat exchanger* (8) to the absorber (4). In the absorber, ammonia is absorbed by water, and the hydrogen, which is practically insoluble, passes upwardly from the top of the absorber through the external chamber of the gas heat exchanger (8) into the evaporator. Perfect separation of gases is of course not possible and some ammonia vapor passes with the hydrogen from the absorber to the evaporator. It is probably more accurate to call the gas flowing from the evaporator to the absorber *strong gas* (hydrogen strong in ammonia) and call the gas flowing from the absorber to the evaporator *weak gas* (hydrogen weak in ammonia).

Since the weight of a gas is proportional to its molecular weight and the molecular weight of ammonia is 17 and the molecular weight of hydrogen is 2, it follows that the specific weight of the strong gas is greater than that of the weak gas. This difference in specific weights is alone sufficient to initiate and maintain circulation between the evaporator and the absorber. Since the absorber is below the evaporator, it is possible to have upward gas flow in the evaporator. The long vertical column of strong gas in the central chamber of the gas heat exchange is heavier than the vertical column in the absorber, external heat exchanger space and evaporator, despite the fact that the gas in the evaporator is heavy. Consequently the gas will flow as above stated due to the difference in specific weights of the gases in the different vertical branches of the circuit. The gas heat exchanger transfers heat from the weak gas to the strong gas. This saves some cooling in the evaporator by precooling the entering gas. A liquid drain at the bottom of the evaporator is connected to the down-flow space of the gas heat exchanger.

Counter-current flow in the evaporator permits the location of the box cooling section of the evaporator in the most effective position, at the very top of the food space. Also, the gas leaving the lower temperature evaporator section (3b) can pick up more ammonia at the higher temperature prevailing in the box cooling evaporator section (3a), thereby increasing capacity and efficiency. There is still another advantage in that liquid ammonia flowing to the lower temperature evaporator section is precooled in the upper evaporator section.

The dual liquid connection between the condenser and the evaporator is advantageous in applying the unit to the cabinet. It permits extending the ammonia condenser below the top of the evaporator to provide more surface while having gravity flow of liquid ammonia to the evaporator.

The two-temperature evaporator partially segregates the ice freezing function from the box cooling function. This provides a better humidity condition in the food space because, due to the higher temperature of the box cooling section (though adequately low for proper preservation), less moisture is extracted to form frost.

In the absorber, a flow of weak solution (water weak in ammonia) comes in direct contact with the strong gas. The liquid and gas flow in counter-current. The weak solution is thus enriched or strengthened while the strong gas is weakened.

From the absorber, the strong solution flows through the *liquid heat exchanger* (9) to the analyzer (6) and thence to the strong liquid chamber (1a) of the generator (1). Heat applied to this chamber causes vapor and liquid to pass upwardly through the small diameter pipe (10), as in an "air lift," to the weak solution standpipe (11). Liberated ammonia vapor passes to the ammonia condenser as above described. The solution flows down through pipe (11) to the weak liquid chamber (1b) of the generator. Here further vapor is driven off which passes upwardly through the down-flowing liquid in the stand-pipe (11). The weak solution flows through the liquid heat exchanger (9) and to the absorber. The liquid heat exchanger precools the liquid entering the absorber and preheats the liquid entering the generator. Further precooling of the weak solution is obtained in the finned air cooled loop (12) between the liquid heat exchange and the absorber.

The heat which is liberated by absorption of ammonia in the absorber is carried away by a small quantity of volatile fluid in the coil (13) surrounding the absorber. The resulting vapor rises to a secondary condenser (14) and is there liquefied, the liquid returning by gravity to the coil (13). This enables an advantageous positioning of air-cooled surface and absorber respectively.

The hydrogen reserve vessel (5) is interposed in the equalizing or vent tube (15) and may be described as a reservoir for hydrogen gas while the refrigerator is operating under normal room temperature conditions. Under these conditions an appreciable part of the hydrogen in the system is stored in the reserve vessel. The remainder is located in the evaporator-absorber circuit and serves to balance the condenser pressure. This pressure must of course be adequate to liquefy the ammonia gas in the condenser. If the pressure is increased, the efficiency under normal conditions will be impaired, and yet it is necessary to have a higher pressure in the system to insure condensation of the ammonia under high room temperature conditions. The reserve vessel and its connection in the system is an automatic pressure variant to take care of the variable room temperature and loads and to permit lower operating pressure at lower room temperature, thereby resulting in better efficiency and higher operating pressure under extreme conditions to insure condensation of ammonia. It operates in the following manner:

Should the room temperature rise materially, ammonia vapor fails to condense in adequate quantity and some vapor flows through the equalizing tube (15). Thus additional ammonia vapor is liberated by the generator and is pushed through the condenser and into the reserve vessel, displacing the hydrogen therein, which is in turn pushed into the active part of the system. The hydrogen is actually transferred from the reserve vessel through a pipe into the absorber and then distributes itself in the evaporator-absorber circuit. This displacement of hydrogen by ammonia and the redistribution of hydrogen has a double effect. The pressure in the system is increased due to the additional ammonia gas present. This results in an adequate condensing pressure at the higher room temperature. At the same time the additional hydrogen in the evaporator serves to balance the increased condenser pressure without increasing the partial pressure of ammonia in the evaporator. Without this additional hydrogen it would be necessary to balance the increased pressure by raising the ammonia pressure in the evaporator which would result in an undesirable increase in evaporator temperature. When the room temperature decreases again, the more effective condensation in the condenser causes the ammonia gas to return from the reserve vessel to the ammonia condenser and hydrogen (weak gas) flows from the absorber into the reserve vessel.

It is possible to further improve performance by providing for automatic variation of the ammonia solution strength. The method employed is similar to that of the hydrogen reserve vessel in that a reservoir or chamber for the storage of liquid ammonia is provided. This is located at the bottom end of the hydrogen reserve vessel. For the sake of efficiency it is desired that the strongest solution consistent with good operation be used under normal room conditions. Yet it is desirable to reduce this concentration if efficient absorption is to be obtained at high room temperatures. To accomplish this, ammonia must be automatically removed from

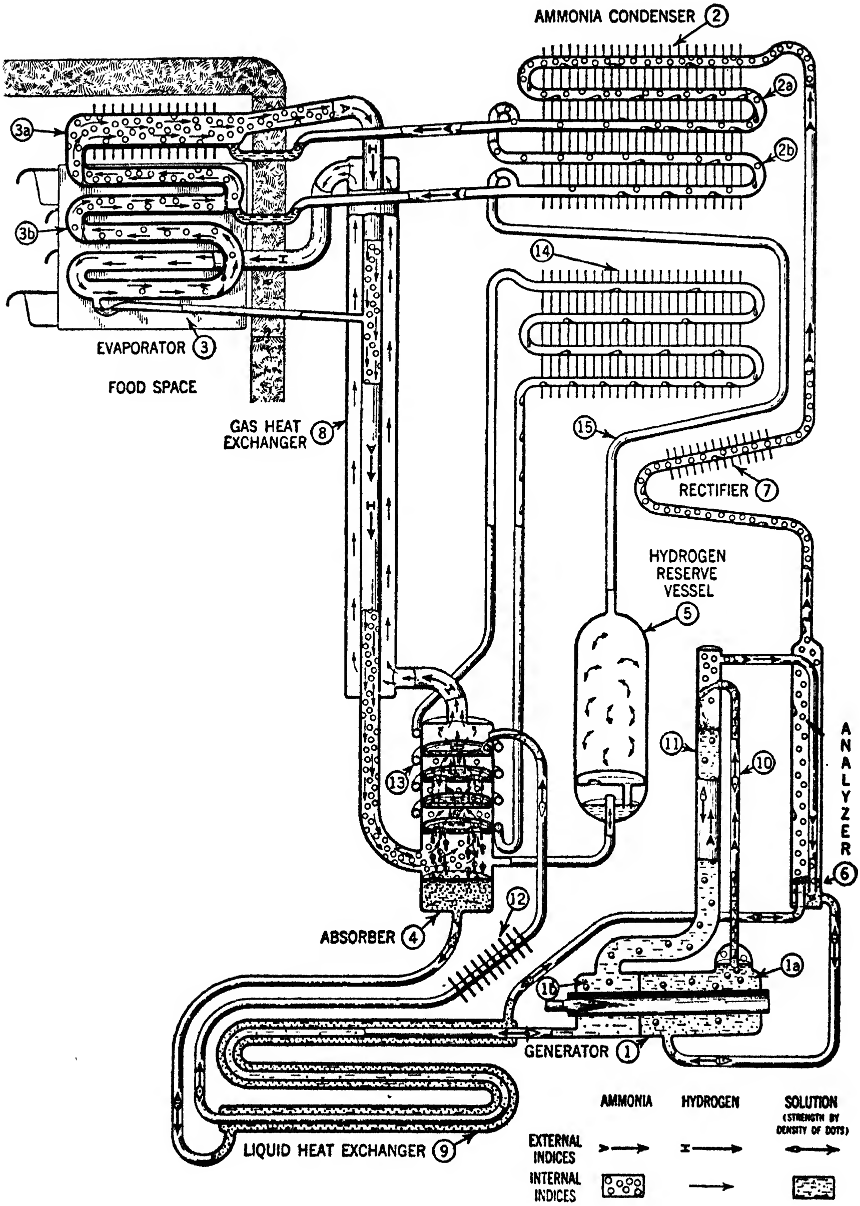


Fig. 16.—Mechanical arrangement of Electrolux cycle. (*Servel, Incorporated.*)

solution and stored somewhere in the system as liquid ammonia. When the extreme conditions are experienced, some of the ammonia which passes into the hydrogen reserve vessel condenses on the inner walls of the vessel and drains to the bottom. It collects at this point and is effectively removed from the active part of the system, thereby reducing the concentration of the ammonia in solution. As soon as normal conditions return, hydrogen flows into the reserve vessel and the liquid ammonia stored in the bottom of the vessel evaporates and is returned to the solution in the absorber by diffusion.

As refrigeration load increases, a thermostat functions to increase the flow of gas to the burner which causes a greater amount of ammonia to be expelled, condensed, and evaporated per unit of time.

PROBLEMS

1. Explain your understanding of the difference between a positive-displacement compressor and a compressor that is not of the positive-displacement type. Give two illustrations of each.

2. Describe the Rollator type of compressor. If the end of the wedge tends to wear because of friction, how is this wear compensated?

3. Describe a vane-type rotary compressor such as used in the Sears-Roebuck Coldspot refrigerators. What holds the vanes outward away from the center of the rotor? How is the check valve used? (You may illustrate your explanation with a sketch.)

4. Describe and compare the two different float-type expansion valves.

5. Describe the automatic expansion valve and the capillary-tube expansion valve.

6. Describe and illustrate with a simple sketch the thermostatic expansion valve. Why is the thermostatic expansion valve the ideal type of valve for dry-expansion (*i.e.*, not flooded) air-conditioning evaporators?

7. Difficulty has been experienced in distributing low-side liquid to all the tubes in an extended surface coil by means of a liquid-distribution header. Why? How may this difficulty be overcome and the header eliminated?

8. Sketch a continuous-absorption system, showing the path of the pure ammonia and the paths of the strong and weak aqua. Show where the pure ammonia is *liquid* and where it is *gas*. Describe or illustrate clearly on your sketch how this absorption system differs from a mechanical-compression system.

9. Assuming a fixed evaporator heat load (constant Btu per hour input to the evaporator), what determines the suction pressure in a mechanical-compression system? Under the same conditions, what determines the evaporator pressure in an absorption system such as described in Prob. 8? (Assume a flooded evaporator in both cases.)

10. (*Optional*). Sketch the Electrolux refrigeration cycle as simply or in as much detail as you wish, and briefly describe the operation of the cycle.

CHAPTER XII

APPLICATION OF REFRIGERATION TO AIR CONDITIONING

Refrigeration serves two related functions in air-conditioning work. Refer to Chap. VII in which we studied the Psychrometric Tables. You will recall that the *total heat* of air is made up of the *sensible heat* of the dry air *plus* the *latent heat* of the water vapor that is mixed with the dry air. In ordinary comfort air conditioning, refrigeration equipment must remove the sensible heat, because the air must be delivered to the conditioned space at a temperature 12 to 18° cooler than the desired dry-bulb temperature in the conditioned space. Refrigeration equipment also must do a *great deal* of work removing latent heat from air in order to reduce the dew point of the air. Suppose we have some air at 95° dry bulb, 76° wet bulb. (These are about the maximum conditions on which the design of comfort air conditioning is based in New York City. The A.S.H.V.E. "Guide" shows 95° dry bulb, 75° wet bulb as their standard design conditions; some conservatively designed installations are based on 95° dry bulb and 77° wet bulb.)

Suppose that calculations have indicated that air must be delivered to a conditioned space at a dew point of 55°. If the conditioner will deliver air at 55°, 100 per cent saturated, and if 5,000 cfm of all fresh air is drawn through our conditioner, calculate the tonnage of refrigeration required to reduce all the air to 55°, 100 per cent saturated. Now calculate the amount of this refrigeration that is removing sensible heat from the air and the amount that is removing latent heat from the air. You will find that almost as much refrigeration is required to remove the latent heat of the moisture that is being condensed as is required to remove sensible heat from the air. Thus you see that refrigeration involves two separate functions—cooling and dehumidifying—although the two functions are related because the dehumidifying is accomplished by cooling below the dew point.

There are means of securing cooling for air conditioning which do not involve the direct use of mechanical refrigeration as part of the air-conditioning equipment. We have studied several different types of refrigeration equipment; let us now note the various methods of obtaining refrigeration which are available for air-conditioning work.

Cooling without Refrigeration Machinery
- Well water
- Meltage of ice
- Evaporative cooling

Refrigeration from Man-made Machines
- Centrifugal water-vapor refrigeration
- Steam-jet water-vapor refrigeration
- Mechanical refrigeration with standard chemical refrigerants

All these items are familiar to you except "evaporative cooling." Since the evaporative cooling is a separate subject requiring a review of psychrometry, we shall leave it until the end of this chapter.

Basic Economics of Air Conditioning. The purchase of air conditioning by any commercial or industrial establishment involves serious thought on the subject of dollar cost factors. Factories, stores, hotels, and restaurants do not install air conditioning unless they feel that it will pay them in one way or another. Department stores have proved that customers stay in the store longer, spend larger average amounts of money, and that fewer "shoppers" stroll in and out of the store without buying anything. Restaurants have shown that the average size of checks increased from 25 to 60 per cent, in addition to which more patrons were attracted. Hotels long ago found it necessary to air-condition their restaurants and certain of their public rooms, and many of them are now air-conditioning the guest rooms. Except for a few old-fashioned customers, hotels found in general that air-conditioned guest rooms brought a higher rate than rooms not conditioned. Factories in all lines, from automobile manufacturers to textile plants as small as those which produce woven labels, air-condition only when the cost of the venture has been proved worth while.

Furthermore, when commercial and industrial establishments make the decision to install air conditioning, they purchase equipment carefully with an eye to the lowest *over-all* cost. It is necessary for one who desires to work intelligently in the air-conditioning industry to understand something about the economics of air conditioning as seen through the purchasers' eyes.

We just mentioned the phrase "over-all cost." From the standpoint of a purchaser and user of air conditioning, over-all cost is made up of the following factors:

Fixed Cost
- Interest on investment
- Depreciation
- Taxes, insurance, etc.

Operating Costs
- Power, water, steam, etc.
- Maintenance and repairs
- Replenishment of refrigerant
- Operating personnel pay roll

It may seem somewhat tedious to have to study finance, bookkeeping, and economics, but at this point it is absolutely necessary.

First let us discuss the items under "operating costs," with which we are already familiar. Electric power, steam, and water are ordinarily metered at definite rate schedules, and their cost to the air-conditioning user can be accurately figured. Repairs and maintenance cover the costs of new V-belts, replacements of bearings, valves, controls, and other items that show wear and must be repaired or replaced. Compressors must be repacked. All these items are charged to maintenance and repairs, including the necessary labor.

The replacement of refrigerants is really a part of maintenance and repairs, but in air-conditioning bookkeeping it is ordinarily shown as a separate item, depending upon the current market price and the quantities purchased. Freon may cost 45 to 64 cts per lb. Through carelessness, large installations have been known to leak several thousand pounds of Freon during the winter shutdown season. Depending upon the condition of shaft seals, as well as valves, piping connections, etc., normal operating losses of Freon may run from $50 to many hundred dollars per year on a large installation. Even small installations as they grow older are liable to lose 10 to 20 lb of refrigerant per year. Since the loss of refrigerant is largely within the control of the operators of the plant, separate cost records are kept regarding this item.

As to operating personnel, small installations may not require the hiring of an engineer whose specific job is to operate the air-conditioning equipment. Many larger installations, however, require a licensed operating engineer in attendance whenever the equipment is operated. This means that an installation operating 12 to 15 hr per day requires two operating engineers daily on different shifts. In some localities these operating engineers are members of a union which requires that the air-conditioning user retain the operating engineers on the pay roll during the winter (although possibly at a reduced salary). Ordinarily, no operating personnel is charged as a part of the operating costs of an air-conditioning plant unless the installation of a particular plant requires the hiring of *additional* employees *not formerly* on the pay roll. Ordinarily in large city buildings, the operating personnel in the building can take care of any reasonable amount of air-conditioning equipment that might be installed.

However, if a building in the city, having no air-conditioning equipment at all, makes an installation of sufficient size so that the owner is required by law to hire additional personnel to operate this equipment, the owner must count the cost of the additional man or men as part of the cost of the air-conditioning equipment. Most legal regulations of this nature are based upon the number of pounds of refrigerant in any one system, or on the number of tons of capacity of any one system. Thus, sometimes we can use, say, two separate 20-ton direct-expansion refrigerating systems with the evaporators located in separate conditioners without being compelled to have a special operating engineer for this equipment. On the same premises it would be possible that a single 40-ton installation would require the service of a special operating engineer. You can readily understand that the greater cost of installing the two smaller machines would be more than worth while to the building owner in terms of the pay-roll saving possible.

So much for the subject of operating costs—these are simpler and more generally understood than fixed costs.

Fixed Costs. We can list the components of fixed cost as follows:

Interest on investment
Depreciation
Taxes, insurance, etc.

If a man invests $10,000 in air-conditioning equipment, he may either use his own money or borrow money to make the purchase. If he borrows the money he must, of course, pay interest to the bank on the borrowed funds. If he uses his own money, he must sell stock or bonds to get the cash or must draw the money from the savings bank. In any event, if an air-conditioning purchase is made with the purchaser's own funds, the purchaser has deprived himself of the use of this cash and of the interest that it can bring. If you have $10,000 invested in 4 per cent bonds, your investment pays you $400 per year interest. If you then sell these bonds and buy air-conditioning equipment with the money, you are sacrificing $400 a year income in order to make the purchase. From this it should be clear that the purchase of any equipment, air conditioning or any other type, involves an *interest cost* to the purchaser. The question remains as to what interest rate should be stated. Some years ago, interest charges of 8 to 10 per cent were made on purchases of this type. Today 6 per cent would be the highest rate of interest that is justifiable on a purchase of this kind.

The actual interest rate to be figured in the cost of a specific air-conditioning installation depends upon the purchaser. This information can ordinarily be secured from the purchaser's accounting department, and the rate today will usually be found to be between 4 and 6 per cent. For our examples in this chapter, we shall arbitrarily pick 5 per cent as the interest rate.

The item of depreciation is the largest contributor to fixed cost. Most purchasers hope that air-conditioning equipment and similar equipment will last 12 to 15 years before it must be discarded entirely. However, progress is being made so rapidly in the design of such equipment that many installations that were made 8 to 10 years ago are being literally scrapped today. Also if equipment is to operate 24 hr a day, 6 or 8 months of the year, it will probably have to be replaced earlier than equipment that operates 10 hr a day, 4 months per year. Ordinarily, 10 to 12 years is considered the life of an installation of equipment such as an air-conditioning installation. (Some installations being incorporated in new buildings of the finest type, such as insurance companies and banks, are being depreciated over a 20-year period.) For purposes of our discussion in this chapter and for simplicity in figuring, we shall consider a depreciation period of 10 years.

Let us return to our imaginary purchase of $10,000 worth of air-conditioning equipment. If this is to be depreciated over a 10-year period, we must consider that the equipment will be worth only its salvage or scrap value at the end of that period. In the case of most heavy equipment the scrap value is not more than sufficient to pay for the dismantling and removal of the equipment from the building. This means that our new $10,000 air-conditioning purchase must be considered as worth $9,000 after the end of the first year, $8,000 after the end of the second year, and so on. Suppose that you were buying this $10,000 air-conditioning installation we have discussed for your restaurant. After having purchased the installation, you would have to withdraw $1,000 per year from the business with which to set up a "depreciation reserve." Thus at the end of 10 years, when the installation needed to be replaced, you would have your original $10,000 cash on hand with which to replace the old installation. (This statement excludes the effect of compound interest. Actually you would have to charge perhaps $825 per year depreciation in order to have your original $10,000 at the end of 10 years when the equipment becomes of zero value.) For purposes of our comparison, we shall take 8½ per cent of the original investment as the yearly depreciation charge.

Taxes and insurance depend upon the city in which the air-conditioning equipment is installed, the type of insurance coverage carried by the purchaser, and on many other factors, Most industrial concerns and large commercial establishments know to the fraction of a per cent the amount of their insurance and tax expense on the purchase of new equipment. This item cannot be included in the cost of an air-conditioning installation. Talk with anyone who has installed $10,000 worth of new equipment of any kind whatsoever and you will find that the tax assessor knows about it and that the purchaser's tax bill is increased. Similarly, the purchase of new equipment means that the purchaser must include such equipment in his fire and casualty insurance. The result means an additional insurance premium each year. The amount to be charged for taxes and insurance may vary from less than 1 per cent to as much as 3 to 4 per cent. For purposes of our comparisons in this chapter, we shall arbitrarily use the value of 1½ per cent of the original investment to cover taxes and insurance expense.

What Type of Refrigeration to Install. Let us first take up the question of well water, which is one way of getting cooling for air conditioning without refrigeration machinery. If well water is available at a temperature of 55°F or lower, in a sufficient and dependable supply, it is by all means the cheapest and best way of providing cooling for air conditioning. The only mechanical equipment involved is the well-water pump. Even if a new well must be dug for the job, the cost of the well including the pump is a good deal less than the cost of refrigeration machinery to do the job. The operating cost of the pump is small, its maintenance expense is small, and you have the cheapest source of cooling for air conditioning.

Unfortunately well water is not often available. In many localities one can drill 400 to 500 ft without striking water. In many other localities the temperature of the well water is too high. Remember the dew point of air is lowered ordinarily by cooling the air. With well water at 55° it is possible to get air at a dew point of about 58°. This is the maximum dew point that can be tolerated in conditioned air in the ordinary comfort air-conditioning job. Naturally the dew point of this air increases after the air has been introduced into the conditioned space because of the humidity load of people and appliances. The western end of Long Island, including Brooklyn, is an example of a territory where well water is available but is too high in temperature. At the start of the air-conditioning season, early in June, this well water is at a temperature above 55°. By the end of August the temperature of the well water rises to 60 to 68°. Before using well water for air conditioning the temperature of the well water in that locality *at the end of the summer must be investigated.* Some localities have well water at 53° in May but at 63° in August. Such well water can produce only the cold, clammy, high-humidity, uncomfortable type of air conditioning and should not be used.

Assuming that well water is not available, let us consider three methods of producing refrigeration by machinery.[1] Since water-vapor refrigeration is still relatively new and is not offered by many different manufacturers, and since it is limited to larger installations, we shall not consider centrifugal water-vapor construction in our cost comparison. Let us consider a small theater installation requiring 40 tons of refrigeration capacity. Let us consider three different theaters each providing three different means of refrigeration, as indicated by Table 1.

Table 1 gives the answer as to the type of refrigeration that should be installed. In order that you may understand this, we want to work out the details of the cost comparison given in Table 1. Every individual job requires its own analysis as to cost because the different installations have individual factors that will influence the economic analysis. Let us base

[1] We are deliberately omitting absorption refrigeration from consideration throughout this chapter because it is little used in air-conditioning work.

TABLE 1

Type of refrigeration	Per season of 1,000 hr		300 hr per season
	Theater *A* near cheap electric power	Theater *B* where cheap steam is available	Theater *C* New York average steam and electric cost
Steam-jet	No	Yes	No
Mechanical	Yes	No	No
Ice meltage	No	No	Yes

our economic analysis on the assumed costs, given in Table 2, for the various services in the respective cities:

TABLE 2

	Theater *A*	Theater *B*	Theater *C*
Electric power (per kwhr)	\$0.02	\$0.04	\$0.03
Steam (per 1,000 lb)	1.00	0.60	0.80
Water (per 1,000 gal)	0.20	0.15	0.20
Operating engineer (per season)	\$1,200.00	\$1,200.00	\$1,000.00
Ice (per ton)	3.50	3.50	3.25

Table 2 shows the comparison of the cost of the various utility services and the cost of operating engineers' services in the three cities where our three theaters are located. Let us now decide on the type of equipment to provide cooling: Compare the items that would determine the fixed cost and the operating cost for steam-jet, mechanical-refrigeration, and ice-meltage systems, *regardless of the city in which the plant might be located.* We shall use a load factor of 70 per cent.

Mechanical Plant. We must estimate the installation cost; \$9,000 is a fair price for a union-installed plant of this size, using city water for condenser cooling. Such a plant would have a 40-hp compressor (or perhaps two 20-hp compressors) and about 10-hp of fan and pump motors. Since 1 hp equals 0.746 kw of electrical energy, we have

$$\frac{(50)(0.746)(0.7)}{0.85} = 30.8 \text{ kw}$$

as the electric load per *average hour* (the 0.7 is load factor; the 0.85 is motor efficiency, see Chap. V).

At full load the plant would use 60 gpm of cooling water, so per average hour it would use

$$60(60)(0.7) = 2{,}520 \text{ gal}$$

This plant would use no steam or ice.

Since a season of 1,000 hr would represent perhaps 120 days (4 months) at 8 hr per day, one operating engineer would cover this plant. "Maintenance and repairs" would, of course, be zero the first year and might be \$600 to \$700 the year of a major overhaul. Let us estimate an average of \$200 per year for this item, and \$40 per average year for refrigerant replacement.

Steam Jet. Installation cost would be slightly greater, say \$10,400 for a plant with the same air-washer ducts, fans, etc. This steam-jet plant would have no large electric motor-driven compressor; it would have about 12-hp electric-driven fan and pump motors. It would use about 22 lb of steam per hour per ton of refrigeration capacity. Hence we get electric power and steam consumption per average hour as follows:

$$\frac{12(0.746)(0.7)}{0.85} = 7.4 \text{ kw}$$

$$22(40)0.7 = 610 \text{ lb per hr}$$

At full load the plant would use 120 gpm of condensing water, so per average hour we have

$$60(120)(0.7) = 5{,}040 \text{ gal}$$

For the operating engineer we have the same situation as in the case with the electric-driven mechanical plant. Maintenance and repairs are less—say an average of \$80 per year—as we have no compressor, V belts, expansion valves, refrigerant driers, etc., and the steam jet has no moving parts. The refrigerant being water, no refrigerant cost is involved.

Ice Meltage. Since an ice-meltage chamber replaces the water-chilling machinery, the installation cost is much lower—about \$5,500 would install the system with identical ducts, fans, air washer, etc. The same chilled-water pump used in the mechanical plant would suffice; so this plant would have about 10 hp of electric motors. It would use no water or steam and would have no refrigerant-replacement expense.

$$\frac{10(0.746)(0.7)}{0.85} = 6.15 \text{ kw}$$

Ice consumption can be easily obtained, knowing the definition of "1 ton of refrigeration capacity" (see Chap. VIII). In 24 hr at full load this plant would melt 40 tons of ice, since it is a 40-ton plant.

Hence in one full load hour it would melt

$$\frac{40}{24} = 1.74 \text{ tons of ice}$$

At 70 per cent load factor this becomes

$$0.7(1.74) = 1.22 \text{ tons}$$

We know of no cities requiring a licensed operating engineer for an ice-meltage plant. Maintenance and repairs are very low—about $50—as pumps and fans are the only moving parts.

Note: Actually the load factor of the electric-driven fans and pumps is greater than that of the refrigeration load; it should perhaps be about 90 per cent. However, this would complicate this analysis without changing it appreciably in final results.

Fixed Cost. You will recall that on page 206 we decided to use fixed charges as follows, expressed as per cent of the original cost of installation:

Depreciation	8.5
Interest	5.0
Taxes and insurance	1.5
Total	15.0

Let us now tabulate the information we have just discussed and derived. Table 3 shows the three plants compared as to these factors and holds *regardless of where the plants are located.*

TABLE 3

	Mechanical	Steam jet	Ice
a. Installation cost	$9,000	$10,400	$5,500
b. Electric power consumption* (kw)	30.8	7.4	6.15
c. Steam consumption* (lb)	....	610	....
d. Water consumption* (gal)	2,520	5,040	....
e. Ice consumption* (tons)	....	....	1.22
f. Operating engineer expense	1 man	1 man	
g. Maintenance and repairs (per year)	$200	$80	$50
h. Replenishment of refrigerant	40	....	....
i. Fixed charges (15 per cent of *a*)	$1,350	$1,560	$825

* Per average hour, not per full load hour; load factor 70 per cent.

Table 3 is merely a tabulation of operating requirements and operating-cost data. Note carefully that this information, as shown in Table 3, compares three different types of installation, regardless of the cities in which the installations may be located. For instance, the requirement as to electric power, steam, water, and ice are shown in terms of the quantities of these items which are required, without regard to the unit cost of these services in the various cities. Before proceeding further, make sure that you understand Table 3, because the following discussion is based upon it.

We now wish to *consider each of the three cities* in which our hypothetical theaters A, B, and C are located. We want to decide then, on the basis of over-all cost, which type of installation is best for each theater. To do this we must, *for each of the three theaters,* calculate the over-all cost based upon the use *of all three alternate systems.* Table 4 illustrates this comparison of the three systems as applied to theater *A*.

Items *a, b, c,* and *d* in Table 4 are reprinted from Table 3 for convenience. Note that the mechanical system uses no steam or ice, the steam-jet system uses no ice, and the ice system uses neither steam nor water. Items *e, f, g,* and *h* show the annual cost based upon the data originally shown in Table 2. Items *i, j, k,* and *l* are again taken directly from Table 3, as these items do not depend upon the particular theater or city being considered. Note that in calculating items *e, f, g,* and *h* an *operating season of* 1,000 *hr* has been used. The mechanical installation has an electric-power requirement of 30.8 kw; for 1,000 hr of operation, 30,800 kwhr would be consumed. Hence 30,800 kwhr multiplied by a power cost of 2 cts per kwhr gives us an electric-power cost of $616 per season for the mechanical-refrigeration installation. Steam, ice, and water costs are calculated in a similar manner.

The over-all cost figures for theater *A* are shown in Table 4, based on a season of 1,000 hr and on the data in Tables 2 and 3.

TABLE 4.—THEATER *A* SEASON 1,000 HR

	Mechanical	Steam jet	Ice
a. Electric power used (kwhr)	30,800	7,400	6,150
b. Steam used (lb)	....	610,000	....
c. Water used (gal)	2,520,000	5,040,000	....
d. Ice used (tons)	....	....	1,220
e. Electric power cost	$616	$148	$123
f Steam cost	....	610	....
g. Water cost	504	1,008	....
h. Ice cost	....	....	4,270
i. Maintenance and repairs	200	80	50
j. Refrigerant	40	....	....
k. Operating engineer	1,200	1,200	....
l. Fixed charges	1,350	1,560	825
Total	$3,910	$4,606	$5,368

Similarly we can set up tables for theaters *B* and *C*, as shown in Tables 5 and 6.

In examining Tables 4, 5, and 6 we are interested only in the last line of each table in getting our picture of which system shows the lowest over-all cost for each of the three theaters. Studying the last lines of each of these tables you will find that the mechanical refrigeration plant with a cost of $3,910 per year is definitely the best for theater *A*. For theater *B* the steam-jet plant is about $150 cheaper than the mechanical refrigeration plant, while the ice installation is about $1,000 higher than the others. However for theater *C*, *where the operating season is only* 300 *hr*, the ice

TABLE 5.—THEATER *B*. SEASON 1,000 HR

	Mechanical	Steam jet	Ice
a. Electric power used (kwhr)	30,800	7,400	6,150
b. Steam used (lb)	....	610,000	....
c. Water used (gal)	2,520,000	5,040,000	....
d. Ice used (tons)	....	...	1,220
e. Electric power cost	$1,232	$296	$246
. Steam cost	....	366	....
g. Water cost	378	756	....
h. Ice cost	....	....	4,270
i. Maintenance and repairs	200	80	50
j. Refrigerant	40	....	....
k. Operating engineer	1,200	1,200	....
l. Fixed charges	1,350	1,560	825
Total	$4,400	$4,258	$5,391

TABLE 6.—THEATER *C*. SEASON 300 HR

	Mechanical	Steam jet	Ice
a. Electric power used (kwhr)	9,240	2,220	1,845
b. Steam used (lb)	....	183,000	....
c. Water used (gal)	756,000	1,512,000	....
d. Ice used (tons)	....	....	366
e. Electric power cost	$277	$67	$55
f. Steam cost	....	146	....
g. Water cost	151	252	....
h. Ice cost	....	....	1,190
i. Maintenance and repairs	$200*	80	50
j. Refrigerant	40	....	....
k. Operating engineer	1,200	1,200	....
l. Fixed charges	1,350	1,560	825
Total	$3,218	$3,305	$2,120

* Might be lower due to short season, but intermittent operation frequently does not lower maintenance cost.

installation is over $1,000 cheaper than the mechanical or steam-jet equipment.

We have seen from Tables 4, 5, and 6 that our answer to this problem of system selection was correct, as given in Table 1. Although this can be recognized from a study of Tables 4, 5, and 6, it is customary to present one further table as a conclusion to an analysis such as this. Such a table summarizes briefly the conclusions indicated in our three preceding detailed tabulations. This table is of interest because for each theater the fixed charges are shown in relation to summarized statement of the operating costs. These operating costs in turn are broken into two items: (1) the utility services (power, water, steam, and ice) and (2) operating costs other than these utility services. Table 7 presents this material.

TABLE 7.—RECAPITULATION OF OVER-ALL COSTS

	Mechanical	Steam jet	Ice
Theater *A*:			
Power, water, steam, ice..............	$1,120	$1,766	$4,393
Other operating costs.................	1,440	1,280	50
Fixed charges.........................	1,350	1,560	825
Total.................................	$3,910	$4,606	$5,368
Theater *B*:			
Power, water, steam, ice..............	$1,610	$1,418	$4,516
Other operating costs.................	1,440	1,280	50
Fixed charges.........................	1,350	1,560	825
Total.................................	$4,400	$4,258	$5,391
Theater *C*:			
Power, water, steam, ice..............	$ 428	$ 465	$1,245
Other operating costs.................	1,440	1,280	50
Fixed charges.........................	1,350	1,560	825
Total.................................	$3,218	$3,305	$2,120

The analysis that has been presented is very simple. It is based upon a good deal of information that we have, in this case, simply assumed. The information that we have shown in Table 2 can be obtained from the companies supplying these various services. The expense of an operating engineer would depend upon the prevailing salary for such men in any particular city and also upon whether one or two engineer shifts were required. The information that we have shown in Table 3 can be obtained only by an air-conditioning man with considerable experience, and even then compiling such a table would involve several hours of work. This is because the data shown in Table 3 must be estimated regarding an air-conditioning installation not yet installed. Familiarity with the proposed equipment and an accurate guess as to the operating conditions are both essential. From this point on, the analysis is simply a matter of arithmetic.

Comparisons of this type are extremely important. In many cases it will not be simply a comparison between mechanical, steam-jet, and ice refrigeration such as we have worked out. In some cases a certain restaurant can perhaps be handled by a central system, by individual units, or by a central compressor system supplying two or three ceiling-hung conditioners. In a case such as this, the three alternates would have to be set up and compared both as regards installation costs and operating charges. The installation to be recommended to the customer would ordinarily be the one having the lowest over-all cost. However, in some cases a customer might, for reasons of his own, prefer one specific type of installation. In this event, he requests the air-conditioning man to investigate how much more this type may cost compared with the cheapest way of doing the job. Also because of city-code requirements in many cities, comparisons frequently must be made between a direct-expansion and an indirect-expansion (chilled-water) installation. Let us consider the following example. In New York City if a shop owner wishes to air-condition the main floor and the second floor of an establishment, the building code prohibits running direct-expansion lines to the second floor from a compressor located below the second floor. The use of a chilled-air duct to the second floor from a direct-expansion conditioner on the first floor is also prohibited. Hence, such an installation would require an indirect-expansion refrigeration machine if one refrigeration machine were to be used. However, entirely separate and independent equipment can be located on each of the two floors. It is impossible to state which of these would be cheaper as a general rule. The answer depends upon the specific case. The two independent direct-expansion installations might be the best in one case, while under other conditions the indirect-expansion job might be cheaper in spite of the extra equipment involved.

A further word in connection with the use of ice is worth while at this juncture. Refer to Fig. 1, which illustrates graphically the comparison regarding theater *C*, as given in Table 7. From this graph you will note that theater *C* should definitely be an ice installation up to an operating season of about 660 hr. Beyond this point, the over-all cost of ice is higher than with mechanical refrigeration. In the vicinity of New York City, installations that operate less than 700 hr or thereabouts should be investigated carefully with regard to the cost of an ice-meltage system. A graph, such as Fig. 1, could be drawn for any proposed installation being studied in this manner. Such studies in the past have resulted in the following situation, regarding which few people are informed: *All* the legitimate theaters in New York City that are air-conditioned at all (including the Shubert theaters, all of which are cooled) utilize ice meltage *as the source of chilled water*. Two large chains of sandwich shops have over a dozen of their locations air-conditioned, and *all of them* utilize ice meltage for chilled water. The reason is easily understood; a legitimate theater is occupied six evenings and two afternoons per week, thus requiring cooling for approximately 35 *hr per week*. Such a theater is fortunate if it is booked for 10 weeks per summer season, which gives an operating season of only about 350 hr. Similarly, a sandwich shop is crowded from 8 to 9 A.M., from 12 to 2, during the luncheon rush, and does only a scattered dinnertime business from 6 to 7 P.M. These sandwich shops cannot afford to be air-conditioned during the entire day, but only during the hours when the restaurants are crowded. Hence they consider their operating season as approximately 5 hr per day, 90 to 100 days per summer. For establishments such as these, *even though* *the ice bill per operating hour is much higher than the power and water bill for a compressor, the over-all cost of an ice installation is lower because of the short operating season.* Theater *C* in Table 7 illustrates this point clearly.

Railroad air conditioning was the subject (about two years ago) of an exhaustive study by the Association of American Railroads. The result of this study showed that a railroad car could be equipped with an ice-activated air-conditioning system for $3,500 to $4,000. Various types of mechanical installations cost $6,000 to $7,000. In spite of the cost of ice, the lower first cost enabled the ice system to show a lower over-all cost for a

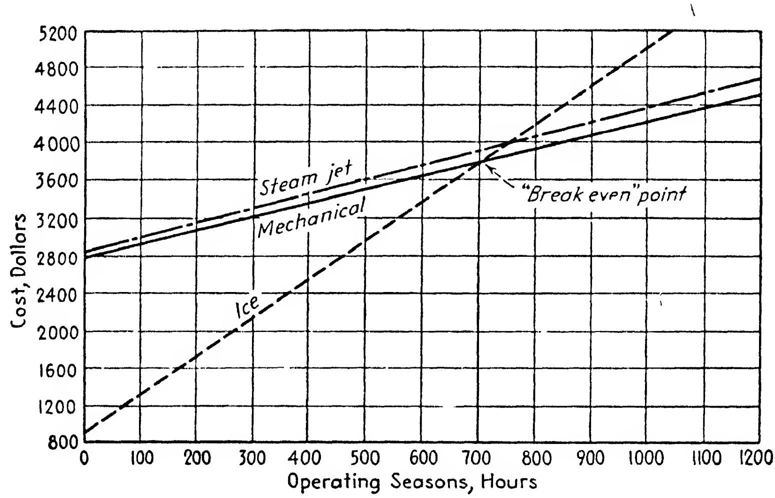


FIG. 1.—Comparison of systems at theater *C*.

cooling season of 3 months, regardless of how many miles the car traveled per season. For a 5-month cooling season, the ice system was cheaper than any other up to 250,000 car-miles per car. As might be expected for an 8-month cooling season, the ice-activated system lost its over-all cost advantage at slightly over 100,000 car-miles per year.

In previous chapters we have discussed centrifugal water-vapor refrigeration and absorption. We do not need to discuss either of these types of refrigeration as they relate to air conditioning and the cost of cooling. In the case of centrifugal water-vapor refrigeration, the initial cost is somewhat higher than for a first-class mechanical-refrigeration installation. Except for the cost of refrigerant (which is nil in the case of water-vapor systems), there are no compensating operating cost savings. Furthermore, this type of refrigeration is available only in fairly large units and is being commercially used only by one manufacturer. Its application is still highly specialized. Absorption refrigeration has almost never been used in air conditioning, and its use both in air-conditioning and general refrigeration work is waning rather than increasing.

Evaporative Cooling. At the beginning of this chapter we mentioned evaporative cooling, but we have not discussed it in comparison to the other types of cooling which are available in air-conditioning work. Evaporative cooling is done in an air washer or in a similar piece of equipment. Suppose

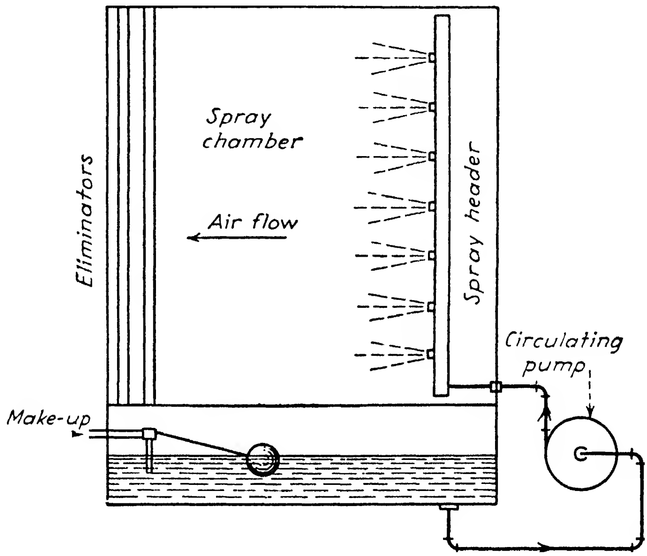


FIG. 2.—Adiabatic air washer.

that you have air at 95°F and at a dew point of about 45°F. From the Psychrometric Tables (or from the chart, if you know how to use one) you can see that this air is below 20 per cent relative humidity. Suppose that we pass such air, the precise conditions being 95° dry bulb and 65° wet bulb, through an air washer, the air washer being insulated and the spray water being recirculated from the pan of the washer. As you know, the temperature of the water in the air washer will soon drop to almost 65°F, regardless of its initial temperature. Remember each tiny drop of spray in the air washer acts like a wet-bulb thermometer and tends to cool to the wet-bulb temperature of the air. Refer to Fig. 2 for the spray hookup.

If the air washer is well insulated, it will be impossible for the total heat of the air to change in passing through the air washer. However, the vapor pressure of the water vapor already in the air is about 0.3 in. Hg, while the vapor pressure of the water in the air washer (at 65°F) is about 0.62 in. Hg. Obviously, water will evaporate into the air, increasing its dew point. After all the water in the washer is cooled to about 65°, the heat required to vaporize the water that evaporates can come only *from the sensible heat of the air.* Therefore, as the air passes through the air washer, its dew point is increased, owing to the evaporation of the water into the air, and its *dry-bulb temperature is decreased,* because sensible heat is removed from the air. Since the *total* heat of the air remains *unchanged,* the wet-bulb temperature of the air does not change. (If you have difficulty in realizing that the total heat of the air does not change, try to figure out where this heat can go. You will see that it can go nowhere. There is no water piping leading away from the air washer. There is no outside source of chilled water or refrigeration. The only external connections to the air washer are the duct work. If the air washer is well insulated, the heat content of the air leaving the air washer *must* be the same as the heat content of the air that entered, because there is no way for heat to get into or out of this air.)

From the foregoing you can see that our original air at 95° dry bulb, 65° wet bulb can be cooled to below 70°F by permitting this relative humidity to increase to about 80 per cent. In other words, as far as *sensible heat* of the air is concerned, we have achieved a *temperature reduction of more than* 25° without any expense for refrigeration or even for cold well water. The use of air such as this in air conditioning does not result in *ideal* comfort conditions in conditioned spaces. However, since the only cost of such cooling is the power for the fan to move the air (plus a small pump on the air washer), large quantities of 100 *per cent fresh outdoor air* may be used and acceptable inside conditions may be produced. *This method of cooling is possible only where the outdoor dry bulb is extremely high,* combined with *abnormally low outdoor dew points.* This method of cooling is not successful around New York City because the outdoor air is only acceptable for this adiabatic (or "evaporative") cooling for about a half dozen days per season. However, such evaporative cooling is used with great success throughout the entire American Southwest. In Arizona, outdoor conditions of 110° dry bulb with relative humidities of 10 per cent are not uncommon. With such conditions prevailing on the street, it is decidedly pleasant to walk into a store that is at 85° and 55 per cent relative humidity. On an average New York City summer day, conditions of 85° and 55 per cent relative humidity would be worse than those outdoors and, hence, would not be acceptable at all. As we shall learn in a later chapter, ideal conditions *within* a conditioned space depend to a great extent upon the *outdoor conditions* prevailing, as well as upon personal taste and the operating cost that can be tolerated.

Rogers Research System of Air Conditioning. In the Research System of Air Conditioning the problem is approached from an entirely different point of view than in conventional systems in which refrigeration is used. In the Research installation, the mechanical equipment is greatly simplified—the major moving parts are blowers and centrifugal pumps.

This reduction of mechanical equipment results in low operating costs; the simplification of equipment results in low maintenance costs. The Research System is a system of "direct dehumidification." It satisfies the requirements of good air conditioning by maintaining specified indoor con-

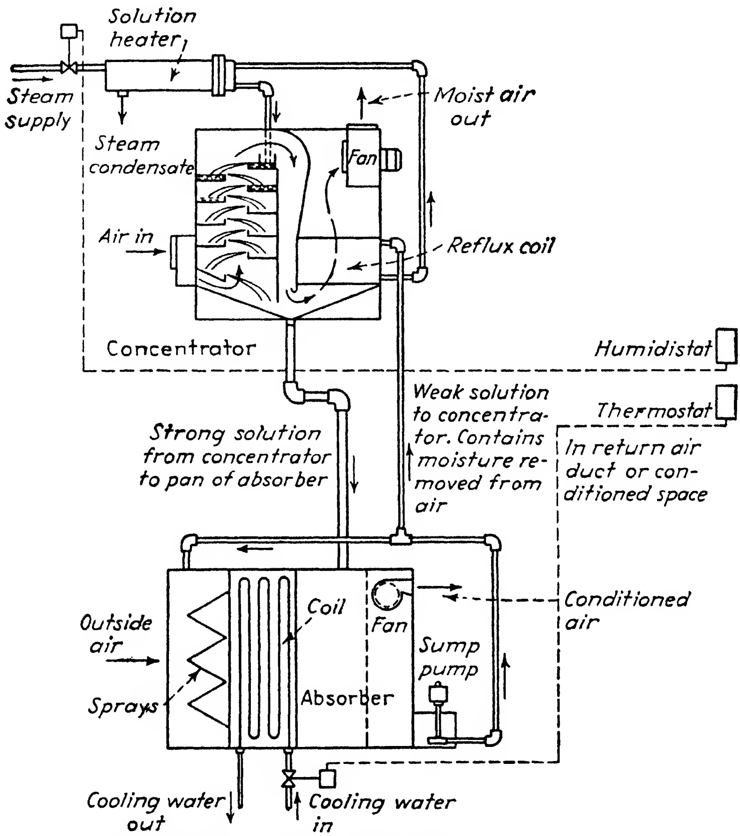


FIG. 3.—Diagram of Rogers Research System of Air Conditioning.

ditions at all times, regardless of the temperature and humidity of the outdoor air. This is possible since the changes in temperature and humidity are accomplished by independent means, even though they occur simultaneously.

"Direct dehumidification" is accomplished by a chemical that is very hygroscopic (adsorbs moisture). Triethylene glycol is the drying agent used. This chemical is a standard commercial product manufactured by several of the large chemical supply companies. Triethylene glycol is colorless, odorless, noninflammable, nontoxic, nonelectrolytic, and noncorrosive.

Limitations of triethylene glycol are few. Its maximum moisture-removing capacity is limited to leaving air in a condition of 20 per cent to 25 relative humidity, but as it is used in the Research System, its properties are constant and its capacity for moisture absorption remains unchanged for any given temperature.

See Fig. 3 for a flow diagram of the Rogers Research System of Air Conditioning.

PROBLEMS

1. List three means by which cooling may be obtained for air-conditioning purposes without the use of mechanical refrigeration or steam-jet refrigeration. Which of these three can duplicate the results of an ordinary refrigeration system?

2. Suppose that a new theater is to be constructed in Summit, N.J. The purchaser has one proposal for a mechanical refrigeration job of 100-ton capacity (assume this is the correct load), the cost of which will be approximately \$21,000. The purchaser has another proposal based upon using 125 gpm of well water at 53°F, the well water to be warmed to 73°F and then discarded. The well-water job will cost approximately \$12,000, including the well and the deep well pump. The contractor quoting on the well-water job makes the same guarantees for the conditioned space as are made for the mechanical-refrigeration job. He also is willing to issue a separate guarantee regarding the well, covering its capacity and guaranteeing that the water will never exceed 53°F. The man who is operating the new theater would like to save money and purchase the well-water installation because its operating cost as well as its first cost offer him attractive savings. However, the savings are so great that he is worried that the well-water installation will not be satisfactory. Which installation should the prospective purchaser buy? (State all your reasons.)

3. Assume that an industrial plant has called upon you to make a cost analysis regarding additional air-conditioning refrigeration which they plan to install in connection with a drying room. Since the plant has waste steam available, which is charged at a very low figure to the departments using it, you have recommended the installation of steam-jet refrigeration. From the plant officials you have received the following information:

Electric power	\$0.02 per kwhr
River water	0.01 per thousand gallons
Steam	0.15 per thousand pounds
Depreciation rate	12.5% of installation cost
Interest rate	5.25% of installation cost
Taxes, insurance, and miscellaneous	1% of installation cost
Season—20 hr per day, 180 days per year	
Load factor—60%	

As a result of your work on the proposed installation, you have gathered the following data regarding the proposed steam-jet equipment:

Installation cost	\$4,800.00
Steam rate at 10 psig	43 lb per ton per hr
Condensing water	6 gpm per ton
Electric power (not affected by load factor)	7 hp (efficiency 80%)
Maintenance—one pump overhauled per year, gauge and valve repair, etc.	\$36 per year

Neglect throughout this problem the item of operating engineer's expense.

Show the fixed cost, item for item, and show the total fixed cost in dollars and in per cent of the installation cost. What is the fixed cost per hour of operation?

4. Find, for the plant in Prob. 3, per *full load* hour, per *average* hour (note load factor), and *per season*, the following:

Electric power consumption
Steam consumption
Water consumption

5. Find the dollar costs for the items calculated in Prob. 4.

6. Tabulate *total* over-all cost per season for the plant covered in Probs. 3, 4, and 5.

7. The plant officials notify you that they just discovered that their pipe line from the river already is overloaded; hence they will have to buy city water at 10 cts per thousand gallons for condensing water. Also, they now want a comparison of an alternate installation using mechanical refrigeration. You obtain the following data concerning the mechanical plant:

Installation cost	\$3,600
Condensing water	2.5 gpm per ton
Electric power (consider it all varies with load factor)	35 hp (efficiency 90%)
Maintenance and repairs	\$50 per year
Refrigerant	\$22 per year

Using the above data, and the information from the plant officials as given in Prob. 3, except for the new water cost, determine the yearly over-all cost of the mechanical plant outlined above.

8. Based on the changed condensing water conditions as mentioned in Prob. 7, compare the steam-jet plant with the alternate of a mechanical plant.

9. A theater having a 400-hr cooling season can install air-conditioning plants as follows:

Plant A—Mechanical refrigeration, 50-ton capacity, installation cost \$12,000, operating cost averaging approximately 5.5 cts per hr per ton of capacity.

Plant B—Ice-activated plant, same capacity, installation cost \$7,000, operating cost averaging approximately \$4 per hr.

Assuming no operating engineer for either plant and assuming *total* fixed charges of 15 per cent of installation cost, which plant shows the lower *over-all cost* per year?

10. Why is evaporative cooling of air of almost no value in New York City and throughout the Eastern states in general?

CHAPTER XIII

AIR FOR HUMAN COMFORT

General. Early in the text we defined air conditioning. Remember that, in its broadest sense, air conditioning implies the control of various physical and chemical qualities of air, especially as it involves the control of temperature, humidity, air movement, and air purity. In this chapter we shall discuss the thermodynamics of the human body and the subject of human comfort as it relates thereto and as it relates to the conditions of the surrounding air. We shall then study a new type of temperature scale, called "effective temperature," and discuss standards to be maintained for indoor conditions in order to ensure human comfort. We shall consider air quality and the amount of fresh outdoor air that must be brought inside. Finally, we shall consider air movement and air distribution insofar as they relate to human comfort.

General Thermodynamics of the Human Body. The human body may be considered to be somewhat like a steam boiler or an internal-combustion engine. The engine or boiler burns fuel and gives out energy in the form of steam or mechanical work. The boiler or engine also discards a good deal of waste heat through the stack, exhaust pipe, and radiant heat thrown off. Similarly the body *burns* food as *fuel*. (Actually, this food is *oxidized* within the cells of the body by oxygen carried there by the red blood corpusles. Oxidation, you will recall, is the chemist's name for burning or combustion.)

Some of the energy produced by the burning food is used in doing mechanical work involved in physical exertion, like the boiler or engine. However, the human body is inefficient in its utilization of fuel, and waste heat is continuously discarded. Even when one is sleeping, doing almost zero work, the combustion of food continues (then at a slower rate), and since none of this energy is used for work, it must all be discarded in the form of heat.

For average adults seated and only slightly active physically, exposed to air at 70 to 80°F, the human body discards heat amounting to a total of approximately 10,000 Btu per 24-hr day. This amounts to about 400 Btu per hr. Under conditions of most extreme physical exertion, when the internal food-combustion rate is at a maximum, an adult person may discard total heat at the rate of 3,000 to 4,000 Btu per hr.

Heat is *always* being *discarded* by the human body. There is *never* a net heat flow *to* the body *from* the surrounding air. When we feel too warm, it means that we are having difficulty discarding the required amount of heat to satisfy the thermodynamic requirements. When we feel too cold, it means that our bodies are throwing away heat more rapidly than waste heat is being made available for discard. Hence to prevent people from being cold, an arrangement must be made by means of which their bodies cannot throw away heat so rapidly. To keep them from being too warm, it must be arranged that bodily heat can be thrown away more rapidly. Before building heating or air conditioning, we should recognize that clothing was the first step taken by primitive man to assist in the thermodynamic balance just discussed. The heavier the clothing, the greater the barrier against heat flow and, therefore, the slower the rate at which bodily heat is discarded.

Even in this day of air conditioning and universal central heating in residences and public buildings, clothing remains an important factor in regulating the rate at which heat is discarded by human beings. Consider two examples. (1) The type of clothing needed varies with exertion. You have probably seen an invalid wrapped in blankets on the porch in summer sunshine. You have also seen a hockey player wearing shorts with the thermometer at 20. Obviously, the hockey player is burning fuel very rapidly and hence must be able to discard the waste heat freely in order to be comfortable; conversely, the invalid is burning fuel slowly so that a barrier must be set up to prevent the rapid discarding of heat. A second illustration is that of the visitor from the tropics who wears a topcoat in New York City on a fall or spring day when the temperature is 65°. The New Yorker, on the other hand, would not consider wearing a tie and jacket when he finds the thermometer at 90° upon visiting his friend in the tropics.

Means of Discarding Bodily Heat. As you probably know, the heat thrown off by human beings consists partly of sensible heat and partly of latent heat. The latent heat is from two sources: (1) the evaporation of moisture from the surface of the skin and (2) the addition of moisture in the lungs to the air we breathe. Although this latent heat actually consists of a latent heat addition to the air, it serves to *remove sensible heat* from the body. Remember that sensible heat must be put into the water to vaporize it and that this sensible heat is furnished by the warm tissues.

With reference to the discard of bodily heat, we want to know three things:

1. How (by exactly what means) is the heat transferred from the body to the surroundings?

2. What affects the rate or speed of each particular means of heat dissipation?

3. To where in the surroundings does the heat go, and how does it affect the air in the room?

As might be expected from our discussion of the means of heat transfer at the end of Chap. V, heat may be transferred from the human body to the surroundings by means of radiation, conduction, and convection. Let us discuss each of these three means of heat transfer, briefly examining it with regard to the three points of interest in the preceding paragraph.

Radiation. The psychrometric conditions of the air and the clothing worn have relatively little effect on bodily heat lost by radiation. In fact, a man standing in a room with the air at 100°F (above body temperature), wearing an overcoat, can radiate heat to a cold (say 40°) wall or floor slab. Remember that radiant heat is a wavelike form of energy not entirely dissimilar to radio waves and light waves. Radiant heat travels even through a vacuum.

Radiant heat thrown off by human beings is converted into sensible heat upon striking the cold object to which it is radiated. Hence this radiant heat finds its way into the air of the room as sensible heat. We stated that our bodies are always discarding heat and never receiving it from the air and surroundings. By this we meant that the net result of all heat flow *to and from* the body is always away from the body. However, when seated next to a very warm object such as a radiator, radiant heat may flow from the hot object to the person. In this case, the person will then have to discard not only the normal waste heat, but also an amount of heat equal to this externally received radiant heat. In fact, this phenomenon is so definitely evidenced that normally clothed men have been made to feel too warm in a room where the dry-bulb temperature of the air was 40°F and the walls, floor, and ceiling were kept at 120°F.

In heating and ventilating, and air-conditioning work of good quality, we try to eliminate cold walls and floors during the winter season and we also try to eliminate the use of radiators very near places where people will be likely to sit or stand. In summertime we try to eliminate any "hot spots" in walls or floors from which heat might radiate to people. In air-conditioned spaces, radiation heat transfer from and to people is of less importance in summer than in winter. This is because building walls and floors, and especially windows, are more nearly at room temperature and skin temperature in summertime.

Conduction. People obviously lose heat when seated on cold furniture, leaning against cold walls, or being otherwise in contact with cold objects. Ordinarily, however, physical contact with objects in the room is limited to furniture, and upholstery and clothing combine to set up a fairly effective barrier against heat transfer by conduction to these objects. In addition, furniture is ordinarily at the same temperature as the dry-bulb temperature in the room. While there is of course some heat flow by means of conduction to articles such as furniture, the heat flow to the air in the room is of greater magnitude.

Heat flows from the exposed skin to the air in contact with the skin, and heat flows through clothing to the air in contact with the clothing, both of these heat flows being by conduction because of the contact between the warm and cold objects. Air is brought in contact with the skin and clothing so that this heat flow may take place by two means, *viz.*, convection currents and man-made air movements. (In the foregoing statement we assume that cold air drafts from around windows and doors are prevented.) Sensible heat flows only from the body to solid objects, the amount depending upon the temperature difference, the type of clothing, and the type of furniture. This sensible heat, upon warming the article of furniture slightly, then flows to the air, tending to raise the dry-bulb temperature of the air.

The heat given to the air passing over the skin and clothing is discussed in the section following.

Convection. Air currents over the skin and clothing are created by convection currents, as described at the end of Chap. V. In addition, in a room equipped with an air-conditioning or ventilating system, there is man-made air motion due to the air being supplied and removed from the space. These mechanical-made air currents probably are greater than the convection currents. This air motion serves to bring air in contact with the skin and clothing so that sensible heat flow may take place by conduction owing to the temperature difference. The faster the air currents, the more rapidly new air is brought in contact with the warm surface; and the colder the air, the greater the temperature difference. Both the speed of air motion and the air temperature affect the rate at which sensible heat is added to the air by this means.

The air passing over the surface of the skin picks up moisture which is evaporated from it. Furthermore, heavy perspiration will cause sufficient moisture to be absorbed by the clothing so that the clothing imparts moisture to the passing air. (This latent heat increase in the air results from sensible heat removed from the body.)

Finally, the air that is breathed is exhaled both warmer and higher in dew point. Therefore, the body discards both sensible and latent heat into the air which is breathed.

We have said that the speed of air motion influences the rate at which both sensible and latent heat are imparted to the air. The psychrometric conditions of the air are of prime importance in this respect because they not only influence the *rate* at which sensible and latent heat are given off, but also determine the *proportion of sensible heat* and the *proportion of latent heat* given off.

For instance, if the air is at a dry-bulb temperature of 100°, which is higher than the normal skin temperature, the temperature difference will cause sensible heat to flow *from the air to the skin.* Extremely heavy perspiration will be evidenced because all the normal waste heat of the body must be dissipated, *plus* the heat that flows from the air to the skin. *All this heat is dissipated by the evaporation of moisture from the skin and clothing.* Fortunately, people are equipped with a remarkable perfectly automatic thermostatic control mechanism. The heat flow described above occurs automatically. (In hot dry boiler rooms, 115°F and less than 20 per cent relative humidity, people perspire at a very rapid rate and yet neither the skin nor the clothing feels very damp. This is because the low relative-humidity air can pick up the moisture readily and evaporation is rapid. When one is "dripping with perspiration," it is not because the body is trying to do most of its cooling by adding latent heat to the air, but because owing to high relative humidity evaporation does not take place rapidly.)

When the dry-bulb temperature of the air is sufficiently low so that we can dissipate the required amount of heat, almost entirely by sensible heat flow, the automatic human thermostat mechanism reduces the perspiration rate to a minimum so that we shall not be chilled by dissipating heat at more than the required rate. The dissipation of heat, both sensible and latent, to the air that moves over the skin and clothing is probably the most important of the means of heat dissipation discussed, with regard to producing human comfort. This is true provided radiation losses are held to desirable minimums, and provided people do not sit down upon cold metal furniture, producing an unusually high heat loss by direct conduction.

Summarizing regarding this most important means of heat dissipation—the flow of sensible and latent heat to the air passing over the skin and clothing—we may make the following general statements:

1. With the air at a dry-bulb temperature lower than the surface temperature of the skin and clothing, some sensible heat is always flowing from the body to the air.

2. Some moisture is always evaporating (although the amount may be extremely small in some cases), cooling the body and adding latent heat to the air.

3. An increase in the speed at which air moves over the skin and clothing tends to increase both of the above means of heat dissipation.

4. As the dry-bulb temperature of the room air rises, reducing the amount of heat that can be dissipated by method 1 above, the perspiration rate automatically increases so that the heat dissipated by method 2 may be increased.

To help you understand the total heat dissipated by people engaged in different activities and the proportion of this which is sensible and latent heat, we present Table 1. "Metabolic rate" simply means the rate at which a person is utilizing and burning his food (fuel) and dissipating the energy from this combustion.

Table 2 is to some extent a condensed version of Table 1 with the type

TABLE 1.—RELATION BETWEEN METABOLIC RATE AND ACTIVITY*

Activity	Hourly metabolic rate for avg. person or total heat dissipated, Btu per hr	Hourly sensible heat dissipated, Btu per hr	Hourly latent heat dissipated, Btu per hr	Moisture dissipated, per hour.	
				Grains	Pounds
Average person seated at rest[1]	384	225	159	1,070	0.153
Average person standing at rest[1]	431	225	206	1,390	0.199
Tailor[2]	482	225	257	1,740	0.248
Office worker moderately active	490	225	265	1,790	0.256
Clerk, moderately active, standing at counter	600	225	375	2,530	0.362
Bookbinder[2]	626	225	401	2,710	0.387
Shoemaker;[2] clerk, very active standing at counter	661	225	436	2,940	0.420
Pool player	680	230	450	3,040	0.434
Walking 2 mph;[3,4] light dancing	761	250	511	3,450	0.493
Metalworker[2]	862	277	585	3,950	0.564
Painter of furniture[2]	876	280	596	4,020	0.575
Restaurant serving, very busy	1000	325	675	4,560	0.651
Walking 3 mph[3]	1050	346	704	4,750	0.679
Walking 4 mph[3,4]; active dancing, roller skating	1390	452	938	6,330	0.904
Stone mason[2]	1490	490	1000	6,750	0.964
Bowling	1500	490	1010	6,820	0.974
Man sawing wood[2]	1800	590	1210	8,170	1.167
Slow run[4]	2290	...	...	....	....
Walking 5 mph	2330	...	...	....	....
Very severe exercise[5]	2560	...	...	....	....
Maximum exertion different people[4]	3000–4800	...	...	....	....

*SOURCE: From "Heating, Ventilating and Air Conditioning Guide," Chap. 3, 1939.

Metabolism rates noted based on tests actually determined from the following authoritative sources:

[1] A.S.H.V.E. Research Laboratory.

[2] Becker and Hamalainen.

[3] Douglas, Haldane, Henderson, and Schneider.

[4] Henderson and Haggard.

[5] Benedict and Carpenter.

Metabolic rates for other activities estimated. Total heat dissipation integrated into latent and sensible rates by actual tests for metabolic rates up to 1250 Btu per hr, and extrapolated above this rate. Values for total heat dissipation apply for all atmospheric conditions in a temperature range from approximately 60 to 90°F dry bulb. Division of total heat dissipation rates into sensible and latent heat holds only for conditions having a dry-bulb temperature of 79°F. For lower temperatures, sensible heat dissipation increases and latent heat decreases, while for higher temperatures the reverse is true.

TABLE 2.—HEAT DISSIPATED BY AVERAGE ADULT PERSONS

Activity	Total heat dissipated, Btu per hr	Hourly sensible heat dissipated, Btu per hr	Hourly latent heat dissipated, Btu per hr	Moisture dissipated	
				Grains per hr	Grains per min
Average person seated at rest	400	220	180	1,210	20.2
Average person standing at rest	430	225	205	1,380	23.0
Office worker moderately active	490	225	265	1,790	29.9
Store clerk moderately active	600	225	375	2,530	42.0
Walking 2 mph, or slow dancing	750	250	500	3,380	56.3
Waiters, very busy	1000	325	675	4,560	76.0
Walking 4 mph, active dancing, active bowling	1390	450	940	6,330	105.5
Very vigorous exercise	2500	*	*	*	.....
Maximum exertion (varies)	3000–4800	*	*	.....	.....

* Proportion varies—to assume 35 per cent sensible and 65 per cent latent would not be badly in error. Values apply for atmospheric conditions of 65 to 85°F, with moderate humidities. Proportion of sensible and latent heat is correct for air with a dry-bulb temperature of 80°. For lower temperatures, the sensible heat portion increases and the latent heat portion decreases, and vice versa.

of activity described in slightly different words. In Table 2, certain of the technical activities shown in Table 1 are eliminated. The heat emissions per hour and the propositions of sensible and latent heat are changed slightly in Table 2, but are substantially the same. For your convenience, Table 2 has an additional column (6) giving the latent heat in grains of water per *minute.*

Concept of Effective Temperature. Suppose that you have determined certain indoor conditions of temperature, humidity, and air motions are that comfortable to average persons clothed in normal business clothes. From the foregoing discussion you can see that certain changes can be made within the room, keeping the same feeling of comfort. For instance, let us consider three such changes that might be made without causing discomfort to the occupants.

1. The dry-bulb temperature and the humidity can both be slightly increased, provided the air motion is increased enough so that the rate of dissipation of body heat remains the same.

2. We could increase the dry-bulb temperature, which would make it more difficult for sensible heat to flow from the people to the air, provided the relative humidity were decreased enough so that the increased speed of

evaporation of moisture could cause an increased cooling effect to compensate for the reduced sensible heat flow.

3. We could decrease the dry-bulb temperature and increase the relative humidity, which would be the reverse of 2.

In other words, the preceding paragraph shows that there can be various *different combinations* of dry-bulb temperature, wet-bulb temperature, relative humidity, and air motion, all of which, if carefully controlled, can result in the *identical feeling of comfort.* As far as people are concerned, "comfort temperature" or the temperature that they feel and are aware of would be the same for various conditions. Careful measurements have been made by the A.S.H.V.E. as to the different combinations of temperature, humidity, and air motion that result in the same feeling of comfort to 97 per cent of the people used in their tests. Since the feeling of comfort remained the same over a range of different temperature, humidity, and air-motion conditions, people subjected to these different conditions would be inclined to say that, as far as they were concerned, the temperature remained the same throughout. Therefore, a new artificial scale of temperature was developed by the society. This scale is called *effective temperature.*

This artificial new scale is really not a *temperature* scale at all; it really cannot be measured by an instrument. Although several devices are on the market, there is still no accepted instrument reading "effective temperatures in degrees." Instead, effective temperature is a measure of *the comfort effect on people of temperature, humidity,* and *air motion.*

Figure 1 illustrates the effective temperature chart of the A.S.H.V.E. The air-velocity range from 20 to 700 ft per min, and also the range of dry-bulb and wet-bulb temperatures shown, is much wider than the range of temperatures that are comfortable to human beings. Note that the effective temperature (abbreviated "° E.T.") lines on this chart go from 30 to 110°E.T. There is a guideline *AB* shown on Fig. 1. If you wish to find the effective temperature on Fig. 1, draw a line similar to *AB* between the dry-bulb temperature and the wet-bulb temperature of the air you have. Run your pencil along line *AB* until it crosses the air-velocity line corresponding to the velocity of your air. At this intersection point, read the effective temperature on the sloping effective-temperature lines. Note that at lower effective temperatures air velocity is a greater factor in determining effective temperature than it is at higher ranges of effective temperature. In fact at effective temperatures above 100°E.T. an increase in air velocity increases the effective temperature rather than decreases it, for a given dry-bulb and wet-bulb condition.

The chart in Fig. 1 is of interest and value in understanding the concept of effective temperature. However, we are interested only in a small portion of this chart as we are not interested in effective temperatures below 60°E.T. or above 80°E.T. Furthermore, as we shall see later in this chap-

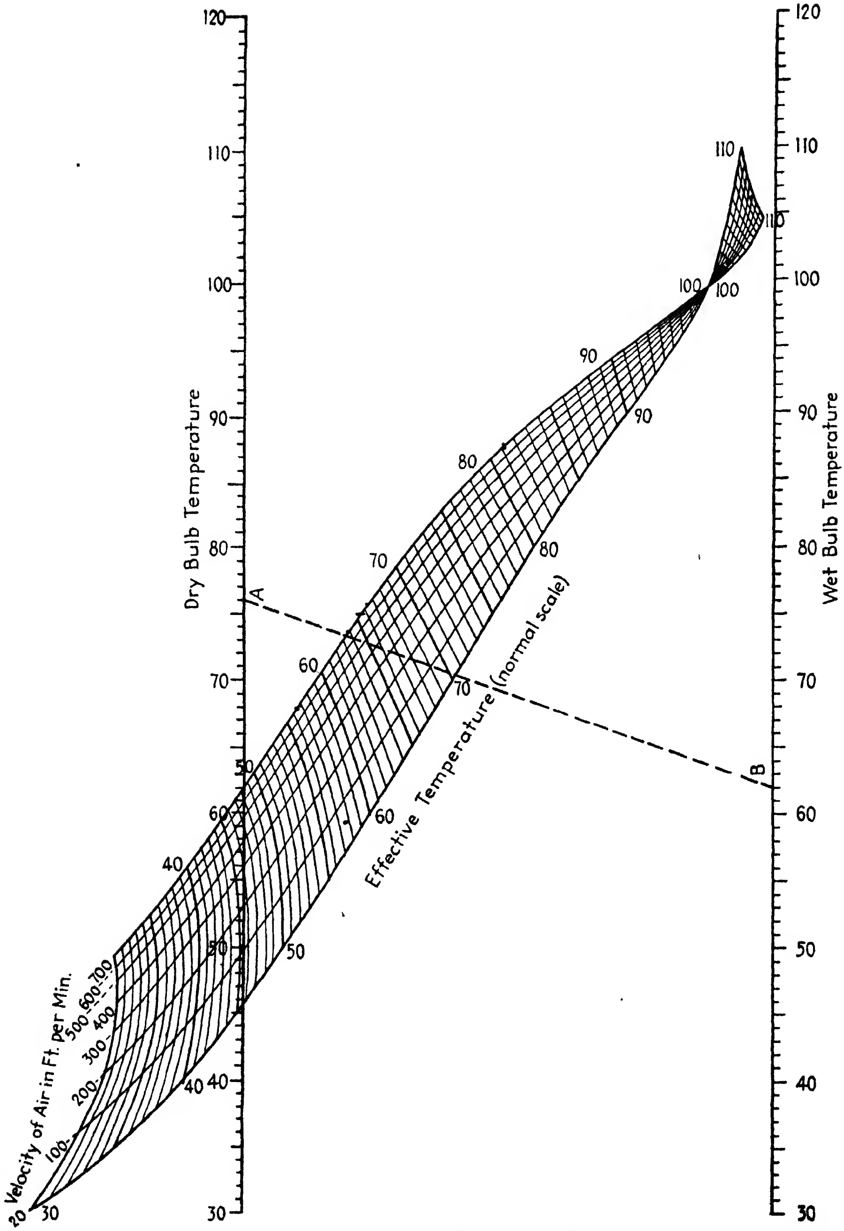


FIG. 1.—Effective temperature chart. (*Courtesy of A.S.H.V.E.*)

ter, air velocities as high as 100 fpm are likely to be objectionable during the winter season in a ventilating or air-conditioning installation. Finally, the effective-temperature chart in Fig. 1 does not show relative humidity. It is possible to plot effective-temperature lines on an ordinary psychrometric chart, the form of chart shown on page 133 being well suited for this purpose. The A.S.H.V.E. Research Laboratory has plotted effective-temperature lines in this manner on different charts, each chart being for a different air velocity.[1] Since for human comfort we are interested in conditions involving an extremely slight air motion—15 to 25 fpm—we are interested only in an enlargement of the central portion of the particular chart drawn for this low air velocity. Such a chart is the A.S.H.V.E. comfort chart, which has become an almost universal chart throughout the heating, ventilating, and air-conditioning industry. This chart is so familiar to most air-conditioning engineers that they can sketch a fairly exact copy from memory on the back of an envelope in a very few minutes. (The reason the comfort chart is based on still air will be made clear under Air Movement later in this chapter.)

The note to Fig. 2 explains that the effective temperature lines shown in heavy black on the figure and labeled "average winter comfort lines" and "average summer comfort lines" may vary somewhat depending upon the type of building, the length of occupancy, and the type of clothing worn. Therefore, let us now study the various factors that would influence a change in the best effective-temperature line for comfort in both summer and winter season, after which we shall set up approximate standards for indoor comfort conditions in summer and winter.

The weight of clothing worn obviously will affect the comfort sensation of people not in heated or air-conditioned spaces. For instance, at an indoor hockey arena where spectators wear their overcoats (and where too warm air would impair the quality of the ice) the average winter comfort effective temperature of 66°E.T. would obviously be much too high. On the other hand, in a ballroom where women customarily wear formal evening dresses, a higher effective temperature than 66°E.T. might be required during the winter season. In summer air-conditioned space where all men wear coats and ties and all women wear stockings and dresses with sleeves, a lower effective temperature is indicated than would be needed in the dining room of a resort hotel where both men and women might be expected to wear less clothing.

Radiant heat to or from walls or to or from exposed radiators affects the best average comfort temperature in both summer and winter. In winter, large areas of exposed radiators permit a lower effective temperature of the air with the same feeling of comfort to the individual. Remember

[1] Students interested in investigating this further are referred to the A.S.H.V.E. "Heating, Ventilating and Air Conditioning Guide," Chap. 3, Figs. 2, 3, and 4, 1939.

that the chart is based upon rooms heated by convection type of heating (or cooled by similar systems) so that the best conditions, as shown on the chart, do not show the influence of large exposed radiator surface. In summer, if arrangements are made to pump chilled water through the radiators used for winter heating, a higher effective temperature of the air in the room may be permitted than is indicated on the chart.

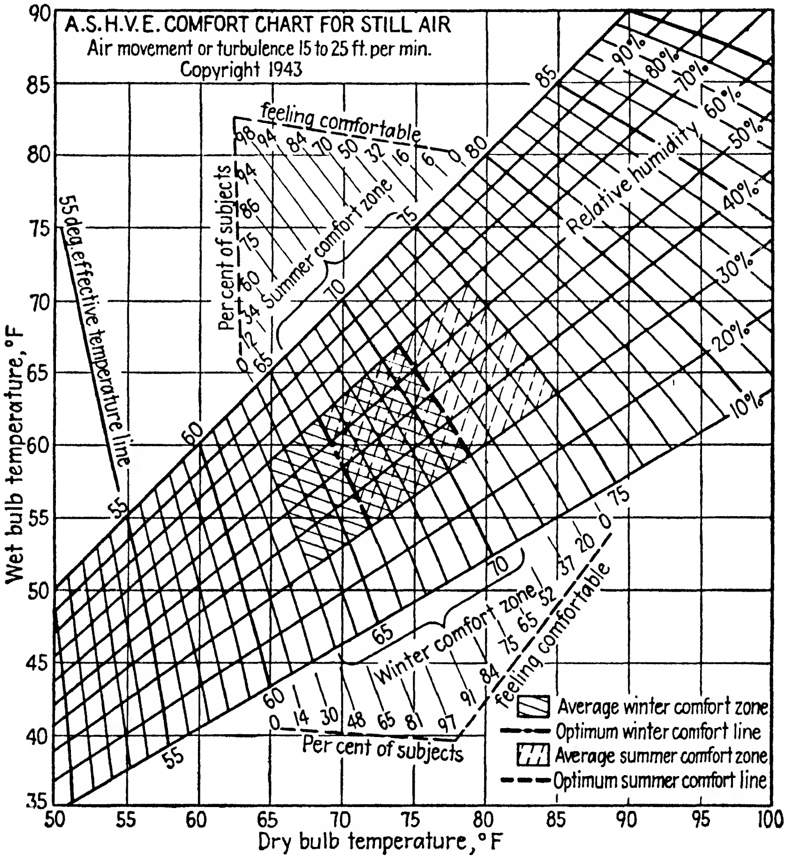


FIG. 2.—A.S.H.V.E. comfort chart for still air. (*Courtesy of A.S.H.V.E.*)

Outdoor temperature also affects the choice of the effective temperature to be maintained indoors. This factor of outdoor temperature is closely related to another factor, *viz.*, the time that most occupants spend within the hot or cold space. The comfort chart is based upon the assumption that people spend the entire day within the air-conditioned space, so that outdoor conditions may be almost completely forgotten. The outdoor conditions may not be entirely forgotten, even though the people spend the entire day within the conditioned space, as the chart shows the ideal summer effective-temperature line is 5°E.T. above its best winter effective-temperature line.

In winter, owing to trips outdoors into the cold weather and the fact that most people sleep breathing substantially outdoor air, the metabolic rate of the average person becomes adjusted higher in winter than in summer. This means that people can be comfortable *while dissipating more heat* from their bodies in winter and, hence, can be comfortable in cooler surroundings. This effect is noticeable even though people spend most of their indoor working hours in air-conditioned or heated space. It is upon this basis that the 66°E.T. line has been determined as the ideal indoor winter condition for space such as homes, schools, and public buildings where people enter and stay ordinarily for at least several hours.

The winter effective-temperature line may be moved downward to 63°E.T. or even to below 60°E.T., for space where people only enter for a short time and in which people remain standing and somewhat active. In the case of special rooms such as hospital rooms, it is sometimes necessary to maintain winter effective temperatures as high as 70°E.T.

In the case of summer air-conditioning installations, a similar situation exists. When the outdoor temperature is above 90, indoor conditions of slightly above 80 will seem comfortable to people entering the space and remaining for short periods up to perhaps half an hour. On the other hand, when the outdoor temperature is between 80 and 85°, the indoor conditions must be below 80° dry bulb even for people entering the conditioned space for short periods. Our comfort chart in Fig. 2 shows a 66°E.T. which, depending upon relative humidity, would be 74 to 79° dry bulb. At 50 per cent relative humidity, the dry-bulb temperature would have to be about 76.5° to give an effective temperature of 71°E.T. This 71°E.T., as a comfort ideal, is based upon people entering the air-conditioned space and remaining in it throughout the day or evening, the same as people remain inside heated buildings in the wintertime. (At the conclusion of this discussion of the factors affecting the indoor design conditions, approximate standards for these conditions will be given.)

Variations among individuals as to the effective temperature producing perfect comfort is a factor in both winter and summer. Only about 3 per cent of an average group of people of 100 or more should manifest discomfort due to their individual peculiarities if effective temperatures conforming to those in Fig. 2 are used. For these few individuals, the only correction to ensure comfort must be a correction of the rate at which bodily heat is dissipated by heavier or lighter clothing. Individual variation from the standard comfort conditions is slight in winter and considerably greater in summer. (This statement is true in spite of the "per cent of subjects feeling comfortable," shown as 97 for the winter comfort line and 98 for the summer comfort line in Fig. 2.) The reason is that in winter most of the dissipation of body heat is by sensible heat flow to the air. In summer with the higher dry-bulb temperatures even in air-conditioned spaces, the greater portion of the heat is dissipated in the form of latent heat added to the air from the evaporation of moisture. The variation from one individual to another seems more pronounced with regard to this latent heat dissipation. As you probably know some persons perspire more readily than others. It is usually true within reasonable limits that persons who perspire readily (not excessively) find less discomfort in warm weather than those persons who do not perspire but instead become flushed and hot when subject to warm air.

Finally from the standpoint of manufacturers and contractors in the air-conditioning business, there is one important factor that influences the inside comfort conditions upon which the design of an installation shall be based. This factor consists of a combination of the *desires* and the *budget funds* that the *purchaser* contributes toward the selection of his installation. There was a time when theater owners and operators insisted upon air-conditioning plants that could maintain the theater at 72° dry bulb regardless of the outdoor temperature. It is true that these plants frequently had indoor relative humidities as high as 90 per cent. It is also true that many patrons of such theaters objected to the cold clammy indoor conditions. Nevertheless, purchasers of many theater installations demanded and offered to pay for these results so that suppliers and contractors were forced to put in these installations, although they knew that it was not the best thing to do. Sometimes a limited budget will mean not only that a higher indoor effective temperature must be tolerated on a summer conditioning installation, but also that the fresh outdoor air requirements (presented later in this chapter) cannot be adhered to because of cost. In this case, the customer should be made to realize that his limited budget is reducing the quality of his proposed installation, in which case additional money sometimes will be appropriated.

Approximate Design Standards as to Effective Temperatures—Winter. The normal occupancy of buildings in the wintertime conforms to the basis on which our comfort chart has been made. Therefore, 66°E.T. may be taken as the indoor design conditions for winter installations where the normal occupancy period is several hours or more. For installations where the indoor relative humidity is not artificially maintained at 25 to 50 per cent R.H., the indoor relative humidity may be lower than 10 per cent. Note that this involves dry-bulb temperatures of 74 to 75° in order that the effective temperature may be 66°E.T. If, on the other hand, the relative humidity can be maintained at 40 to 50 per cent without condensation of moisture on cold windows, dry-bulb temperatures of 70 to 71° will produce an effective temperature of 66°E.T. Always keep in mind that the air motion must not exceed 25 fpm, or people will complain of drafts and "insufficient heat," even though the temperature and humidity are correct.

Approximate Design Standards as to Effective Temperatures—Summer. Indoor design conditions are broken, for convenience, into two classes: buildings where the average occupancy is less than 3 hr and buildings where the occupancy is more than 3 hr. Obviously, for a building such as a theater where the occupancy is almost exactly 3 hr, a compromise condition midway between the two given below may be selected. In New York and other cities having similar outdoor conditions, the indoor condition of 75°E.T. is generally used for buildings where the average occupancy is less than 3 hr, this indoor condition to prevail when the outdoor conditions are 95° dry bulb 75 to 77° wet bulb. With less severe outdoor conditions, the indoor dry-bulb temperature is reduced somewhat as indicated in Table 3. Table 3 is not as yet a definitely accepted standard, but may vary depending upon the wishes of the purchaser and his engineer and architect if they are engaged in the project. Relative humidity and effective temperature are not mentioned in Table 3, the reason being that the relative humidity will be kept as low as possible, but may rise above 50 per cent, and this relative humidity cannot always be exactly controlled. Hence, for outdoor conditions lower than the minimum, operating practice has been to endeavor to hold the dry-bulb temperature approximately according to Table 3 and to endeavor to hold the relative humidity as low as possible, the resulting effective temperature being determined by how low the relative humidity can be held.

TABLE 3.—DESIRABLE INDOOR DRY-BULB DESIGN CONDITIONS FOR SUMMER AIR CONDITIONING AT VARIOUS OUTDOOR DRY-BULB TEMPERATURES BASED UPON EXPOSURES OF LESS THAN 3 HR

Outdoor dry-bulb temperature, °F.	95	90	85	80	75	72
Indoor dry-bulb temperature,* °F	82	80	77	75	74	73
Indoor dry-bulb temperature,† °F	80	78	76	74	72	72

* This set of conditions is recommended where the indoor relative humidity can be kept always at 50 per cent relative humidity or below.

† This set of conditions is recommended where the indoor relative humidity is to be kept at 50 per cent and may rise to 65 per cent relative humidity.

From the foregoing you can make the following deductions: If a large office building, such as the Metropolitan Life Insurance Building in New York City, which is completely air-conditioned and in which the employees remain throughout the day including lunch hour, is to be air-conditioned at 71°E.T., as indicated in the comfort chart, the indoor conditions will be too cold for visitors entering the building for short periods. On the other hand, a department store if air-conditioned at 75°E.T. for the benefit of the customers who are in the store an average of about 1 hr will be uncomfortably warm to employees after they have been in the store 3 to 4 hr. These deductions are perfectly true, and nothing can be done about this situation. In the case of visitors calling at an air-conditioned office, they are likely to find the conditioned space too cold upon entering and to suffer a noticeable shock upon leaving it for the street. Very little can be done about this situation. As to the employees of the department store, they must, by lightweight clothing, adjust themselves as best they can to the fact that the indoor condition is a higher effective temperature than would be ideal for their all-day occupancy.

In buildings in which there is almost complete occupancy all day, especially where the employees eat luncheon in an air-conditioned cafeteria in the building, it has sometimes been found necessary to maintain the indoor effective temperature as low as 69 or 70°E.T. This has been necessary to avoid complaints from the employees, the complaints usually having been made between 11 a.m. and 1 p.m. after the employees had been in the building for several hours. When the indoor temperatures were dropped in response to such complaints, an idea was conceived for a means of dropping this temperature automatically by a time-clock mechanism which would change the setting of the thermostats. Recently, consideration has been given to the idea of having a time clock mechanism raise the setting of the thermostats so that the indoor temperature will *rise*, during the last hour of the business day, to as high as 76°E.T., to reduce the shock when employees leave the conditioned building for the hot street.

Regarding the effect of relative humidity on effective temperature in both summer and winter, it is important to note considerations regarding practicability and cost involved in having ideal indoor relative humidity. From the comfort chart you can see that relative humidities from 30 to 70 per cent relative humidity are considered within the comfort zone. Relative humidity at 40 to 50 per cent is considered ideal for human comfort and health. Adding humidity in the winter is not expensive, and humidity can be added until the *practical limit* is reached. Since most buildings today have single glass windows, the *practical limit* of winter indoor humidity is reached when the indoor dew point closely approaches the temperature of the inside surface of windows. Humidification beyond this point results in condensation of water on windows. Therefore, it may be possible to maintain the ideal winter indoor relative humidity with mild outdoor temperatures as much as 35 per cent. When the outdoor temperature drops to zero, it probably will not be possible to maintain indoor relative humidities even as high as 30 per cent in buildings with single glass windows. In summer there is no practical limit (comparable to the condensation problem in winter humidification) that prohibits lowering the indoor relative humidity below any set point. Instead it is the *cost factor* that limits lowering the relative humidity in the summer. Dehumidification, as you will learn in a later chapter on psychrometric work, becomes extremely costly when refrigeration is used for *dehumidifying alone*. Therefore, indoor relative humidities, especially in the milder summer weather, are often allowed to go above 60 per cent, although it would be better to have them below 50 per cent.

Geographical Modifications. Most of the present comfort-chart information is based upon persons in the northeastern part of the United States and in those portions of the Middle West and Far West where the outdoor conditions are similar to those in the Northeast. Outdoor conditions in the Southwest are much higher in dry-bulb temperature, but extremely low in relative humidity. Most of the time the southeastern part of the country is higher both in dry bulb and in dew point. The residents of areas differing in outdoor conditions from those prevailing in the northeastern section of the country require slightly different indoor comfort conditions.

For the winter season very little difference in the indoor conditions is evidenced. For sections here the prevailing outdoor winter temperatures are from 30 to −30°F, the inhabitants become so accustomed to the severe winter cold that indoor effective temperatures slightly below 66°E.T. are permissible. In summer the inside design varies markedly, depending upon the prevailing outdoor conditions. Table 4 gives information that may be used in a manner similar to that in Table 3. It differs slightly from the data in Table 3, but you will recall that we said that this information had not yet been standardized in a manner meeting with the approval of all concerned. Table 4, however, is of primary usefulness in considering the outdoor dry-bulb temperatures shown in the table as a *maximum outdoor summer* dry-bulb temperature in *different geographical locations*. Hence if the maximum outside design dry bulb is 100° for certain cities in Texas, the inside conditions to be maintained, with 100° dry bulb outdoors, are given in Table 4. With less severe outdoor conditions, the indoor effective temperature or at least the indoor dry-bulb temperature should be reduced, as indicated in Table 3.

Air Quality and Quantity. The A.S.H.V.E. defines ventilation as "the process of supplying or removing air by natural or mechanical means to or from any space. In most air-conditioning installations, whether winter or summer, a large percentage of the *total* conditioned air supplied to the conditioned space is *recirculated air* that has been removed from the conditioned space, reconditioned, and then returned to the conditioned space. Recirculated air should never be taken from kitchens or bathrooms. These spaces should be equipped either with a grille leading to the outdoors through which the air supplied to the room will leave or, better still, with an exhaust fan to remove air and prevent air from leaking to other rooms whence it might be recirculated. Though recirculated air may be made ideal as to temperature and dew point, may be thoroughly purified and have its oxygen content and CO_2 content adjusted to the same as outdoor air, some mysterious phenomenon occurs and the 100 per cent recirculated air becomes dead or "vitiated." One hundred per cent recirculated air, provided it is treated as outlined above, has not been shown to have any adverse effects from a health standpoint. On the other hand, air that has been mechanically handled and recirculated can instantly be detected by persons as "stale air."

TABLE 4.—DESIRABLE INSIDE CONDITIONS IN SUMMER CORRESPONDING TO OUTSIDE TEMPERATURES*

(Occupancy 40 min to 3 hr)

Outside dry bulb, °F	Eff. temp	Dry bulb °F	Wet bulb °F	Dew point °F	Relative humidity, per cent
100	75	82	67	59	45
	75	80	70	65	60
95	74	82	64	53	36
	74	80	67	60	51†
	74	78	70	66	68
90	73	80	64	54	41
	73	78	67	61	56
85	72	79	63	53	41
	72	77	66	60	56
80	71	77	63	54	45
	71	75	66	61	61

* Applying to active office workers seated at work.

It is not yet definitely known what properties of air give it the tingling freshness of outdoor air. It is thought that the matter may be largely psychological since instances are known where employees in research laboratories refused to remain in closed test rooms even though the air was psychrometrically perfect and had the proper oxygen content and was free from odors. The question of stale air has a definite *physiological* basis, however, because almost all persons can detect so-called "stale air," even if this air is psychrometrically and chemically perfect.

Even though vitiated air has been recently shown (contrary to prior beliefs) *not* to have a harmful effect upon health, even if the oxygen content is somewhat reduced, it has been shown that vitiated air could cause a loss of appetite and a definite feeling of lassitude (tiredness). Measurements made of fresh air and vitiated air have shown that the vitiated air had a smaller number of positive and negative ions than the fresh air.[1] (Ions are small electrically charged particles that exist in gases and in liquid solutions.) Methods have been tried for revitalizing vitiated air by means of ionizing machines and introduction of small quantities of ozone (O_3). No accepted methods of improving air in this manner have yet been developed. Medical opinions regarding ozone vary, some thought being that 0.01 part of ozone per million parts of air are harmful, while other authorities claim that 10 to 12 parts of ozone per million parts of air are not harmful.

The problem of ventilation then is to bring in a proper quantity of fresh outdoor air so that the air in the conditioned space will be healthfully fresh and, even more, *fresh enough to be appealing to people.* In buildings without complete duct systems, ventilating, or air conditioning, where the heating is provided by standard radiation (ordinary radiators), air leakage ordinarily provides sufficient air for ventilation purposes. Ordinarily, especially in new buildings or buildings where the construction is so good that air leakage is minimized, the only dependable way of securing proper ventilation is to introduce the fresh air by positive mechanical means. This system involves the use of fans and ductwork. Tables 5 and 6 give information regarding the quantity of fresh air to be supplied to conditioned spaces having different types of occupancy. These tables also cover different types of installations, *viz.*, heating without complete conditioning, heating with conditioning, and complete summer conditioning. Table 6 gives approximate tentative standards for types of spaces and conditions not covered in Table 5 and is based upon experience with a limited number of installations. The air space per person per cubic foot varies widely, and the selection of the fresh air quantity is one involving personal judgments; cost influence each case.

Air Movement and Air Distribution. Absolutely stagnant air, no matter how ideal its psychrometric and chemical conditions, is undesirable and has, to some extent, the same unpleasant effect upon people as vitiated air. To avoid the difficulty of stagnant air, the ventilating or air-conditioning systems should have air-supply grilles and return-air openings so located that *slight* air motion is produced throughout the conditioned space at least at a level of 3 ft above the floor to 6 ft above the floor. In winter, when air in certain spots might be 2 or 3° below the desired winter comfort condition of 66°E.T., a study by the A.S.H.V.E. Research Laboratory has indicated that objectionable drafts are likely to occur if air currents of 40 fpm are

[1] Yaglou, C. P., L. C. Benjamin, and S. P. Choate, Changes in Ionic Content in Occupied Rooms Ventilated by Natural and Mechanical Methods, *A.S.H.V.E. Research Rept.* 921 (*A.S.H.V.E. Trans.*, vol. 38, p. 191, 1932).

permitted.[1] Therefore, the air motion of 15 to 25 fpm per min, upon which our comfort chart in Fig. 2 is based, has been selected as the desirable

TABLE 5.*—MINIMUM OUTDOOR AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS

(Provisional values subject to revision upon completion of work)

Type of occupants	Air space per person, cu ft	Outdoor† air supply, cfm per person
Heating season with or without recirculation. Air not conditioned		
Sedentary adults of average socioeconomic status.......	100	25
Sedentary adults of average socioeconomic status.....	200	16
Sedentary adults of average socioeconomic status.....	300	12
Sedentary adults of average socioeconomic status.....	500	7
Laborers..	200	23
Grade-school children of average class................	100	29
Grade-school children of average class................	200	21
Grade-school children of average class................	300	17
Grade-school children of average class................	500	11
Grade-school children of poor class..................	200	38
Grade-school children of better class...........	200	18
Grade-school children of best class...........	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gal per hr. Total air circulation 30 cfm per person		
Sedentary adults..................................	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person		
Sedentary adults..................................	200	<4

*SOURCE: YAGLOU, C. P., E. C. RILEY, and D. I. COGGINS, Ventilation Requirements, *A.S.H.V.E Research Rep.* 1031 (*A.S.H.V.E. Trans.*, vol. 42, 1936).

† Impressions upon entering room from relatively clean air at threshold odor intensity.

velocity. This velocity is enough to prevent undesirable stagnation of air, but is not so high as to cause drafts.

Slightly different standards should be set for air motion in summer. For

[1] HOUGHTEN, F. C., H. H. TRIMBLE, CARL GUTBERLET, and M. F. LICHENTFELS, Classroom Drafts in Relation to Entering Air Stream Temperature, *A.S.H.V.E. Research Rep.* 1016. (*A.S.H.V.E. Trans.*, vol. 41, p. 268, 1935).

buildings equipped only with ventilating systems—and there are many of these, both public buildings and most of the new residences equipped with winter air conditioning—the indoor temperature in summer is likely to be about the same as the outdoor temperature. Under these conditions,

TABLE 6.—FRESH OUTDOOR-AIR REQUIREMENTS FOR ADEQUATE VENTILATION TO REMOVE ODORS AND STALENESS

Type of occupants	Air space per person, cu ft	Outdoor air supply, cfm per person
Heating season. Air conditioned in air washer.‡ Total air circulation of 30 cfm per person		
Seated adults, no food or smoking*	200	12
Seated adults, food and smoking†	100	20
Seated adults, food and smoking†	200	16
Adults seated and dancing, bar, food, and smoking†	200	25
Summer season. Air cooled and dehumidified in washer‡ with water renewed daily. Total air circulation 30 cfm per person		
Seated adults, no food or smoking*	200	5
Seated adults, food and smoking†	100	15
Seated adults, food and smoking†	200	10
Adults, seated and dancing, bar, food, and smoking†	200	20

* Conforms to tentative standards A.S.H.V.E. See A.S.H.V.E. "Guide," Chap. 3, p. 69, 1939.

† Average values applying to restaurants, night clubs, etc. Will vary with special conditions. Can be *slightly* decreased if cost dictates; should be increased for best results as much as cost budget will permit.

‡ These air quantities are based upon the practice of changing air-washer water every day. If this is not done, or if cooling coils are used instead of an air washer, it may be necessary to increase the fresh air quantities to obtain the desired conditions.

velocities as high as 100 fpm have been used in our experience without complaints as to drafts. In complete summer air-conditioning jobs, velocities higher than 40 fpm may be used. However, reference should be made to Fig. 1, and the effective temperature should be calculated on a basis of *actual* air velocity. Furthermore, care must be exercised that the air velocities must be uniform, as persons subject to local high-velocity drafts will invariably complain.

Air Movement and Air Distribution. Tests may be made for the velocity of air currents within a room using either an *anemometer* or *velometer*. Ordinary anemometers are of no value with velocities below 100 fpm, although low-speed anemometers are available. After you become familiar with the use of ordinary anemometers and the velometer, you will realize that the velometer makes an ideal instrument for measuring these low-velocity air currents.

Air for Human Comfort. We hope that the foregoing discussion has made clear the complexity of the problem of providing correctly conditioned air in summer and winter. Several of the phases of the discussion in this chapter will be amplified in further chapters. For instance, the problem of air distribution within the conditioned space is a problem of itself. It should be obvious to you that air must leave supply grilles at a much higher velocity than 25 fpm, and yet we have stated that we usually do not want to exceed this velocity at the breathing level. Also air must be distributed within a conditioned space so that there are no "dead pockets" left in the conditioned space where air remains still and does not find itself replaced with conditioned air from the supply outlets. From this you can see that the subject of air distribution requires considerable amplification in a later chapter.

PROBLEMS

1. State briefly the comfort conditions of air, with regard to its psychrometric properties and the degree of air motion desired, which will provide the most comfortable conditions in all cases.

2. Are the chemical and psychrometric conditions of air which are best for health also best for comfort, or should people be willing to endure conditions not ideal for comfort in order to gain the maximum advantage from a health standpoint?

3. In the winter we have often heard the suggestion, "Do come inside and get warm." Does this mean that when we go indoors in winter there is a net input of heat from the warm indoor air to the body?

4. The human body gives off heat, adding both sensible heat and latent heat to the surrounding air. What affects the rate at which latent heat is given off? (Exclude heat radiated from the body and exclude heat given off by conduction, by sitting on cold furniture, etc.)

5. Suppose a landlord owns two houses, one of which has provision for humidification and is maintained at 35 per cent relative humidity in wintertime. The other house has no humidifying equipment, and the relative humidity has been measured and found to be usually below 10 per cent relative humidity. In order that people may have the same feeling of comfort in both houses, *viz.*, 66°E.T., what must the dry-bulb temperatures be in the two houses? (Assume air motion of 20 fpm in both houses.)

6. A customary summer inside design condition in New York City is 80° dry bulb, 67° wet bulb, 50 per cent relative humidity. A usual winter design condition is 71° dry bulb, 57° wet bulb, 40 per cent relative humidity for a building having double windows and provision for humidification. From Fig. 2 find the effective temperature produced by these two design conditions. (Assume air motion of 20 fpm.)

7. Take the two sets of design conditions from Prob. 6, and for each set draw a line such as line *AB* on Fig. 1. Read on Fig. 1 the effective temperature at an air velocity of 20 fpm, check the effective temperature figures you obtained in Prob. 6, using Fig. 2. (Your figures should substantially agree.)

8. On the lines you draw in Prob. 7, get the effective temperatures at an air velocity of 200 fpm. Compare these effective temperatures to the effective temperature you

determined in your answer to Prob. 7. For both the summer and winter design conditions, how much lower is the effective temperature at the higher air velocity? The text states that air velocity is a greater factor in producing a feeling of coolness under winter design conditions than under summer design conditions. Do your answers in this problem confirm or disprove this general statement in the text?

9. Architects and promoters are planning a Glorioso Casino, a new Broadway "night spot." There is to be a dining room known as the "Orchid Room" and a room having a small dance floor, etc., to be known as the "Glitter Room." You have design information as follows regarding the two rooms:

	Orchid Room	Glitter Room
Total number of patrons	300	400
Patrons seated at rest	300	0
Patrons as active as officer workers, including those seated at bar	0	200
Patrons actively dancing	0	200
Waiters not busy	30	0
Waiters very busy	0	25
Bartenders very busy	0	5
Smoking?	Yes	Yes
Bar?	No	Large rotary
String orchestra	6 men	0
Dance orchestra	0	12 men

From the information in Tables 5 and 6, determine the fresh air in cubic feet per minute required for each room for the winter season, based upon using an air washer, the water of which is changed daily, and based upon a total air circulation of 30 cfm per person. Assume approximately 200 cu ft of space per person in each room, based upon outdoor air at 65° dry bulb, 54° wet bulb, and a specific volume of 13.3 cu ft per lb. How many pounds per minute of fresh air is required for each room?

10. On a day when the outdoor air is 65° dry bulb, 54° wet bulb (specific volume 13.3 cu ft per lb), it may be possible to operate the air-conditioning system described in Prob. 9, using 100 per cent fresh air (no recirculated air), not operating the air washer. The duct work as designed has a capacity of approximately 13,000 cfm for the Glitter Room duct work and 3,500 cfm for the Orchid Room duct work.

a. From Tables 1 and 2 calculate the total heat, the sensible heat in Btu per hour, the latent heat in Btu per hour, and the latent heat in grains of water per minute, added to the air in each of the two rooms by dissipation of body heat.

b. Based on the psychrometric conditions for the air given in the problem, determine the pounds per minute of air supplied to each room.

c. Find the dew point and relative humidity of this outdoor air supplied to the rooms.

d. Assuming that the body-heat load is the *only* heat load affecting the air supplied, and assuming that *all* the body heat load goes to increase the sensible and latent heat of the air, find the dry bulb, dew point, and relative humidity that will exist in each of the rooms. (Each minute the number of pounds of air you figured in *b* will enter the room at the dry bulb and dew point you figured in *c*. This air will have the sensible and latent heat you calculated in *a added* to it. The result of this addition gives you the answer to *d*.

CHAPTER XIV

HEAT TRANSMISSION AND AIR LEAKAGE

General. In our previous discussion of heat flow we learned that heat will flow from the warmer of two bodies to the cooler body. When we have warm air inside a building and cold air outside the building, heat tends to flow from the warm air to the cold air, in spite of the fact that the building wall constitutes a barrier making this heat flow somewhat difficult. Under the opposite circumstances, when we are cooling the interior of a building below the outdoor temperature, heat tends to flow inward through the walls of the building. The *heat flow* mentioned above is entirely independent of any actual leakage of warm air through the walls. As you know, a steel wall would not permit the leakage of any air, yet heat would flow right through the steel wall.

Heat, as you know, will flow more readily through a single sheet of plate glass than it will through a thick brick wall. You also know that heat will flow more readily through the "paper-thin" wall construction of a summer cottage, having no inside wall finish, than it will through the walls of a modern insulated house.

In this chapter we shall learn how to calculate the flow of heat that will occur through various walls of different construction, based upon the number of degrees of temperature difference from one side of the wall to the other. We also shall learn how to calculate air leakage through walls and around windows and doors, so that we can determine the number of cubic feet of air that will leak into a given space in a certain length of time. Note that in the preceding two sentences no mention was made of *summer or winter* seasons. After having learned the method of calculating heat flow and air leakage in this chapter, we have a basic knowledge with which to learn the calculation of building heating loads and building cooling loads in two later chapters.

Heat Flow through a Wall. Ordinarily building walls are built of three to six different substances in layers, with one to three air spaces between the various layers of solid material. In the simplest case, however, let us consider a wall built of a solid uniform material having no airspaces.

For this example, let us take a 6-in. thick concrete wall, as illustrated by the three sketches in Fig. 1. Heat would flow through the wall by the three different methods—radiation, conduction, and convection—that we studied in connection with heat transfer. Sketch *a*, Fig. 1, illustrates the part that radiant heat plays in heat flow through our wall. Radiant heat impinges (strikes upon) on the warm side of the wall from the warm air and from any source of heat on that side of the wall. This action is illustrated by the arrows leading *from* the dots *to the wall* in sketch *a*. This heat then flows through the wall by conduction, as illustrated by the broken line arrows drawn through the interior of the wall. Upon reaching the cold side of the wall, the heat we are tracing warms the surface of the wall somewhat. This warm surface radiates heat to the cold medium to the left, as indicated by the arrows leading *from the dots* on the cold surface of the wall.

Sketch *b*, Fig. 1, illustrates the flow of heat that is imparted to the warm side of the wall by *conduction* from the warm air on that side of the wall.

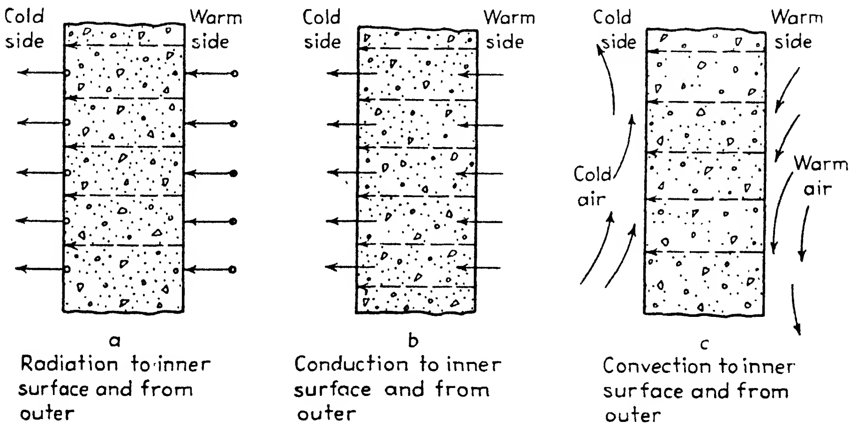


FIG. 1.—Heat transfer through solid homogeneous wall.

Upon entering the wall, this heat flows through the wall itself by conduction exactly the way heat flowed through the concrete in sketch *a*. This conduction flow is illustrated by the same broken arrows in sketch *b*. Sketch *b* illustrates this heat being given from the wall to the cold air outside the wall by conduction. Heat transfer, as illustrated in sketch *b*, would occur if the air on both sides of the wall were *absolutely motionless*.

Sketch *c*, Fig. 1, illustrates heat transfer, *in addition* to that illustrated in sketch *b*, which would occur provided *air motion* were permitted on both sides of the wall. Warm air having imparted some of its heat to the wall on the warm side would be itself *cooled* and hence *increased in density*. Convection currents would thus be started and the air would drop as indicated by the arrows. The heat would flow through the wall by conduction as was the case in the two previous sketches. If air motion is possible, the air on the cold side of the wall, when heated by conduction of heat from the wall surface, would become lighter in weight. Convection currents on the cold side of the wall would occur as indicated by the arrows in sketch *c*.

Actually heat transfer through a homogeneous wall is, of course, a *combi-*

nation of heat flow *at the same time* by *all three of the methods* illustrated in Fig. 1. In all three cases illustrated, the heat flowed through the solid material of the wall by means of conduction. When a high temperature source of heat is located near the surface of the wall, some radiant heat may be thrown right through the wall, remaining *radiant* heat. However, most radiant heat will be transformed into ordinary sensible heat when it impinges on the surface of the wall.

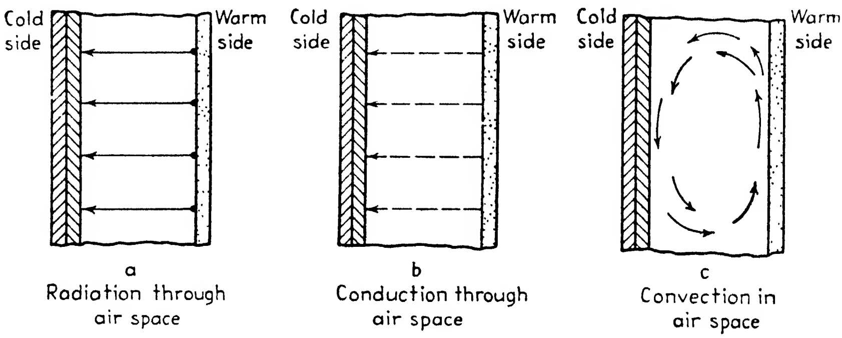


FIG. 2.—Heat transfer through air spaces in a hollow wall.

As we have said, most building walls are not so simply constructed as the 6-in. thick solid concrete wall we have just discussed. A frame wall might have a plaster interior finish, a 3⅝-in. air space, a layer of wooden sheathing, and finally a layer of wooden "siding" or shingles. Between the sheathing and the outside wooden finish layer, a coat of heavy paper is ordinarily applied, in average frame construction. The heat flow through the solid plaster layer and the heat flow through the solid wooden layer takes place in exactly the same way as the heat flow through a solid wall, which we discussed in connection with Fig. 1. With the wall illustrated in Fig. 2, we want to examine how the heat is transferred across the air space, from one of the solid layers of the wall to the other solid layer.

Here again, the total heat transfer is a combination of heat transfer by radiation, conduction, and convection. Sketch *a*, Fig. 2, illustrates the transfer of *radiant* heat across the air space. This heat transfer would occur even if this hollow space were a vacuum instead of an air space. Sketch *b* illustrates heat transfer across the air space by means of *conduction*. This heat transfer would occur if this hollow space were filled with solid material or if the air in the actual air space *remained motionless*. Sketch *c* illustrates heat transfer by convection from the warm side to the cold side of the air space. In actual walls, this convection heat transfer is an important factor.

One might say that a 3-in. air space would be twice as good a heat barrier as a 1½-in. space, and perhaps that a 6-in. air space would be twice as good a barrier again. Actually, if the air spaces are made wider than about ¾ to 1 in., very little improvement as a heat barrier is noticeable as the air space is widened. The reason that the expected increase in resistance against heat flow does not occur as the air space is widened indefinitely is that convection currents, as illustrated in sketch *c*, Fig. 2, become more and more pronounced. In fact it is definitely desirable to separate air space 1½ in. wide and wider into two air spaces, even though the separating material may be only thin paper or cardboard, or a thin film of metal. The separator itself is not an appreciable barrier to heat flow, but it breaks up the convection currents and interposes *two more surfaces* at which heat must be transferred from air to surface and from surface back to air.

Calculations of Transmission-heat Flow. k = thermal conductivity: the amount of heat expressed in Btu transmitted in 1 *hr* through 1 *sq ft* of a homogeneous material 1 *in. thick* for a difference in temperature of 1°F between the two surfaces of the material. (The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.)

f = film or surface conductance: the amount of heat expressed in Btu transmitted by radiation, conduction, and convection *from a surface to the air* surrounding it, or vice versa, in 1 *hr per square foot* of the surface for a difference in temperature of 1°F between the surface and the surrounding air. To differentiate between inside and outside wall (or floor, roof, or ceiling) surfaces, f_i is used to designate the *inside* film or surface conductance and f_0 the *outside* film or surface conductance.

U = thermal transmittance, or *over-all coefficient of heat transmission:* the amount of heat expressed in Btu transmitted in 1 *hr* per sq ft of the wall, floor, roof, or ceiling for a difference in temperature of 1°F between the air on the inside and that on the outside of the wall, floor, roof, or ceiling (often called the U factor).

For simple homogeneous walls, having no air space, it is easy to calculate U, knowing k and the proper values for f_i and f_0. The values of f depend upon the *speed* of the air brushing against the surface and upon the *nature of the surface* itself. In general, the rougher the surface, the higher the value of the film or surface conductances f. For instance, at a wind speed of 15 mph, f for glass and new white paint on wood is 5.1; f for rough concrete, rough brick, and rough plaster is 7; and f for stucco is 9. At an air speed of zero, f for almost all materials is between 1.5 and 2. For practical calculations, f_i is usually taken as 1.65, the air velocity being assumed to be zero. In practice, assuming a 15 mph wind and a surface finish smoother than that of rough brick, f_0 is taken as 6.00.

In order to find U, knowing k, the thickness of the wall being X, the equation for a solid homogeneous wall with no air spaces is

$$\frac{1}{U} = \frac{1}{f_i} + \frac{X}{k} + \frac{1}{f_0} \tag{1}$$

Suppose that we have a 2 in. thick solid wooden wall for which $k = 1.15$. By using Eq. (1) we may find U for this wall as follows:

$$\frac{1}{U} = \frac{1}{1.65} + \frac{2}{1.15} + \frac{1}{6.00}$$

$$\frac{1}{U} = 0.605 + 1.64 + 0.167 = 1.812$$

$$U = 0.55\ Btu/(hr)(sq\ ft)(°F) \quad \text{(note units)}$$

Suppose that you wished to calculate U for the 6 in. thick concrete wall illustrated in Fig. 1. k for dense concrete $= 12.0$; the wall is 6 in. thick; hence we have

$$\frac{1}{U} = \frac{1}{1.65} + \frac{6}{12.0} + \frac{1}{6.00}$$

$$\frac{1}{U} = 1.27$$

$$U = 0.788, \text{ or say, } 0.79\ \text{Btu/(hr)(sq ft)(°F)}$$

Refer to the definitions of U, k, and f_0. Note that U may briefly be defined thus:

$$U = Btu/(hr)(sq\ ft)(deg\ diff)$$

In other words, if we know U for a wall and if we know the temperature on each side of the wall we have only to measure the square feet of wall in which we are interested and we can determine the heat flow through the section of wall. This may be expressed as an equation as follows:

$$H = AU(t_2 - t_1) \tag{2}$$

where H = Btu per hour transmitted through the material of the wall, glass, roof, or floor
A = area in square feet of wall, glass, roof, floor, or material, taken from building plans or actually measured. (Use the net *inside* or heated surface dimensions in all cases.)
t_2 = temperature on warm side of wall
t_1 = temperature on cold side of wall

The expression $t_2 - t_1$ in Eq. (2) is the temperature difference between the warm and cold air and, hence, may be written in the equation directly as a *temperature difference*. For instance, if it is 90° on one side of a wall and 78° on the other side of the wall, we usually write the temperature difference of 12° directly in Eq. (2), instead of going through the step of writing $(t_2 - t_1)$. [Note that Eq. (2) will apply to the calculation of heat flow through a wall *either in* the case of a *cooling* installation *or* a *heating* installation. Note also that the inside temperature may not necessarily be a dry-bulb temperature as measured *at the breathing line* in the room. The latter distinction will not be used until Chap. XV, at which time it will be explained.]

Thus far we have seen that it is not difficult to calculate the value of U for a single *solid homogeneous* wall. Suppose we consider a compound wall, built up of three layers of three different materials, each of the three being itself a homogeneous material. For instance, we might have a wall consisting of a 3-in. layer of concrete, a 3-in. layer of wood, and a 2-in. layer of insulating board. Let us assume that the three layers of the wall are close together with *no air space* in between layers. Let us now call the thickness of the three materials X_1, X_2, and X_3 and their conductivities k_1, k_2, and k_3. We would then find U by means of Eq. (3).

$$\frac{1}{U} = \frac{1}{f_i} + \frac{X_1}{k_1} + \frac{X_2}{k_2} + \frac{X_3}{k_3} + \frac{1}{f_0} \tag{3}$$

Equation (3) begins to illustrate the difficulty of calculating U mathematically for building walls that are built up of different materials. If we change the wall [to which Eq. (3) applies] to a wall having *two air spaces,* we add at least *two more terms* to Eq. (3). This procedure is complicated and difficult, since U factors must be determined for every wall, floor, ceiling, door, window, and roof of a building regarding which a heating or cooling load is to be calculated. At this point, students begin to wonder about the advisability of tabulating U factors for most of the walls that are ordinarily encountered in actual building construction in order to save the lengthy task of calculating a U factor every time one is needed.

The A.S.H.V.E. has available tables giving just this information. These tables are based upon a considerable amount of test data as a result of work in their Research Laboratory. This does not mean that every single wall section shown in the tables has been tested to determine its U factor. Some of the U factors have been *calculated* by the method we have examined briefly, with additional complicating factors due to air space and surface of the wall. Other sections have been tested so that there is no question that the A.S.H.V.E. heat-transmission coefficients are more than sufficiently accurate for practical work. Furthermore these data are by far the best and most complete on heat-transmission information available. Note that the heat-transmission coefficients are based upon *still air* on the *inside* of walls and a wind *velocity of* 15 *mph* on *outside* walls. Since the outdoor wind velocity averages 12 to 17 mph for most locations in the United States, these tables are accurate without correction except for special locations known to be exposed to unusually high wind.

A rough index to the tables of heat-transmission coefficients is given below. Table 1 gives information as to the *type* and *thickness* of the various construction materials upon which the tables in general are based.

Table XIX[1]	Coefficients of Transmission (U) of Masonry Walls
Table XX	Coefficients of Transmission (U) of Masonry Walls with Various Types of Venners
Table XXI	Coefficients of Transmission (U) of Various Types of Frame Construction
Table XXII	Coefficients of Transmission (U) of Frame Interior Walls and Partitions
Table XXIII	Coefficients of Transmission (U) of Masonry Partitions
Table XXIV	Coefficients of Transmission (U) of Frame-construction Floors and Ceilings
Table XV	Coefficients of Transmission (U) of Concrete-construction Floors and Ceilings
Table XVI	Coefficients of Transmission (U) of Concrete Floors on Ground with Various Types of Finish Floorings
Table XVII	Coefficients of Transmission (U) of Various Types of Flat Roofs Covered with Built-up Roofing
Table XVIII	Coefficients of Transmission (U) of Pitched Roofs
Table XXIX	Coefficients of Transmission (U) of Doors, Windows, Skylights, and Glass Walls

Tables XIX, XX, and XXI are used in the same manner. In each case the table extends over two pages. At the extreme left is a *sketch* illustrating the typical construction of each type of wall. The next column, entitled "Type of wall," describes the construction *in words*. In many cases considerable detail is given in this description. Information is then given either as to the *thickness* of the wall, if the wall is of the same material throughout, or as to other slight modifications upon the same general type of wall. The column farthest to the right on each *left-hand page* of Tables XIX, XX, and XXI, gives a "Wall number." These numbers are given to provide a quick method of referring to any particular wall without writing a lengthy description of it.

On the right-hand pages of Tables XIX, XX, and XXI, about a dozen modifications regarding *inside wall finish* are given. Each of these interior-

[1] **Tables with roman numerals are given in the Appendix.**

finish modifications is lettered *A*, *B*, *C*, etc. Suppose that you wish to find the *U* factor for a 12-in. solid brick wall having an interior finish of ¾ in. of plaster on metal lath that is fastened to furring strips on the inside of the brick. Turn to Table XIX. Note that the first wall listed is *solid brick*, with information for thickness of 8, 12, and 16 in. Wall 2 is a 12-in. wall. Opposite wall 2 on the right-hand page follow to the right to the column under interior finish *D*. This is the inside finish of our wall, and the *U* factor is given as 0.25 Btu/(hr)(sq ft)(°F).

TABLE 1.—THICKNESSES AND TYPES OF CONSTRUCTION MATERIALS UPON WHICH TABLES XIX THROUGH XXIX ARE BASED

Material	Inches
Brick veneer	4
Plaster and metal lath	¾
Plaster (on wood lath, plasterboard, rigid insulation, board form, or corkboard)	½
Slate (roofing)	½
Stucco on wire mesh reinforcing	1
Tar and gravel or slag-surfaced built-up roofing	⅜
1-in. lumber (S-2-S)	25/32
1½-in. lumber (S-2-S)	1 5/16
2-in. lumber (S-2-S)	1⅝
2½-in. lumber (S-2-S)	2⅛
3-in. lumber (S-2-S)	2⅝
4-in. lumber (S-2-S)	3⅝
Finish flooring (maple or oak)	13/16

Solid brick walls are based on 4-in. hard brick (high density) and the remainder common brick (low density). Stucco is assumed to be 1-in. thick on masonry walls. Where metal lath and plaster are specified, the metal lath is neglected.

The remaining tables of heat-transmission coefficients are used in a similar manner. When looking up a *U* factor in these tables, always make a note of the wall number you use to pick your factor. For instance, the transmission coefficient we looked up in the preceding paragraph should be presented as follows in the solution of a problem:

$$\text{Brick wall, A.S.H.V.E. 2-D} \quad U = 0.25$$

We cannot urge you too strongly to make clear exactly what wall and interior finish you have used in selecting *U* factors. You should do this in a manner similar to that illustrated above in your problems and examinations during this course. You should continue to do it in any practical work you engage in, the reason in both cases being to enable others readily to check up on your work.

The remaining tables of coefficients of transmission are used in a manner similar to the use of Tables XIX, XX, and XXI. In each case the fundamental floor or ceiling construction is followed by a number of modifications, as to type of finish, each labeled A, B, C, etc.

Heat Transmission through Non-heated Attics, Sloping Roofs. You all know that the attic in a residence becomes a good deal warmer than the outdoor temperature in the summertime because of the sun beating on the roof. This fact would mean a tremendous heat leakage through the ceilings of the second floor rooms if we were endeavoring to cool these rooms. The practice in summer is to provide a fan to ventilate the attic thoroughly so that the air therein will be at the same temperature as outdoors. Under these conditions, heat flow down into the second-story rooms is calculated in the conventional way, based upon the temperature difference between the outdoors and indoors.

In the winter many attics are not heated at all, and hence there is a problem of heat leakage through the ceilings of the second-story rooms and into the attic whence the heat leaks out of doors through the roof. There is a simple method of obtainining an *over-all* heat-transmission coefficient for the second-floor *ceiling*, the *attic space*, and the *roof*. In using this method, the area of any vertical side walls in the attic must be added to the area of the roof. Of course, in some cases the sloping roof comes right down to the level of the attic floor and there are no vertical side walls. If the U factor for these vertical side walls is different from that of the roof or if there are any windows in the attic, these must be taken into consideration in determining the U factor for the roof (called U_r in the method below).

The over-all coefficient of transmission is determined by Eq. (4), keeping in mind the precaution in the preceding paragraph:

$$U = \frac{U_r(U_{ce})}{U_r + \frac{U_{ce}}{n}} \tag{4}$$

where U = the over-all coefficient to be used, multiplied by the *ceiling area* in our heat-transmission equation (2)

U_r = coefficient of transmission of the roof

U_{ce} = coefficient of transmission of the ceiling

n = area of roof divided by area of ceiling

Air Leakage or Infiltration. There are two primary causes of infiltration of outdoor air in a building. The major cause, which is present in all buildings except those completely protected from wind, is the pressure caused by wind against the sides of the building. The other cause exists only in the case of very large buildings. This second cause is the familiar convection circulation tendency which we have discussed several times. In the case of tall buildings it is generally called "stack effect" because the tall buildings filled with warm air in wintertime act a great deal like a smokestack. A draft is created tending to draw air in any of the cracks or openings in the lower portion of the building.

Both of the above causes of infiltration are *able to produce actual air leakage* because of cracks or porousness in the construction of the building. Plain brick walls are surprisingly porous even though they may be as much as 1 ft thick. However, a first-class plastering job on the interior of masonry walls, coupled with proper sealing of the plaster at the baseboard (floor level), reduces infiltration so low that it *may be neglected* with wind velocities up to 20 mph. Frame walls without any interior finish can have an air leakage of as much as 60 to 80 cu ft of air per hour per square foot of wall, at ordinary wind velocities. Table 2 gives the infiltration through a well-constructed frame wall with a lath and plaster interior finish.

TABLE 2.—INFILTRATION THROUGH FRAME WALLS

	Wind velocity, mph					
	5	10	15	20	25	30
Leakage, cu ft of air per hr per sq ft of wall*—I	0.03	0.07	0.13	0.18	0.23	0.26

* Wall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath, and three coats of gypsum plaster.

When air conditioning becomes as universal as central heating is today, it is likely that frame buildings and stone and masonry buildings will all be constructed with windows that are completely sealed. Such windows will be no disadvantage because they cannot be opened, and they will provide almost perfect sealing against infiltration. A few buildings today have been constructed in this manner, in some cases glass and glass brick have been used in the walls with no *conventional windows* at all. However, most existing buildings and most new buildings being constructed today utilize a method of installing and a method of constructing windows such that air leakage is unavoidable.

We have just discussed the *causes* of infiltration. What *effect* does infiltration have? During the winter season, outdoor air that leaks inside is cold and ordinarily very low in dew point. If this air is to be brought well within our comfort chart, we must *add sensible heat* and *humidity* to it. Even in a non-air-conditioning building heated by standing radiation, it is imperative that the heating plant have sufficient capacity to furnish the sensible heat required to warm up the air that leaks in. During the summer season, the "stack effect," tending to cause air leakage in tall buildings, is largely eliminated. The major causes of air leakage in a building, outside wind velocity, remains, and infiltration presents a definite problem. The outdoor air that leaks into an air-conditioned space in summer is *higher in dry-bulb* temperature and *higher in dew point* than the desired conditions inside. Therefore, this air has the same effect as though additional people were in the conditioned space, adding sensible and latent heat to the air. The cool dehumidified air supplied to the conditioned space must be able to stand not only the addition of sensible and latent heat from transmission and normal internal sources, but also the *effect of mixing* with the amount of warm humid air that enters the conditioned space because of infiltration.

It is obvious from the foregoing brief discussion that infiltration is undesirable from two standpoints. (1) It places a tax upon the cooling or heating systems, requiring additional capacity in the equipment. (2) There is an additional *operating* cost involved in conditioning all the air that leaks into the conditioned space.

Calculation of Infiltration. In this chapter it is our purpose to learn to calculate infiltration *in cubic feet of air per hour*. The effect of this air on the heating (or cooling) load will be covered in future chapters. The material is presented in this way because the calculation of the *amount of* infiltration air is the *same* for summer and winter seasons. The calculation of the *effect* of this air is somewhat different for a winter system than for a summer conditioning system. We shall calculate infiltration through *frame* walls only, as we have pointed out that infiltration through a well-plastered masonry wall is negligible. We shall base all our infiltration calculations on the *cause* of wind velocity alone. Students interested in calculating infiltration in tall buildings are referred to the A.S.H.V.E. "Guide."[1]

Table 2 gives information regarding infiltration through frame walls. In using Table 2, measure the number of square feet of exposed wall and call this A, read the proper value I from Table 2. The number of cubic feet per hour of air leakage (which we shall call Q) is then found as follows:

$$Q = AI \tag{5}$$

Infiltration around window *frames*, where the frames connect to the building, and around the window *sashes* within the frames, is found from Table 3. Table 3 with its footnotes is self-explanatory. For the definitions of "crack" and "clearance" which are used in Table 3, refer to Fig. 3. In using Table 3 you will determine the infiltration in cubic feet of air per hour *per foot of crack*. To obtain the total length of crack for a double-hung

[1] "Heating, Ventilating and Air Conditioning Guide," Chap. 6, p. 127, formulas (1) and (2), 1939.

wooden sash window, take *twice the height* plus *three times the width* of the window. These measurements allow for the outside perimeter of the window plus the length of crack across the center of the window at the lock. For metal pivoted windows of the industrial type, take the total distance around the movable or swinging sections.

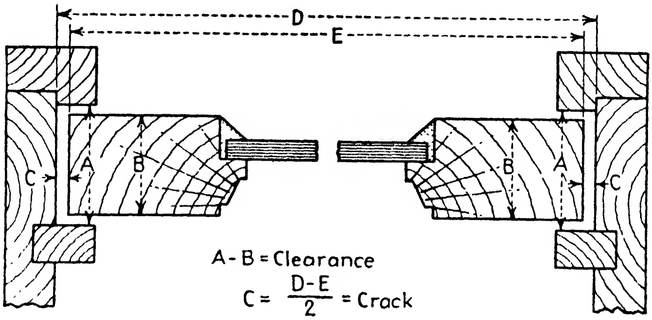


FIG. 3.—Diagram illustrating crack and clearance. (*Courtesy of A.S.H.V.E.*)

In calculating the infiltration air quantity by means of Table 3, proceed as follows:

Obtain the leakage factor around the outside of the *frame* from the first three lines in Table 3. Take the total perimeter of the window as the length of crack, and multiply these two quantities by one another; the result is the air leakage in cubic feet per hour *around the frame* of the window.

Choose the proper leakage factor around the *sash* of your window from the remainder of Table 3, determine the proper length of crack as described above. Multiply the length of crack by the leakage factor from Table 3. The result is the air leakage in cubic feet per hour *around the sash.*

Add the two leakage quantities just calculated. The result is the *total air leakage* pertaining to the one window upon which the computation is based. If there is more than one window of this type, multiply by the number of windows.

Where there are two heavy weather-stripped tightly fitting doors, forming a vestibule or small reception hall, there is extremely small infiltration. In general, infiltration around doors may be figured as follows:

For a well-fitted average size door (approximately 3 $\times$ 7 ft), not weather-stripped, the leakage values for a poorly fitted double-hung wood sash window may be used. For a similar door not weather-stripped, but fitting loosely, use twice these values. If the door is weather-stripped, use 50 per cent of either one of the preceding values, depending upon whether the door fits tightly or loosely.

TABLE 3.—INFILTRATION THROUGH WINDOWS*
Expressed in Cubic Feet per Foot of Crack per Hour

Type of window	Remarks	Wind velocity, mph					
		5	10	15	20	25	30
Double-hung wood sash windows (unlocked)	Around frame in masonry wall, not calked[b]	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall, calked[b]	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction[b]	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weather-stripped, 1/16-in. crack and 3/64-in. clearance.[c] Includes wood-frame leakage[d]	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weather-stripped[d]	4.3	15.5	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weather-stripped, 3/32-in. crack and 3/32-in. clearance.[e] Includes wood-frame leakage[d]	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weather-stripped[d]	5.9	18.9	34.1	51.4	70.5	91.5
Double-hung metal windows[f]	Non-weather-stripped, locked	20	45	70	96	125	154
	Non-weather-stripped, unlocked	20	47	74	104	137	170
	Weather-stripped, unlocked	6	19	32	46	60	76
Rolled section steel sash windows[k]	Industrial pivoted, 1/16-in. crack[g]	52	108	176	244	304	372
	Architectural projected, 1/32-in. crack[h]	15	36	62	86	112	139
	Architectural projected, 3/64-in. crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in. crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in. crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in. crack[j]	3	10	18	26	36	48
	Heavy casement section, projected 1/32-in. crack[j]	8	24	38	54	72	92
Hollow metal, vertically pivoted window[f]		30	88	145	186	221	242

* A.S.H.V.E. "Guide," Chap. 6, p. 124, 1939.

[a] The values given in this table, with the exception of those for double-hung and hollow metal windows, are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in the papers listed at the end of this chapter.

[b] The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and notcalked tests.

[c] The fit of the average double-hung wood window was determined as 1/16-in. crack and 3/64-in. clearance by measurements on approximately 600 windows under heating-season conditions.

[d] The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called "elsewhere" leakage. The frame leakage values included are for wood-frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking.

[e] A 3/32-in. crack and clearance represents a poorly fitted window, much poorer than average.

[f] Windows tested in place in building.

[g] Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

[h] Architectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and it has refinements in weathering and hardware. Used in semimonumental buildings such as schools. Ventilators swing in or out and are balanced on side arms. 1/32-in. crack is obtainable in the best practice of manufacture and installation, 3/64-in. crack considered to represent average practice.

[i] Of same design and section shapes as so-called "heavy-section casement" but of lighter weight. 1/64-in. crack is obtainable in the best practice of manufacture and installation; 1/32-in. crack is considered to represent average practice.

[j] Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening. 1/64-in. crack is obtainable in the best practice of manufacture and installation; 1/32-in. crack is considered to represent average practice.

[k] With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With 3/64-in. crack, representing poor installation, leakage at contact with steel framework is about one-third and at mullions about one-sixth of that given for industrial pivoted windows in the table.

Infiltration through single doors that are opened frequently is calculated differently for summer and winter. (Note that this is one of two times in this chapter where the calculation of an item is different for the two seasons.) For a single door in a shop or store, use three times the infiltration value for the door calculated as described above for the *winter season*. For the summer season, use five times the value by the method above. (This difference is recommended because experience has shown that people passing through doors of this type tend to close them quickly in most severe winter weather, whereas they are not so careful in severe summer weather.)

Most larger public buildings and department stores use either revolving doors or have a second set of swinging doors 10 to 20 ft inside the outer doors. Some smaller buildings and stores adhere to this practice also. The infiltration through such arrangements obviously depend upon the rate at which people are entering and leaving the building as well as the tightness of the seal of doors. In the case of the double set of swinging doors, the infiltration rate also depends upon the distance between the two sets of doors.[1] Because of the complexity of the problem of accurately determining the infiltration rate through swinging and revolving doors, no accurate information is available at this writing. According to the A.S.H.V.E., the values in Table 4 represent approximate values which may be used and which represent current practice in this respect.

Effect of Exposure. Let us assume the extreme case of a square building with windows along *only one wall.* If this building were erected in a location where the prevailing wind comes from the west, we could imagine two extreme conditions. The building can be located with the wall having *windows facing west,* in which case infiltration would be a *maximum.* On the other hand, the building could be located with the wall having windows facing east, in which event if the masonry walls were well sealed there would be very little infiltration. Remember it is the pressure caused by wind velocity acting against cracks or porous walls which causes infiltration.

The preceding paragraph indicates merely two extreme conditions. You can imagine many different circumstances where the same build-

[1] In many cases, department stores and public buildings, using two sets of swinging doors provide auxiliary special heating equipment in the space between the two sets of doors. Such equipment is frequently sized to have a capacity to heat the space between the two sets of doors 40° above the outdoor temperature. This means that with the outdoor temperature at 0°F the space between the doors can be heated to 40°F. With ordinary outdoor weather of 30°F, the space can be heated to 70°F, under which conditions the net infiltration into the *inside* of the building imposes no additional load upon the building heating system. A few theaters and department stores have tried to reverse this system in the summer-cooling season. Because of the high cost of cooling, this more or less wasteful practice has been little used during the summer season, although it is widely used in winter.

TABLE 4.—INFILTRATION THROUGH OUTSIDE DOORS FOR COOLING LOADS*
(Cubic feet per minute per person in room)

Application	Pair 36-in. swinging doors, single entrance†
Bank	7.5
Barber shop	4.5
Broker's office	7.0
Candy and soda	6.0
Cigar store	25.0
Department store	8.0
Dress shop	2.5
Drugstore	7.0
Furrier	2.5
Hospital room	3.5
Lunchroom	5.0
Men's shop	3.5
Office	3.0
Office building	2.0
Public building	2.5
Restaurant	2.5
Shoe store	3.5

* For doors located in only one wall or where doors in other walls are of revolving type.
† Vestibules with double pair swinging doors, infiltration may be assumed 75 per cent of swinging door values.
Infiltration for 72-in. revolving doors may be assumed 60 per cent of swinging door values.
SOURCE: A.S.H.V.E., "Heating, Ventilating and Air Conditioning Guide," Chap. 6, 1939.

ing might be subject to different amounts of air leakage depending upon the direction faced by the most windows. In calculating the total infiltration, we do not use the infiltration around *all windows* in the building, calculated as we have learned to calculate the infiltration around *one window*. Instead, we include certain windows in our calculations and exclude other windows entirely, not endeavoring to adjust the infiltration value based on all windows in the building. General rules as to what windows to include and what windows to exclude are given in the following paragraph.

For a building or space having no interior partitions, use one-half the total length of crack for all windows on the floor being computed. For spaces broken up by partitions, consider the individual rooms rather than the floor as a whole. In computing the individual rooms, follow this procedure.

Room with one exposed wall having windows	Entire crack length around all windows
Room with two exposed walls having windows	Total crack length around all windows in only the wall having the greatest length of crack

Rooms with three or four exposed walls having windows	Total length of crack around all windows in the wall having the greatest length of crack. *But do not use less than one-half the total length of crack for the entire room*

Approximate Method of Estimating Infiltration. In the past few pages, we have outlined the complete method of calculating infiltration of air by the most exact methods that are accepted practice today. In many cases, especially when proposed installations are merely being estimated, it is not worth while putting this amount of effort into calculations regarding infiltration. This is especially true in the case of a building of excellent construction where it is known that infiltration will be held to a minimum by the smallness of cracks and the tightness of the seal around window frames. Therefore, in many cases, a quick rough method of estimating infiltration is desirable. Many thoroughly experienced men in air-conditioning work use this approximate method, revising the estimated values upward or downward in the light of their own experience. Until one accumulates a satisfactory background of experience, this approximate method should be used only for rough work and as a rough check on the values determined by the more accurate and lengthy method.

Table 5 shows approximate values for infiltration. In using this table, consider the side of a room "exposed" only if that side of the room contains at least one window. Again we wish to caution you against using Table 5 for anything except approximate values until your own experience and judgment become developed to a high degree. Consider the example of a living room 20 by 15 ft by 10 ft high. This room has a volume of 3,000 cu ft. Using Table 5 for living rooms, the infiltration would be one to two changes of air per hour, or 3,000 to 6,000 cu ft per hr. If this room is in a new house with tightly weather-stripped windows, we might feel safe in using the lower value, *viz.*, 3,000 cu ft per hr. Suppose, however, that the entrance to the house opens *directly into this living room.* We then have a door to the outside and might use the *entrance hall value* in Table 5. This value would indicate an infiltration of 6,000 to 9,000 cu ft per hr. Based only upon the information in this paragraph, it would be impossible to say which of these values is most nearly correct. The correct value can be obtained only by the estimate of an experienced person or by actually computing the infiltration around the various doors[1] and windows.

Effect of Mechanical Ventilation. The methods of calculating infiltration that we have discussed so far apply to buildings heated with standard

[1] Unweather-stripped French doors containing many small panes of glass, opening out of living rooms onto open porches, admit a great deal of air leakage. Such doors should be figured as poorly fitting doors, and the length of crack where the doors join should be included.

radiation and also to buildings heated with winter air conditioning where 100 per cent recirculation is used. Many of the inexpensive homes equipped with winter air conditioning operate with 100 per cent recirculation in an attempt to reduce the fuel cost. In the preceding chapter, with a building equipped with a duct system, we saw the desirability of introducing a measured amount of fresh outdoor air for ventilation by means of the

TABLE 5.—AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR PROVIDED FOR VENTILATION

Kind of room or building	Number of air changes taking place per hour
Rooms, 1 side exposed	1
Rooms, 2 sides exposed	1½
Rooms, 3 sides exposed	2
Rooms, 4 sides exposed	2
Rooms with no windows or outside doors	½–¾
Entrance halls	2–3
Reception halls	2
Living rooms	1–2
Dining rooms	1–2
Bath rooms	2
Drug stores	2–3
Clothing stores	1
Churches, factories, lofts, etc.	½–3

SOURCE: From A.S.H.V.E., "Heating, Ventilating, Air Conditioning Guide," Chap. 6, 1939.

duct system. If in a certain office we take 8,000 cfm of recirculated air plus 2,000 cfm of outdoor air and deliver this air back into the office, *we are supplying approximately* 2,000 *cfm more than we are removing.* This excess air would then build up a slight pressure within the office until the pressure difference between indoors and outdoors is sufficient to make this 2,000 cfm leak out of the office. If the office were not connected to the floors above or below by means of open stairways, this air would have to leak out of the office through the *same passages* normally used by infiltration air leaking in. You can readily understand that this would greatly reduce the infiltration. This conclusion is correct. You might have thought that infiltration would be entirely eliminated, but this is not true.

We must, therefore, have a procedure for modifying the infiltration we have calculated in the case of a building introducing fresh air by mechanical means. This method must of necessity be just as general and approximate as our infiltration information in Table 5, because so many factors are involved. The following procedure is satisfactory:

1. Obtain the infiltration value which you are satisfied is fairly correct by either the approximate or the accurate calculation method.

2. Compare the infiltration quantity of air expressed in cubic feet *per minute* to the quantity of fresh air supplied by the ventilating system, the latter also being expressed in cubic feet per minute.

3. Compute the fresh air supplied by the ventilating system in *per cent of total air* handled by the ventilating system.

4. If the fresh air is *more than* 20 *per cent* of the total air, use only the infiltration through *doorways* of the conditioned space.

5. If the fresh air is *less than* 15 *per cent* of the total air handled by the ventilating system, choose between the following:

a. If the infiltration is greater than the fresh air supplied, use the *total amount* of infiltration as calculated.

b. If the infiltration is less than the fresh air supplied mechanically, use *one-half* of the infiltration as calculated.

PROBLEMS

1. Describe how heat flows through a solid concrete wall 6 in. thick. Describe how heat flows through a wall composed of two 3 in. concrete slabs, with a 2 in. thick insulating material between the concrete. Suppose air space is substituted for the insulation; what is the effect?

2. Discuss briefly heat flow through an 8 in. thick brick wall, finished on one side, containing *no air space*. What is the meaning of the heat-transfer coefficient U?

3. If $k = 12$ for concrete, $k = 0.3$ for the insulation, $f_0 = 6.00$, and $f_i = 1.65$, calculate the U factor for the insulated wall in Prob. 1 and for the 6-in. solid concrete wall.

4. Look in Tables XIX to XXIX, and select what you consider typical residential construction sections for the list below, referring to the sections selected *by number*. Show the U factor for each section: Ordinary second-story floor with a plaster finish ceiling below. An inside partition plastered on both sides. An outside wall assuming shingles outside and metal lath and plaster inside. A single ⅛-in. thick window glass. A plywood wooden door (very thin wood panel).

5. Roof section 1-A in Table XXVIII is used for a pitched roof, the total area of which is 1,000 sq ft. The attic floor and its ceiling below is 3-B from Table XIV, and the area of the attic floor is 700 sq ft. Obtain the over-all U factor that should be used for determining heat transmission through the uninsulated attic based upon the area of the attic floor (which is the same as the area of the second-floor ceiling). Calculate the heat loss by transmission in Btu per hour, if the temperature indoors at the second floor ceiling is to be 80°F with the outdoor temperature as 0°F.

6. Find the U factor for a masonry wall having a 4-in. cut stone veneered facing backed up by an 8-in. concrete wall, the interior finish being plaster on wood lath—furred. Find the U factor for a masonry floor, the concrete of which is 7 in. thick, the ceiling being furred metal lath and plaster, the floor being ¼-in. thick heavy linoleum on top of yellow pine. (Both of these U factors involve a "judgment interpolation" in the tables. It is not possible to perform *ordinary* interpolation in the heat-transmission coefficient tables, but it is still possible to estimate values not shown exactly in the tables.)

7. What are the two major causes of infiltration? In what type of structure is infiltration due to the difference between the indoor and outdoor temperatures of real importance? Cite also a building where the latter type of infiltration is of no importance at all.

8. Consider a living room 30 by 20 ft by 9 ft high, the building being of ordinary frame construction. One 30-ft wall has *four* 3- by 6-ft *windows* weather-stripped, having about 1/16-in. crack and 3/64-in. clearance, and faces west. One 20-ft wall contains *two similar windows* and faces north. The other two walls are inside walls containing no windows. The prevailing wind is from the northwest, and the average velocity is 20 mph. Calculate the infiltration into this room in cubic feet per hour based upon the information in Table 2. (A sketch will help you visualize this problem.)

9. Estimate the infiltration into the living room described in Prob. 8 by the approximate method, using the information in Table 5. If a mechanical ventilating system is used in connection with the living room described in Prob. 8, supplying 200 cfm of fresh outdoor air to the room, what infiltration value would you use? Use your answer to Prob. 8 and to the first part of this problem to help your decision.

10. Take an average of the two air-leakage quantities you computed in Probs. 8 and 9, *dis*regarding the mechanical ventilating system. For outdoor air at 0°F, 100 per cent humidity, how many pounds of air per hour leak into the room? If the indoor condition is to be 66°E.T., 30 per cent relative humidity, find the following: (*a*) interior dry-bulb temperature and dew point, (*b*) sensible heat content and absolute humidity in grains per pound for the outdoor air, (*c*) how many Btu per hour of sensible heat must be added to the air that comes in to bring it up to the indoor conditions, (*d*) how many pints of water per hour must be added to bring the outdoor air, which leaks in, up to the desired indoor dew point (1 pint = 1 lb almost exactly, for water).

CHAPTER XV

CALCULATION OF HEATING LOAD

General. In the design of any winter air-conditioning system, or even a simple winter-heating system, it is necessary to calculate as accurately as possible the probable maximum heating load that will be imposed on the system. This must be determined for the building as a whole and also for the independent sections or rooms. Only in this manner is it possible to select the size of the equipment to be used in the central heating plant, and also the size of the equipment that is to supply heat to the individual rooms.

In the preceding two chapters we have learned the desirable indoor conditions that should be maintained both in summer and in winter. We have also learned the method for calculating transmission heat flow into or out of a building and the method of calculating air leakage in cubic feet per hour during either season. In this chapter, we wish to develop a workable practical procedure for calculating the total winter heating load due to all causes. In the next two chapters, we shall study the means of supplying the required heat by systems using boilers and radiators and then by systems utilizing ducts to distribute warm air. We shall also investigate the possibilities of combining some heating by means of standing radiation with a duct system for a complete conditioning installation.

Selection of Indoor Dry-bulb Temperature. From Chap. XIII we have seen that the indoor effective temperature during the winter season should be approximately 66°E.T. The relative humidity that will be maintained indoors will depend upon what means are provided for humidifying and upon whether storm windows (double windows) or ordinary single windows are provided. Where the outdoor dry-bulb temperature does not go below 0°F, the following practical standard may be used:

1. No humidification—indoor relative humidity 10 per cent
2. Good means for humidification; single windows—indoor relative humidity 30 percent
3. Good means for humidification; double windows—indoor relative humidity 40 to 50 per cent

From the above and from the comfort chart, the indoor dry-bulb temperature at the breathing line may be selected such that the effective temperature will be approximately 66°E.T. This dry-bulb temperature will probably be between 68 and 74°F. The customer's wishes may, of course, always modify the indoor dry-bulb temperature selected.

Since warm air is lighter in weight than cool air, the warmest air in a room will be at the ceiling and the coldest air at the floor. Therefore, the average temperature of a heated room will be warmer than the dry-bulb temperature we are maintaining at the breathing level, and the best practice calls for taking into account this warm air near the ceiling in figuring the heat transmission through the walls. For rooms with a ceiling height up to 20 ft, the following formula may be used to get the average indoor temperature to be used in calculating heat flow. [See Eq. (2), page 248, here one average indoor temperature is t_2.]

For rooms heated by radiation, not having duct systems that provide air motion and stir up the air, use the formula

$$T_m = T_d\,(1 + 0.02\,h) + \text{mean (or average) temperature inside walls} \qquad (1)$$

where T_m = inside temperature at h ft above level where temperature is T_d
T_d = design temperature at breathing line (or line 30 in. above floor)
h = distance from breathing line (or 30-in. line) to the average (or mean) height for the room, *i.e.*, a point halfway between the floor and ceiling

The breathing line is usually taken 30 in. to 5 ft above the floor, depending upon whether the room is primarily used by occupants seated or standing, or arbitrarily set according to a customer's wishes.

Suppose you have a room 18 ft high, using a 5-ft breathing line. The mean height is ½(18) = 9 ft. Value h for formula (1) is then (9–4) = 5 ft. We then have

$$T_m = 70\,[1 + (0.02)(4)] = 70(1.08) = 75.5° \qquad (2)$$

Since this method of allowing for the warmer air toward the ceiling is approximate, we round off the above figure to 76°. For t_2 in Eq (2), page 248, we use T_m as calculated, or t_2 = 76°. In the case of rooms having duct work, such that the air in the room is stirred up to a considerable extent, it is permissible to change the decimal 0.02 to 0.01. (This correction is often neglected entirely for residential work, because rooms are about 9 ft high. Hence h would only be about 1 ft.) When a ceiling is under a roof, or has unheated space above it, the temperature of the air under the ceiling must be found by Eq. (1). Use T_c = temperature at ceiling instead of T_m; use h as distance from breathing line to ceiling.

Heat loss from a basement is determined by one of two methods, depending upon whether the basement is heated or not. Except for basement recreation rooms, cellars are usually heated by leaving the necessary surface of steam pipes uninsulated, or in some cases by providing a small radiator. If a basement is heated, its temperature may be taken as 60°F. The outside temperature for the floor, and the portion of the side walls more than 1 ft below the ground, may be taken as 50°F. The ordinary outdoor temperature may be used for the remainder of the basement side walls. If the basement is not heated, use the same outdoor temperatures, but take the indoor temperature as 35°F. (A basement must be heated to at least 35°F if there are any water pipes within the basement.)

Outdoor Winter Design Temperature. It is obvious that it would be wasteful of money and equipment to design heating installations of sufficient capacity to maintain the desired indoor temperature on a day so cold as the coldest weather *ever recorded* in any particular city. For instance, the official weather-bureau records show the coldest weather ever recorded in New York City as −14°F. Many residents of New York City boroughs, other than Manhattan, personally recorded (though perhaps not accurately) −18°F during the winter of 1934 to 1935. Ordinarily the temperature in New York City does not go below 0°F for more than a few hours during the winter, and many winters pass without the temperature dropping as low as 0°F. Therefore, an outdoor temperature 10 to 15° above the lowest temperature ever recorded is usually selected as the outdoor winter-design temperature for heating systems. For New York, the accepted outdoor winter-design temperature is 0°F, for Boston −5°F, for Baltimore, 5°F, and for Philadelphia 5°F. Table 1 gives certain winter-design information taken from data of the United States Weather Bureau.

Wind Velocity. In Chapter XIV we stated that wind velocity affects air leakage and also affects the value f_0 which is used in arriving at over-all coefficients of heat transmission U. From Table 1, you can see that very few cities show average winter wind velocities higher than 15 mph. The U factors given in Tables XIX to XXIX are based on outdoor wind velocities of 15 mph so that they may be used for any cities where the average wind velocity is *under* 18 *mph.* The correction for the U factor, due to an increase in the wind velocity from 15 to 18 mph, would be so small that it would not be worth while. (Even Chicago, often called the "Windy City," shows an average winter wind velocity of only 12.5 mph.)

In figuring air leakage, the most exact practice would be to use the wind velocity shown by data such as that in Table 1. However, this would usually involve interpolation, so a practical short cut is frequently used. It is suggested that where the average winter wind velocity is below 10 mph infiltration be calculated using the wind velocity *as* 10 *mph.* If the average winter wind velocity is more than 10 mph calculate infiltration *on the basis of* 15 *mph.* For average wind velocities between 15 and 20 mph, use table values for 20 mph, etc.

Calculation of Transmission Load. The building or the plans of the building must be examined. This examination is for two purposes. (1)

The construction of the various walls, floors, doors, windows, etc., must be noted so that proper U factors may be selected from Tables. Measurements must be taken either from the building itself or from the plans so that the number of square feet of wall area, glass area, floor area, etc.,

TABLE 1

State	City	Average temp, Oct. 1 to May 1, °F	Lowest temperature ever reported, °F	Average wind velocity, Dec., Jan., Feb., mph	Direction of prevailing wind, Dec., Jan., Feb.
Conn.	New Haven	38.4	-15	9.7	N
D.C. . . .	Washington	43 4	-15	7.1	NW
Fla.	Jacksonville	62.0	10	9.2	NE
Ga.	Atlanta	51.5	-8	12.1	NW
Ill.	Chicago	36.4	-23	12.5	W
Maine . . .	Eastport	31 5	-23	12.0	NW
Mass. . . .	Boston	38 1	-18	11.2	W
Md.	Baltimore	43.8	-7	7.8	NW
N.H. . . .	Concord	33.3	-35	6.6	NW
N.J.	Atlantic City	41.6	-9	15.9	NW
N.Y. . . .	Albany	35.2	-24	8.1	S
	New York	40.7	-14	17.1	NW
Ohio	Cleveland	37.2	-17	13.0	SW
	Columbus	39.9	-20	12.0	SW
Pa.	Philadelphia	42.7	-6	11.0	NW
	Pittsburgh	41.0	-20	11.7	W
R.I.	Providence	37.2	-17	12.8	NW
Va.	Lynchburg	46.8	-7	7.1	NW
	Norfolk	49.3	2	12.5	N
	Richmond	47.0	-3	7.9	SW
W.Va. . . .	Parkersburg	42.6	-27	7.5	SW

may be computed. From Eq. (2) in Chap. XIV, you will recall the need for computing these areas. [Area is represented by A in Eq. (2), Chap. XIV.] In measuring areas of walls, either from plans or from the actual building, the *inside dimensions* of each room should be used. This is because the heat-transfer formula and the U factors that are used are based upon A being the area on the inside of the wall sections.

After selecting the various U factors and computing the areas as described above, it is simply necessary to calculate the transmission heat loss, using Eq. (2), Chap. XIV. For your convenience, this equation is repeated.

$$H = AU\,(t_2 - t_1)$$

In using this equation, remember to calculate A in *square feet* by the method described above. Be careful also in the selection of U factors and be sure that t_2 and t_1 are selected in the manner which we have discussed.

Exposure Factor. For many years it has been customary to use arbitrary exposure factors increasing the heat transmission through the walls exposed to prevailing winds by these factors. There is still a need for actual test data regarding the correct increase which should be allowed for walls exposed to prevailing winds. Since our transmission loss is calculated with U factors based upon a 15 mph wind, and since infiltration is calculated on a wind velocity *at least as great as the prevailing wind*, calculations of heat load made on the basis outlined in this chapter should be correct unless the wind velocity exceeds 15 mph. On this basis and taking account of the fact that low buildings are usually sheltered from wind by adjoining buildings, one school of thought favors using no exposure factor at all.

TABLE 2.—PER CENT TO ADD TO TRANSMISSION LOSS FOR ROOMS WITH WALLS EXPOSED TO PREVAILING WINDS

Exposure to prevailing wind	Direction of prevailing wind					
	W	NW	N	NE	E	SE, S, and SW
One wall exposed	10	10	15	10	5	3–5
Two walls exposed	15	15	20	15	10	5–8

A second school of thought favors *adding a flat* 15 *per cent* to the transmission heat loss through any walls facing northwest, north, or northeast, *regardless of wind direction.* A third school of thought favors adding a flat 15 per cent to the transmission heat loss through any walls exposed to the prevailing wind, *regardless of* the *direction of exposure.* You can readily see that these three methods are sharply conflicting.

Those who favor using no exposure factor at all, provided the prevailing wind is no greater than 15 mph, are theoretically justified. On the other hand, all our discussion of wind velocity is based on *average* wind velocity. Since there are bound to be cold days when the wind is blowing 25 to 40 mph, there is some justification for the use of some type of exposure factor. The second school of thought assumes that the prevailing wind will probably be from the northwest, north, or northeast on a cold winter day so that maximum load is being imposed upon the heating system. The third school of thought assumes that the high-speed cold winds will always come from the *direction of the prevailing wind.* Actually, winds are capricious and do not follow any set law. Districts having prevailing southwest winds at 10 mph may have storms involving *cold north winds at* 30 *mph.* We feel that Table 2 represents a reasonable compromise regarding exposure factors.

Infiltration. Calculate the infiltration in cubic feet per hour using the method given in Chap. XIV. The air that leaks into the building *must at least be raised in temperature*; if the building has a winter air-conditioning system providing humidification, the air that leaks in must also be increased in humidity. These two loads should be calculated separately, since there are many cases where little or no *sensible* heat must be added to the infiltration air. The *total* heat and *latent* heat may then be calculated if the air is to be both heated and humidified. In the latter case the difference between the total heat and the sensible heat gives the latent heat, or humidification load, from which the weight of water to be added may be calculated. The weight of water to be added may also be calculated on an absolute-humidity basis instead of a Btu basis. Let us now set up standards for these various calculations.

To find the sensible heat that must be added to the infiltration air, use the following equation:

$$H_s = 0.24\left(\frac{Q}{v}\right)(t_2 - t_1) \tag{3}$$

where H_s = sensible heat to be added to raise the temperature of the air from the outdoor temperature to the indoor temperature, Btu per hr
0.24 = specific heat of the air, Btu per lb
Q = infiltration, cu ft of air *per hour*
v = specific volume of the outdoor air, cu ft per lb
t_2 = indoor temperature
t_1 = outdoor temperature

In the above equation the quantity $Q \div v$ equals the *weight of infiltration air in pounds per hour.* Hence the right-hand side of Eq. (3) is the same as the right-hand side of our original heat-transfer equation in Chap. V, which was

$$cW\ (T_2 - T_1)$$

If a winter-heating installation does not provide for humidification, the above calculation is the only one necessary in determining the added heat imposed by air leakage.

Where humidification is used and a definite indoor relative humidity is to be maintained, water must be evaporated into the air in an amount that will raise the dew point of the outdoor air to the desired indoor dew point. The heating system must of course supply the heat required to vaporize this water. Therefore, insofar as the heat output of the furnace is concerned, we can calculate the *total* heat that must be added to the outdoor air that leaks inside. This is done by means of the following equation:

$$H_t = \frac{Q}{v} (\mathrm{H}_{ti} + H_{to}) \tag{4}$$

where H_t = total heat to be added to the infiltration air, *Btu per hr*

$\frac{Q}{v}$ = same as in Eq. (3); *i.e.* pounds per hour of infiltration air

H_{ti} = total heat of the indoor air, Btu per lb

H_{to} = total heat of the outdoor air, Btu per lb

The latent heat that is added to the infiltration air can be determined by taking the difference between the heat quantities calculated in Eq. (3) and (4). If we let H_L equal the latent heat to be added, expressed in Btu per hour, we may from our knowledge of psychrometry from Chap. VII write the following equation:

$$H_L = H_t + H_s \tag{5}$$

It would be awkward to determine the amount of water to be evaporated into the air from the latent heat, which we could find by the method described in the preceding paragraph. If we were to determine the amount of water to be evaporated in this manner, we would use the following equation:

$$W = \frac{H_L}{h_{fg}} \tag{6}$$

where W = pounds of water per hour to be evaporated

H_L = latent heat, Btu per hr from Eq. (5)

h_{fg} = latent heat of vaporization of water at the condition at which the water is evaporated, Btu per hr

The difficulty with this means of calculating the weight of water to be evaporated is that we are not sure of the exact temperature at which the evaporation takes place. Depending upon the type of humidifying equipment used, water might be evaporated into the air at any temperature from 70°F (room temperature) up to as high as 212°F (actively boiling water). As a result our latent heat of vaporization of the water (h_{fg}) might be anywhere from 975 to 1050 Btu per lb. Since the correct value is not known, common practice is to use the method described above to get an *approximate value* that will be correct within plus or minus approximately 3 per cent. Realizing that this is an approximate rather than an exact method, Eq. (6) may be used, "calling h_{fg} 1,000 Btu per lb." The use of this round figure simplifies the calculation and gives an answer that will not be in error more than 3 per cent. (For h_{fg} values, see Table 1, Chap. VI.)

The latent heat can be calculated by another method, which is somewhat more exact and which is equally easy if a psychrometric table or psychrometric chart is available. This more exact method is based upon the absolute humidity (or water-vapor content in grains per pound) of the outdoor and indoor air. You are already familiar with this method from the work in Chap. VII. The equation to use would be

$$W = \frac{Q}{v} \frac{(G_i - G_0)}{7{,}000} \tag{7}$$

where W = weight of water to be evaporated per hour

$\frac{Q}{v}$ = same as in Eq. (3)

G_i = absolute humidity (grains per lb) of indoor air

G_0 = absolute humidity (grains per lb) of outdoor air

Let us summarize briefly the effect of infiltration air on a heating system or winter air-conditioning system. If we have a simple heating plant, the plant must have sufficient capacity to warm the outdoor air that leaks in to the desired room temperature. In addition to providing this sensible heat, if the plant is a complete winter air-conditioning system, it must provide the heat and the means for evaporating sufficient water into the infiltration air to raise its dew point to the desired indoor dew point.

Indoor Heat Sources. Indoor heat sources would include the heat produced by any machinery or equipment, including lights, motors, etc. In certain cases the heat from occupants might be considered as an interior heat source. With any type of interior heat source, it is necessary that caution be exercised. The important point to be considered is that the heating plant must be able to provide the desired indoor comfort requirement under the actual operating conditions that are likely to occur. Let us consider a theater as an example. Theaters often use many thousands of watts of electric lighting in illuminating the stage; the occupants of the theater furnish additional auxiliary heat.

Suppose that we have a theater using 100,000 watts of stage lighting and having a seating capacity of 1,000 persons. When *all* the *lights* are in use, they supply a certain amount of heat to the stage area, and hence the heating-plant load is decreased by this amount. The heating-plant load for the house proper is reduced by the auxiliary heat output from the 1,000 patrons. If it were certain that *all these people* would be present and *all these lights* would be operating *at all times* it was desired to heat the theater, the capacity of the heating plant could be reduced by the amount of this auxiliary interior heat. However, it is almost certain to be necessary to heat this theater for rehearsals, where *very little* stage *lighting* will be used and the total *occupancy may be* 25 *persons or less.* Also for actual performances, plays may be run which involve little stage lighting, and attendance at some performances may be as poor as 200 persons. Although the theater manager hopes that he will not suffer such poor attendance, the heating plant must be arranged so that it can heat the theater even if this condition does occur.

Let us formulate a general rule regarding interior-heat sources. We may take into account only those interior-heat sources *that are certain* and *can be counted on to exist at all times when the heating plant is to be called upon for its maximum output.* All other interior heat sources—those which are intermittent or uncertain in quantity and duration—*must be neglected*—if the heating plant is to be of conservative design. Even based on this safe rule, interior-heat sources are sometimes of importance and we must learn to calculate them. Some factory buildings using large amounts of power have interior-heat sources greater than the total heat loss of the building under most severe winter conditions. Such buildings need only a heating plant that can keep the building above the freezing temperature when the plant is idle. The heating plant is then *turned off* when the plant is operating.

You learned in Chap. XIII how to calculate the heat that is given off by occupants within a building. Use the sensible heat dissipated by occupants in making a deduction from the heat load on the building. Use only the *minimum number of occupants* that are certain to be present at all times when the heating plant is serving the building. For theaters, convention halls, etc., it is not safe to count on more than 25 per cent of the occupants, and unless this constitutes at least 200 people we often exclude this factor entirely. Otherwise, we are "figuring too closely."

Electric lights or electric-heating units dissipate 3.415 Btu per watt. Since kw equals 1,000 watts, electric lights and heating appliances dissipate 3,415 Btu per kw. For gas-burning appliances, the gas consumption in cubic feet per hour must be known. The heat given off per cubic foot of gas is about 500 Btu per hr for manufactured gas and about 1,000 Btu per hr for natural gas. Remember that heat from gas or electric devices may be deducted from the heating load only if it is certain that the appliances *will be in operation* and contributing their heat to help heat the building.

Heat from power machinery is of assistance in building heating only in the case of industrial buildings. We may then have two different conditions. (1) Where a shaft or belt brings power into a room from a motor or engine located outside the room, the brake horsepower (actual horsepower) must be determined. The Btu supplied per hour is then determined by the following equation:

$$\text{Btu per hour} = (\text{brake horsepower})\ (2{,}456)$$

(2) Where an electric motor driving machinery is located *within a room,*

the heat added is determined as follows:

$$\text{Btu per hour} = \frac{\text{brake horsepower}}{(\text{efficiency of motor})\ (2{,}456)} \tag{9}$$

Summary of Procedure. We have discussed in detail each step of the procedure in calculating the heating load for a building. Let us now summarize these steps briefly in approximately the order in which they would be performed in making the actual design calculations.

1. Select indoor temperature.
2. Select outdoor temperature.
3. Examine construction and determine.
   a. Heat transmission coefficients.
   b. Area of walls, doors, windows, etc.
4. Calculate transmission heat load (see Chap. XIV).
5. Apply exposure factor to transmission heat load.
6. Calculate infiltration and its effect on the load.
7. Determine interior heat sources, if any.

Total all the foregoing, remembering that 7 is a *deduction* if used at all. The only method of becoming proficient at making heat-load calculations is actual practice. Only in this way can the procedure become automatic and the computations rapid and accurate. Practice problems can give you everything except the judgment factors which have been discussed regarding infiltration and exposure factors. We suggest that you make a practice problem for yourself from your own home and place of business. Calculate heating load for the entire establishment, also for individual rooms. The review problems at the end of this chapter present a factory building which illustrates the method of calculating heating loads from beginning to end.

PROBLEMS

Data Regarding Building

Location—Philadelphia, Pa.

Direction of prevailing wind—Northwest

Breathing-line temperature, selected by customer—60°F dry bulb, 40 per cent relative humidity, 5 ft above the floor.

Occupancy—20 to 180 people

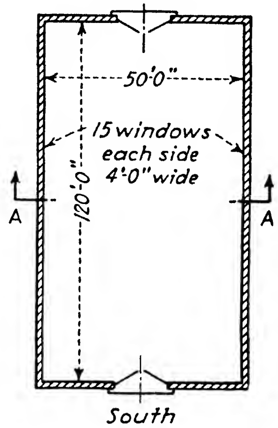
Electric-powered machinery in building—800 hp maximum, 200 hp used during slack periods

Over-all interior dimensions—120 by 50 ft by 16 ft high (see plans)

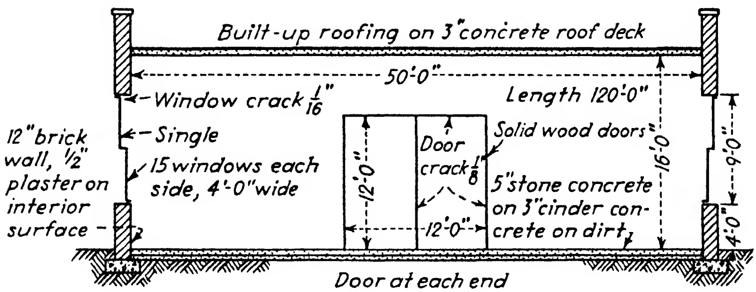
Large doors—The large doors are used by trucks. In cold weather, the trucks drive inside and the doors are closed

Note: Problems 1 to 10 are based upon the building sketched on page 273.

1. Indoor conditions to be maintained at the breathing line are given above. Select the outdoor temperature to be used in making heat-load calculations. Determine the average outdoor wind velocity of the prevailing winter wind. Calculate the following indoor temperatures (see page 264): (*a*) T_c, the indoor temperature directly below the ceiling; (*b*) T_m, the average indoor temperature inside the masonry walls; (*c*) the average indoor temperature applying to the doors; (*d*) the average indoor temperature applying to the windows.



Plan



Elevation A-A

2. Tabulate the construction materials of the walls, roof, floor, doors, and windows. From this information, determine the proper heat-transfer coefficients from Tables XIX to XXIX.

3. Compute the areas in square feet of each wall of the building separately. Wall areas are the *net masonry wall areas*, deducting the area of any windows or doors. Determine also the areas of the doors, windows, floor, and ceiling.

4. Determine the length of crack that you are to use as a basis of computing infiltration. From Chap. XIV, determine a value for infiltration per foot of crack in connection with the windows and with the doors. Finally, determine air leakage in cubic feet per hour.

5. Calculate the transmission-heat loss of the building. Determine the exposure

factor, and increase the transmission-heat loss as indicated by your choice of exposure factor.

6. In Prob. 4 you calculated the air leakage in cubic feet per hour. Now calculate the sensible, latent, and total heat load this infiltration air will impose on the heating system. Determine the rate at which water must be evaporated by the humidifying apparatus.

7. Analyze the data regarding this building, as given above, and determine if any deduction may be made for interior heat sources. If so, compute the amount of this deduction.

8. Make a table summarizing the results of your heat-load calculations, and show the grand total heat load which the plant must supply.

9. In Prob. 1 you computed fairly accurate average temperatures inside windows and doors, and these temperatures were found to be slightly different from the average temperature used as the inside temperature in calculating heat transmission through the masonry walls. Now neglect these refinements and recalculate the transmission-heat loss through the doors and windows. This means that you neglect your previous refinements and *use the same inside temperature for windows and doors as you have already used for the masonry walls.* In your opinion, *for this particular building,* was it necessary to use the extremely accurate method you first used, or could these refinements have been neglected?

10. If this were a new building, not yet constructed, would it be worth while for the owner of the building to use 1 in. of rigid insulation in his roof construction, still having no inside ceiling? How would a metal lath and plaster ceiling compare with the 1 in. of rigid insulation? In other words, if the cost of these two items were about the same, which should the building owner use?

CHAPTER XVI

RADIATION HEATING SYSTEMS, STEAM AND HOT WATER

General. In this chapter we shall discuss certain piping details and certain accessories that are used in connection with the piping in steam-heating systems. We shall then discuss steam-piping systems themselves, comparing the different piping arrangements and the advantages and disadvantages of each. Finally, we shall learn how to select boiler sizes for steam-heating systems.

Accessories and Piping Details. Vent valves of one form or another are essential accessories in almost all steam-piping systems. The different types of vent valves all share one characteristic in common: all vent valves close when contacted with live steam and refuse to let the steam pass outward. Bearing in mind that all vent valves exhibit this characteristic, let us consider the different types of vent valves.

The simplest type of vent valve is known as an "air-relief valve" or "simple air-vent valve." The setting of valves like this is ordinarily between 135 and 175°F. If the valve is cooler than this set temperature, it is open. It closes when it is heated by the presence of live steam.

Vacuum air-vent valves act in a manner similar to simple air-relief valves, but they have one additional feature. If cold, a simple air-vent valve will let air travel *in* through the valve from the room. A vacuum-vent valve permits air to travel out through the valve. It is, however, a one-way valve and will *not* permit air to flow back *in*. Vacuum air-vent valves also close when warmed by the live steam.

Either of these vent valves is available with an additional complicating feature. Ordinary air-relief valves and vaccum air-vent valves require that the air inside the pipe be only a small fraction of a pound higher in pressure than the atmospheric pressure, for air to flow out through the valve. Adjustable air-vent valves of either the vacuum or the simple type have an adjustment screw mounted on the body of the valve. By means of this adjustment, an increased pressure may be required for air to be driven out of the valve. This screw usually may be set requiring any pressure from a fraction of an ounce per square inch up to as high as 1 or 2 lb. psi.

Probably the next most important accessory in connection with steam piping is the device known as a "trap." In general, the function of a trap is to permit condensate (the water produced from condensing steam) to pass through, but to refuse to permit gaseous steam to pass through. Two different types of traps are available to perform this function. Obviously, after considerable steam has condensed within a radiator or a steam-heating coil, there is an appreciable pool of liquid condensate which should leave the radiator or heating coil.

The simplest type of trap consists of a ball or float that rests upon an orifice (opening) having carefully ground edges. If there is no water in this type of trap, the ball makes a tight seal against the rim of the orifice. As the trap fills with condensate, the float rises, opening the trap and permitting the water to flow through. Sometimes float traps of larger sizes are arranged with ordinary valves operated by a lever arm from the float. Modified needle valves are used in some types of float traps.

As you know from the Steam Tables, steam at 5 psig has a saturation temperature of 228°F. This means that steam at 5 psig, condensing in a radiator, turns first to liquid water at 228°F, after which the water cools off somewhat. If the water is led through an exposed uninsulated pipe for a distance of several feet from the radiator, the water will probably cool well below 200°F—it may cool as low as 170°F. This temperature difference between the steam and the cooled condensate is the principle of operation of another type of trap called the "thermostatic trap." A thermostat within this type of trap is set somewhere between 180 and 200°. The mechanism of this thermostat is such that a valve is opened when the temperature is below the setting of the trap thermostat. This means that if live steam is inside the trap the valve will be closed. As soon as an appreciable pool of condensate has accumulated and cooled off (the condensate may only cool to 180°), the thermostatic mechanism opens the valve within the trap and the water passes through.

It may be helpful to think of traps as a barrier between the live-steam piping and the return piping. The trap must function to remove any water, permitting the water to pass to the return pipe lines. The trap must, however, prevent steam from passing into the return line.

In order that air may be removed rapidly from steam-piping systems when it is desired to fill the piping with steam, some float traps contain a mechanism similar to an air-vent valve. It is not necessary for thermostatic traps to be equipped with any special device, as the air in contact with the trap would be cold. In the case of a thermostatic trap, this would mean that the trap would be open. Thus, the air would pass through the main passage of the trap, and the trap would snap shut as soon as steam reached it.

Steam valves are important in all steam-piping systems, and you should be familiar with several types. Steam shutoff valves in steam pipes are different from the ordinary valves used in water piping. Remember, steam valves must stand temperatures depending upon the pressure of the steam in the system. For steam at 100 psig, such as we use on our steam-jet refrigeration machine in the laboratory, the steam is at about 326°F. Shutoff valves in the piping of steam-heating systems may be subjected to temperatures ranging from room temperature to 250°F. Therefore, these steam shutoff valves are of special construction, and care should be exercised not to permit an ordinary shutoff valve to be used.

At the entrance of manually controlled radiators, there is almost always a steam valve. These particular steam valves are called "radiator valves" and are of a somewhat special nature. In the first place, they are available to go in a straight run of pipe and they are also available built into a piping tee. This type of valve serves the function of a piping elbow. When a steam pipe comes up through a floor and must turn 90 deg to enter a radiator, it is convenient to have this type of valve available. As to valve mechanism, radiator valves are divided in two general classes.

The valves found on older systems, but only found on the cheapest systems today, are simple shutoff valves. The operation of these valves is almost the same as that of an ordinary hand-wheel type water faucet. With this simple shutoff valve, it is difficult to obtain any degree of control over the steam flow through the valve. The more modern type of radiator valve has a handle instead of a wheel, this handle being capable of *less than one full turn.* There is often a scale—shut, ¼, ½, ¾, open—marked on the body of the valve. The interior mechanism of these valves consists of a variable size orifice. These valves are called "modulating" or "graduated" valves. They are located at the top, rather than at the bottom of a radiator. Modulating valves may be used only when steam is the sole fluid that is going to pass through. They may not be used if appreciable quantities of water are to pass through the valve.

Steam Heating Piping Systems. *One-pipe Systems.* Figures 1 and 2 illustrate schematically the piping arrangement for one-pipe steam systems. Figure 1 is the simplest type of one-pipe steam system and, hence, the cheapest to install. It is by no means the most satisfactory and can be much improved by a slight increase in installation cost.

In the one-pipe steam system in Fig. 1, the steam leaves the boiler and flows through the steam main from which it goes up the riser pipes to the radiators on the floors above. The condensate returns from each radiator through the risers and into the steam main whence it runs back through the same steam main to the boiler. For this reason, the steam main must be pitched back to the boiler as indicated in Fig. 1. The fact that steam flows in one direction through all pipes, while water must flow in the opposite direction through these same pipes, is the source of most of the disadvantages of this simple one-pipe steam system. Water hammer can result, and the flow of steam is often impeded unless oversized pipes are used. Note the vent valve at the upper corner of each radiator and the simple radiator shutoff valve at the entry of each radiator.

Since each riser serves only one to three radiators, in most cases it is not too great a disadvantage to have all the condensate from these radiators return to the basement through the riser. However, in the steam main near the boiler, all the collected condensate from the various risers is trying

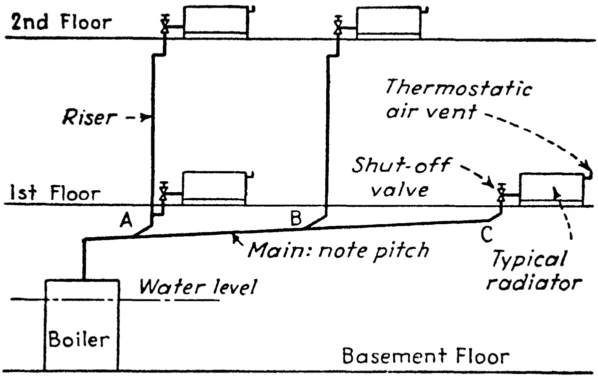


FIG. 1.—Simple one-pipe steam heating system.

to return to the boiler at the same point where all the supply steam from the boiler is trying to go in the opposite direction. By pitching the steam main in the opposite direction, as indicated in Fig. 2, considerable improve-

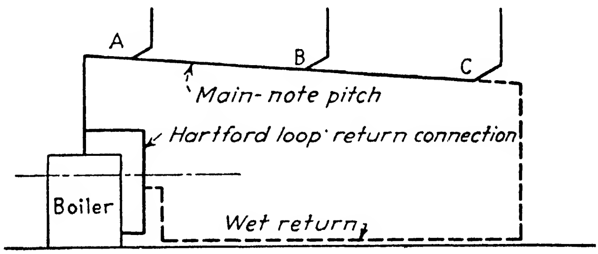


FIG. 2.—Improved one-pipe steam heating upfeed wet return system.

ment is made. The condensate then returns to points A, B, and C from the risers in the same manner as in the system of Fig. 1. Once in the steam main, however, condensate flows in the *same direction* as the steam instead of having to flow against the steam. You will note that the system illustrated in Fig. 2 involves a separate return pipe run from the end of the steam main back to the boiler. As this return is illustrated in Fig. 2, it lies full of water at all times and is, therefore, called a "wet return." If the return were run below the cellar ceiling, it would drain dry and would then be called a "dry return." Dry returns have no particular advantage over wet returns, and the lack of headroom in the basement often makes them undesirable.

The one-pipe steam system, as illustrated in Fig. 2, will suffice for a house that is not too large. It is possible to have two steam mains running in different directions from the boiler so that quite a few risers may be accommodated. Because of its low installation cost, the one-pipe steam system is often used in small apartment houses up to four-story buildings containing 30 to 40 apartments. In such applications, the system, as illustrated in Fig. 2, would suffer from a steam main overburdened with condensate. The final refinement of the one-pipe steam system, used to relieve the steam main of carrying any radiator condensate at all, is illustrated in Fig. 3. This system is known as the "one-pipe relief system." The word "relief" is used because the steam main is relieved of carrying condensate. The only new factor introduced in Fig. 3 is the extending of risers downward to the wet-return line so that condensate flows directly to the wet return without entering the steam main.

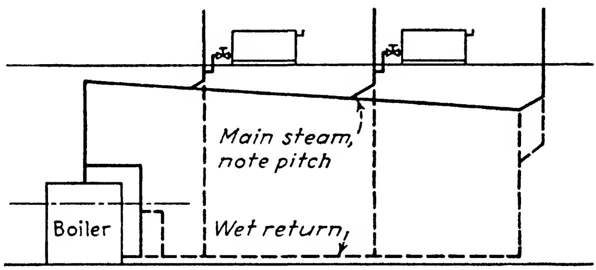


FIG. 3.—Simple one-pipe relief system (end of main, and each riser, is "dripped").

The main advantage of one-pipe steam systems is low installation cost. One-pipe systems do not heat up evenly unless the adjustable air-vent valves, described on page 275, are used. With simple air-vent valves, the steam will reach the radiator nearest the boiler first and all steam will condense in this radiator until it is completely heated. By means of an adjustable air-vent valve, it is possible to require a slight pressure in the steam main before any air can leave the radiators nearest the boiler. This means that the steam will have been forced to the more remote radiators from which air can leave easily and more uniform heating will result. A one-pipe steam system can have radiators either *hot* or *cold*. It is not possible to have a radiator at an intermediate temperature. Since the water must leave the radiators through the radiator valve, it is not possible to use a modulation type of radiator valve, and hence it is not possible partly to fill the radiator with steam. This means that when the outdoor temperature is, say, 35°F the radiators are hot 50 per cent of the time and the steam is off the other 50 per cent of the time. The temperature within the heated rooms, therefore, cycles (or varies) from 1 to 4°. It would be much better if it were possible to heat only a portion of each radiator, and even better if it were possible to have radiators at temperatures varying from 140 to 210°. Both these variations are possible with other types of steam systems.

Two-pipe Systems. Many years ago a separate pipe was provided to carry condensate from each radiator back to the return main. The first two-pipe steam systems were exactly like the one-pipe system illustrated in Fig. 1 or 2, except that a second pipe was added to return the condensate to the basement. This arrangement made the systems more quiet in some

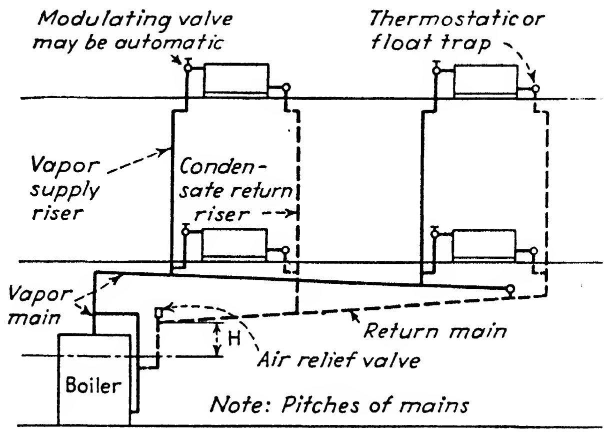


FIG. 4.—Simple two-pipe steam heating vapor system gravity return.

cases and permitted smaller steam-supply pipes, but otherwise gave no advantages whatsoever. Therefore, this simple two-pipe air-vented steam system is now completely obsolete. We mention this system only because installations of this type may be encountered in the field and it is possible to modernize them.

Figure 4 illustrates a two-pipe steam-heating system of the modern type. Note that the steam main is pitched away from the boiler so that any condensate formed by steam condensing in the main may travel in the same direction that the steam is traveling. This system, as illustrated in Fig. 4, is called a "vapor system" because the steam in the supply side of the system is ordinarily at or near atmospheric pressure. Little if any gain is possible by increasing the supply steam pressure above 2 or 3 psig. In the system illustrated in Fig. 4, note that a *modulating* type of radiator valve at the *top* of each radiator is used. Also a trap is used at the outlet of each radiator and at the end of the steam main. This trap at the end of the steam main is to permit any condensate that forms in the steam main or steam-supply risers to flow back to the return side. If the trap were not used, the entire return-piping system could become filled with steam. If thermostatic traps are used (thermostatic traps are least expensive for small sizes and low pressures), air can be driven out of the system through the single air-relief valve shown on the dry-return main near the boiler. In some cases, one return riser is extended above the roof and left open to the atmosphere, in which case the air-vent valve is not needed. Systems with the return side open to the atmosphere are known as "open-vapor systems." Atmospheric air fills these systems whenever the boiler becomes cold.

The vapor system, as we have been discussing it, offers two advantages over the one-pipe steam system. (1) Pickup is considerably quicker because of the fact that no radiator condensate returns through the steam-supply pipe. (The pickup speed of the system can be further increased by locating air-vent valves near the top of each supply riser. These offer speedier renewal of air from the system and, hence, the radiators become filled with steam more quickly.) (2) The radiators may be regulated so that only a portion of the radiator is warm, because the steam flow into the radiators can be accurately controlled by the modulating valves. In some high-class jobs, where installation cost was not considered, automatic modulating valves have been placed on all radiators with a thermostat in each room. This type of installation is too expensive for ordinary use. However, the open type of vapor system still involves steam at a pressure slightly above atmospheric so that the radiator temperature is bound to be above 200°F.

At little additional expense, the open-type vapor system may be converted to what is known as the *closed* vapor system, which offers the advantage of variable radiator temperature. Since more than half of the days during the heating season have only average outdoor temperatures of 30 to 50°F, it is highly desirable that radiator temperature be lowered on these days. This means that the radiators may remain warm more continuously without overheating the heated rooms, and as a result the temperature indoors will show less variation.

To have a closed vapor system, only two things during installation need be done differently from the installation of an open vapor system. (1) Great care must be exercised in making all pipe joints so that they will be absolutely tight. (The reason for this will become obvious in a moment.) (2) Instead of using simple air-vent valves, vacuum-type air-vent valves are used. This means that once air has been expelled from the system no more air can enter except through slightly leaky pipe joints. A pipe joint sufficiently tight to prevent steam from leaking out might still permit air to leak in slowly. Remember that air, once driven out of a closed vapor system, cannot be replaced except by minute leakage. With this in mind, let us trace an operating cycle of a closed vapor system being started for the first time in the fall. As soon as a steam pressure of a few ounces is created, air will be driven out of the system through the air-relief valves. Soon the steam-supply mains and radiators will be hot and the return lines will be carrying condensate plus a small amount of air. The return pipes are small, and the volume is small compared with the volume of the steam-supply pipes plus the volume of the radiators. Let us assume that the boiler is now allowed to cool off. The steam in the system condenses. No air can come in to replace the steam. A pressure gauge on the system would, under these circumstances, drop until it showed a vacuum of 20 to 28 in. Now what will happen if the boiler is again heated up?

As you know from the Steam Tables, the water in the boiler will boil

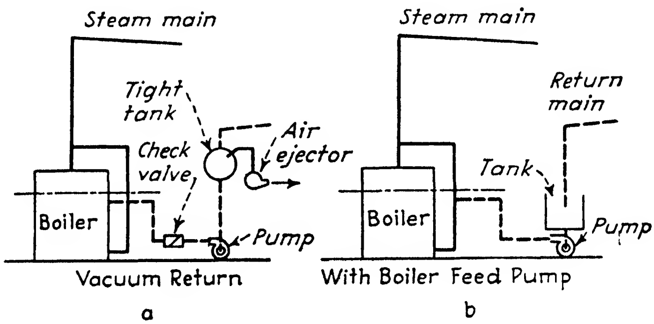


FIG. 5.—Simple closed vapor system (mechanical return).

at a low temperature because of the low absolute pressure over the surface of the water. This means that it is possible to heat radiators with steam at a temperature as low as 135°F. Furthermore, if the piping system is tight, a closed vapor system should hold its low absolute pressure for several days.

The closed vapor system can be used only where the pressure on the supply side of the system (*i.e.*, boiler pressure) is to be held at not more than a few pounds above atmospheric pressure. The reason for this is that the closed vapor system can be used only in the form that we have discussed, where it is possible to return the condensate to the boiler by gravity. The ease or difficulty of returning condensate to boilers depends to a certain extent on the operating pressure of the boiler. Therefore, we shall now discuss various means of returning condensate to boilers under different conditions.

Condensate-return Devices. In most domestic heating systems it is possible to return condensate to the boiler by gravity. In many commercial buildings, however, it may be necessary to use mechanical power for one reason or another to return condensate to the heating boiler. As a matter of fact, it is not essential that condensate be returned to the boiler at all in order to have the heating system operate. In fact, steam used in restaurant kitchens, in steam-jet refrigeration machines, and in other similar applications often cannot be recovered as condensate, so that new city water must be fed to the boiler. The disadvantage of new water is that all water contains dissolved impurities. These impurities are deposited as a crusty boiler scale inside the boiler when the water is evaporated. The problem of boiler scale may be reduced to a minimum if the condensate is recovered and reused.

A clever but simple piping arrangement called the "Hartford Loop" or the "Underwriters' Loop" is used on almost all heating boiler return connections whether of the gravity or mechanical return pump type. The Hartford Loop return connection is illustrated in Fig. 6.

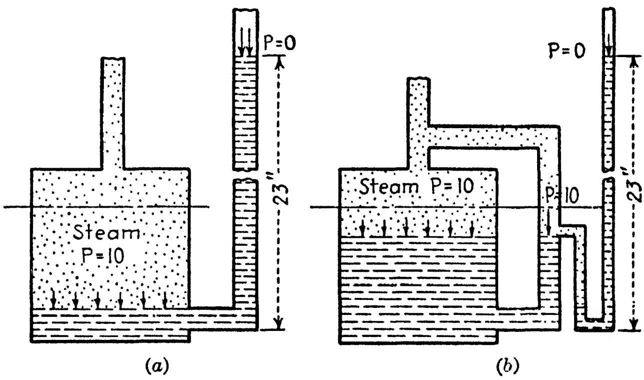


FIG. 6.—*a*. No Hartford Loop; *b*. boiler with Hartford Loop.

When this Underwriters' Loop type of return connection is used, it is impossible to blow all the water out of the boiler by means of an unusually high steam pressure. The Hartford Loop return connection does not prevent water from being backed up into the condensate return lines. It does not prevent the steam pressure either from keeping the condensate in the return lines or from entering the boiler. What it does do is to limit the low water level in the boiler which could be reached by means of forcing water back out of the boiler into the return lines. From Fig. 6 it is clear that as soon as the water level in the boiler has dropped a few inches below normal, so that it reaches the horizontal return pipe connection at the point where this return connection joins the Hartford Loop, the water in the lower section of the Hartford Loop is subjected to the same pressure (*i.e.*, boiler pressure) as the water inside the boiler. The water level will therefore drop no further, as there is no place for the water to go.

Turn back to Fig. 3 and notice the vertical water column H in. high shown in the return line near the boiler. As you know, a pressure of psi will support a water column approximately 2.3 ft high (or approximately 2 ft 4 in. high). Therefore, in a heating system, if the pressure on the supply side is to be 1 lb higher than that on the return side, the water leg or water column, indicated by the dimension H in Fig. 3, must be 2 ft 4 in. If a wet return system is used, the water column in the risers of the return system must also extend 2.3 ft above the water level in the boiler for each pound of pressure difference between the supply side and the return side of the system. In many cases, especially where the boiler is operated at a pressure of 3 to 5 psi continuously, the height of the water column necessary in the return lines is so great that there would be danger of first-floor radiators filling with water (many hotel boilers are operated continuously at this pressure to supply laundry steam.) In all cases where heating steam is taken off through a reducing valve from a high-pressure steam boiler, it is obviously necessary to use some other means than gravity to return condensate to the boiler, otherwise the condensate water leg might have to be over 200 ft high.

Figure 4 illustrates a simple tank receiver and pump system for returning boiler feed water by positive mechanical means. The condensate drains into the open tank, and a float, sensitive to boiler-water level, starts the pump whenever the boiler-water level drops below a predetermined setting. The condensate receiver tank is vented to atmosphere so that the return side of such a system is always at atmospheric pressure. The condensate tank is equipped with an overflow pipe. (Some systems of this type start the pump when a float in the condensate receiver tank reaches a certain predetermined high level. In this case, the boiler must be equipped with a low-level float operating an electrical switch and alarm device.) Our high-pressure steam boiler in the shop is equipped with an open-type receiver tank and a feed-water pump, capable of forcing water into the boiler against the pressure of 100 psi.

Devices are available known as "alternating automatic return traps." These consist of a tank having three external piping connections, a steam line from the steam header above the boiler, a water connection to the Hartford Loop, and the dry (or wet) return main, all connecting to this automatic-return mechanism. No electric pump is used. Instead a float within the return trap actuates a set of valves. The arrangement of these valves is such that when the tank fills with water a valve to the condensate return main *closes* and the valves to the steam connection and the return connection to the Hartford Loop both open. The tank is located at least 3 ft above boiler-water level so that when the two valves open, as just described, the pressure throughout the interior of the tank becomes the same as boiler pressure and the water flows into the boiler. When the water level within the return-trap tank reaches a predetermined low level, the float snaps *shut* the steam valve and the valve in the return line to the

Hartford Loop and then *opens* the valve in the return main to allow the tank to refill with return condensate.

When using the open receiver tank with condensate pump or using the alternating return trap, it is not possible to gain the advantages of a closed vapor system, *viz.*, low temperature radiators and instant generation of steam. Furthermore, the pressure drop across radiators is limited to approximately the gauge pressure maintained in the boiler. From this it would seem that the advantages of the closed vapor system are limited to buildings where a gravity return directly to a Hartford Loop is possible. This is not true, however, for there is one more type of mechanical condensate return device that makes it possible to gain all the advantages of the closed vapor system, plus the additional advantage of a 12- to 13-lb pressure drop from the steam side to the return side, even though the pressure in the steam side is only a few ounces per square inch gauge.

Figure 5*a* illustrates a vacuum return system. Only the boiler and the vacuum return tank are illustrated, because the piping above this point is the same as that of a closed vapor system. The only difference is that the traps must be of a design built to operate with a vacuum on the return side of the system. Let us trace the starting of a cold vacuum system as we traced the starting of a vapor system earlier in our discussion. Before starting the fire in the boiler, the vacuum pump is turned on and permitted to operate 5 to 20 min. When real heat is applied to the boiler, the vacuum pump has already removed all air from the return side of the system. Perhaps the vacuum in the return side might be 26 in. Hg, or an absolute pressure of just less than 2 psi. Even before any steam is generated, the absolute pressure in the supply side of the system would be atmospheric pressure. This would mean a pressure drop from the supply side to the return side of 13 psig. This pressure difference speeds up the distribution of steam throughout the system and aids in passing the initial large "slugs" of condensate through traps.

By slow firing of the boiler (or a reduced heat output from the oil burner or gas burner), the vacuum may be made to extend back to the supply side of the system. This involves the use of vacuum-type air-vent valves at the top of the supply risers, as well as the use of a trap that readily permits the passage of air when the trap is cold.

Thus we might say that the vacuum two-pipe heating system, when arranged so that the radiators may operate at subatmospheric pressures, represents the very best steam-heating system. It has all the advantages of a closed vapor system, but may be used for *large buildings* where the condensate cannot return to the boiler by gravity. Furthermore, uniform distribution of steam to all radiators is assured because of the high vacuum that can be maintained on the return side of the system at all times. The main disadvantage of this system is its mechanical complexity in traps and the vacuum-pump return mechanism. If the vacuum-pump return mechanism becomes defective, the system can be operated as a gravity-return system with a boiler-feed pump, provided that the *water-return* pump is in operating condition. Repair and maintenance of this equipment requires a combined steam fitter and mechanic—the type of man often found as operating engineer in buildings. Unless such a man is a full-time employee in a building, the vacuum two-pipe steam-heating system is not to be recommended. Furthermore, it is not required as most of its advantages (except speed of pickup) can be obtained by the closed vapor system.

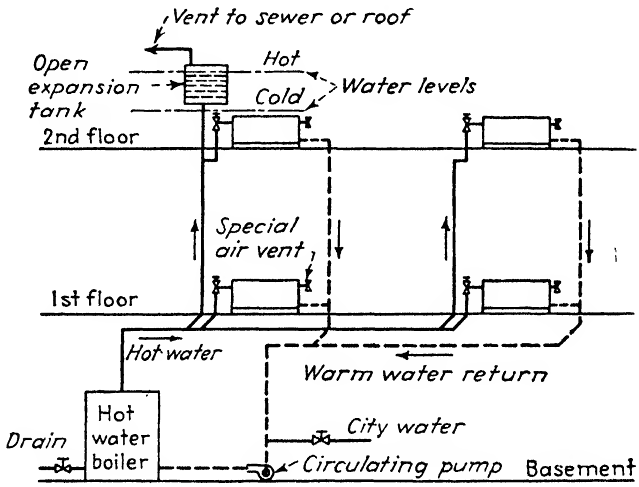


Fig. 7.—Hot-water heating system with open expansion tank.

Hot-Water Heating. Figures 7 and 8 illustrate typical hot-water heating systems. In the past, hot-water heating systems were installed without the circulating pumps shown. The circulation of water was then by natural convection circulation, or gravity circulation. Gravity hot-water heating systems are as obsolete as gravity warm-air heating systems and are no longer being used in up-to-date installations. A hot-water heating system requires two pipes to each radiator, and the return pipe must be *just as large as the supply pipe.* In steam systems, the return line is carrying condensate having a volume of only 1/1,000 (approximately) of the volume of the supply steam. Therefore, the return pipes in a steam system may be much smaller than the supply pipes. Therefore, the *piping* for a hot-water heating system is more expensive than for a steam system. Since the radiators are filled with hot water at about 180°F, compared with radiators being filled with steam at about 215°F, larger radiators are needed for hot-water heating systems. This represents another item of increased cost.

Hot-water heating has the advantage that radiators are never extremely hot and the radiator temperature can be controlled in steps almost as fine as 1° throughout the entire range from 100°F, in extremely mild weather, up to 180°F. Furthermore, on days colder than the outside design temperature for the system, the heat emission from each radiator may be increased by forcing the system and heating the water to as high as 200°F.

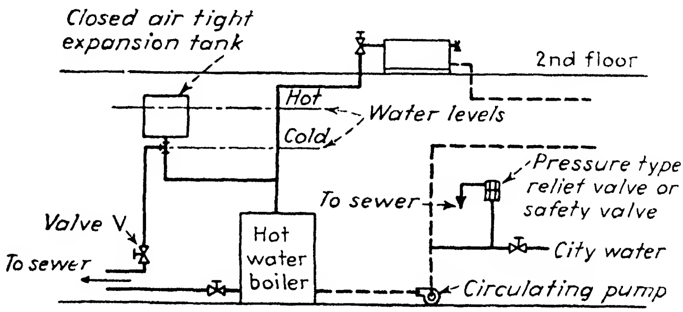


FIG. 8.—Hot-water heating system with closed air-tight expansion tank.

There is no sharp "on" or "off" effect from hot-water heat, whereas even a subatmospheric steam system must turn on and off. Finally, hot-water systems are quieter in operation because the piping never is subjected to sudden changes in temperature.

Special hand-operated air-vent valves are required at the top of each hot-water radiator or heating coil. When dissolved air is released from solution in the water, it tends to rise to the top of various radiators. This air will cause the top of these radiators to remain relatively cold. Air must ordinarily be "bled" from hot-water heating systems at the start of the fall season and once or twice during the season. Contrary to popular opinion, less corrosion (rust) will occur if a hot-water heating system is left filled with water during the summer instead of being drained.

Since the large body of water in the system goes through appreciable changes in volume due to variations in its temperature, an expansion tank must be provided with all the water-heating systems. Figure 7 illustrates the simple open-type expansion tank. Open expansion tanks are not being used as frequently in small domestic installations as the newer closed expansion tank. The closed expansion tank is only partly filled with water when the system is cold. There is a large air space above the surface of the water, and the tank must be *absolutely airtight*. When the water in the system expands, the volume of air is compressed and made smaller to permit the water to expand, and the absolute pressure within the system increases. Because of the increased cost for heating surface and for piping, and also because of the larger *physical size* of heating radiating surface, hot-water heating is not often incorporated with air conditioning on new installations. Steam is preferred instead. For a simple heating installation with standing radiation, hot water represents as fine heating as can be obtained, but the first cost of the system is higher than the most elaborate gravity-return steam system.

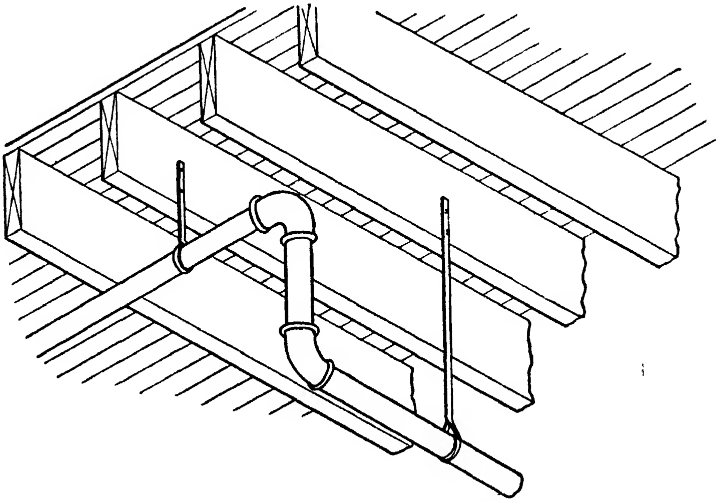


Fig. 9.—Typical "swing joint."

Furthermore, when different groups of radiators and more than one air conditioner are to be operated from one boiler, the control of the system is easier and better in the case of steam. Except to mention hot water briefly in connection with radiators and boilers, we shall not discuss hot-water heating further.

Piping Flexibility. By the term "piping flexibility" we mean actual mechanical flexibility of the pipes as installed, the need for this flexibility being caused by the fact that the piping expands and contracts as it heats and cools. Imagine a large L-shaped section of pipe rigidly fixed at the two ends of the L. This piping is, of course, installed at room temperature. When it is heated to above 200°, each of the straight lengths expands an appreciable amount—a small fraction of an inch. If the right angle of the L is simply a 90-deg pipe elbow and if each end of the L is fixed rigidly, this expansion will cause the point of the L to move and the pipes will bend slightly. This bending produces undesirable stresses in the piping and results in snapping and banging noises when the piping is either heating or cooling. Fig. 9 illustrates a "swing joint" which makes possible the stretching of a large L of piping without any strain on the pipes. You can see that each straight length of pipe in Fig. 9 could stretch as much as half an inch, resulting only in a slight turning in the threads of the fittings making up the swing joint.

Since most steam mains are pitched in one direction or another, it is impossible to bring a riser straight out of a steam main, then up through the floor to the radiator (see Fig. 10*a*). It is ordinarily necessary to come out of the steam main from the side in order that the riser may be truly vertical. This could be done as illustrated in Fig. 10*b*, but we would then have a small L of piping with a 90° rigid elbow. Even in the case of

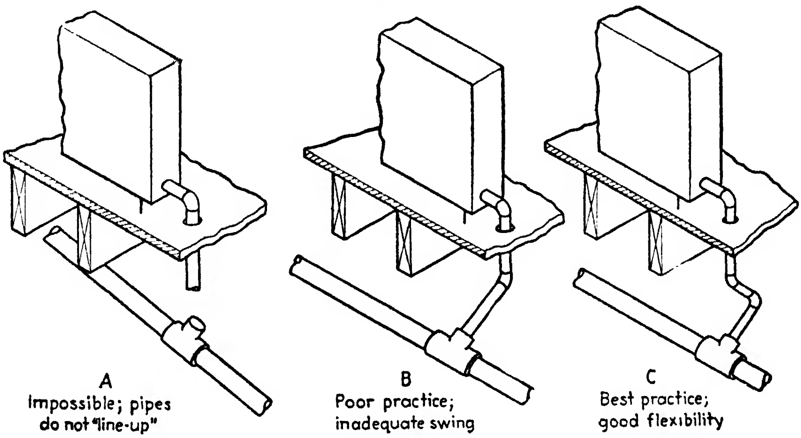


FIG. 10.—Runouts to steam radiators.

a small L-shaped section such as this, swing should be provided because the radiator and the steam main are both probably rigidly fixed. Although the expansion and contraction in a short L-shaped section like this is probably only a few hundredths of an inch, objectionable strains and noise will occur if swing is not provided. A proper runout and radiator connection, using a swing joint, is illustrated in Fig. 10*c*.

Radiators. Ordinary radiators, used *within* the rooms they heat, are divided into three general classes. The old-type cast-iron radiators, fully exposed within the room, are almost obsolete today except in factory-type buildings. Flat, narrow cast-iron radiators concealed behind grilles that protrude only a short distance from the wall, constitute one refinement over the old-fashioned fully exposed standing radiation. Exposed standing radiation emits slightly less than half its heat as radiant heat, the remainder being given to the air by conduction and convection. The enclosed modification of a cast-iron radiator emits even less of its heat as radiant heat and more by conduction and convection.

The final refinement as a radiator for location within the rooms heated is what is called the "indirect" or "convector" type of radiation. Convectors, rarely of cast-iron design, are ordinarily constructed more like a finned extended surface coil. It is difficult to cast the extended secondary surface needed. Convectors are small, but because of the many fins a large area of hot metal is exposed to the air. Although a convector may be only 4 to 8 in. high, it is located in a recess in the wall, the recess being 2 to 4 ft high. The recess is covered by a grille having openings at the extreme bottom and top. See Fig. 11 for an illustration of convector

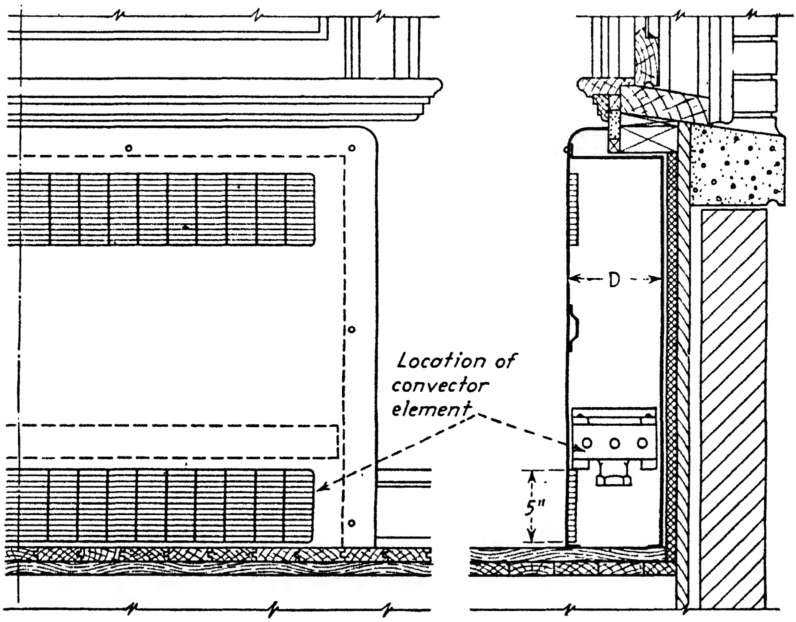


FIG. 11.—Typical convector installation.

installations. A convector emits almost no radiant heat to the room. It is enclosed in the tall recess in order that the warm air rising from the convector may produce strong convection currents. The "stack effect of the enclosed recess is greater with increased height of the recess. The faster cold room air from the floor is made to pass over the surface of the convector, the more rapidly steam will be condensed within the convector, and hence the higher its heat output.

The logical basis for rating the capacity of radiators would be *per thousand Btu per hour* of heat emitted. The A.S.H.V.E. is advocating the use of this accurate standard for rating all types of heating surface. They propose the following abbreviations for the units:

Thousand Btu = Mb
Thousand Btu per hour = Mbh

At the present time, the heating industry is still using an old rule-of-thumb means of rating radiation. Steam radiators are considered to function with a heating medium (steam) temperature of 215°F, and room air is considered to be 70°F. For convectors, which are subjected to the coldest room air near the floor, the air temperature is considered to be 65°F. On this basis, the amount of radiation that emits 240 *Btu per hr* is called "one square foot of *equivalent* direct radiation" (abbreviated E.D.R.). Note the use of the word "equivalent." To say that a radiator has 25 sq ft E.D.R. does not state anything regarding the *actual* square feet of metal surface of the radiator. A 25 sq ft E.D.R. radiator might have from 40 sq ft of actual surface in the case of the poorest possible design to considerably *less than* 25 sq ft of actual surface.

Hot-water radiators contain a heating medium (water) for which the maximum design temperature of systems is usually 180°F. Based upon this heating-medium temperature and upon the same room and air temperatures as given for radiators and convectors above, 1 sq ft E.D.R. for hot water is considered that surface which has a heat output of 150 *Btu per hr*. The old constant of rating in square feet E.D.R. is even worse in the case of hot-water radiation than it is for steam. Steam temperature in a heating system ordinarily varies during severe cold weather from only perhaps 200 to 225°F. Hot-water temperatures, on the other hand, may vary from 130° or lower to above 210°F. The A.S.H.V.E. feels so strongly against the use of E.D.R. ratings for hot-water radiators that data in this connection in the 1939 "Heating, Ventilating and Air Conditioning Guide" are expressed in the new units *Mb* and *Mbh*.

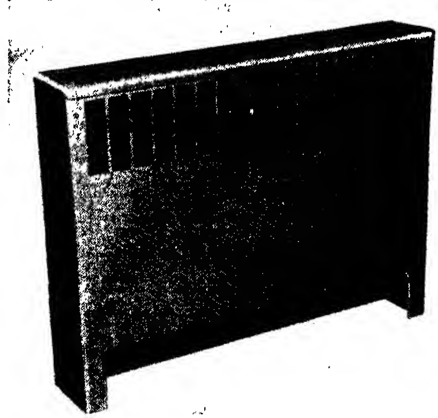


FIG. 12.—Convector cabinet. (*The Trane Company.*)

In purchasing and sizing radiation, especially steam radiation, it is still necessary to use the E.D.R. method of rating. In this case, divide the heat loss for a room, expressed in Btu per hour, by 240 for steam or 150 for hot water and the result is the number of square feet of E.D.R. needed to heat the room. If the heating-medium temperature is greatly different from 215° for steam or 180° for hot water, the number of square feet of E.D.R. becomes *less* with a *higher* heating medium *temperature*, and the E.D.R. becomes *greater* with a *lower* heating medium *temperature*. Also if the room temperature is *below* 70°F, the E.D.R. required is lower; if the room temperature is to be higher than 70°, a greater number of square feet of E.D.R. will be required. Variations from the standard conditions should be referred to the supplier of the radiatio , and the standard E.D.R. should be corrected according to his recommendations, and his guarantee should be based upon the new conditions.

Air-conditioning Heating Coils. The heating coils within air-conditioning units, or the large custom-made heating-coil installations used in connection with air washers and other central systems, are almost invariable of copper construction with various types of extended surface or fins. When these heating coils are installed as an integral part of a factory-assembled conditioning unit, they are ordinarily called "blast coils" or simply "heating coils." When multiple installations of heating coils are custom installed in central-system installations such as air-washer installations, the complete installation of the bank of heating coils is often referred to as a "heating stack." Some years ago it was customary to use cast-iron heating surface with and without fins for this purpose, but cast-iron heating stacks are almost completely obsolete today.

The method of rating air-conditioning steam-heating coils or blast coils is based upon three factors, *viz.*, *air velocity* over the coil, *air temperature* before reaching the coil, and *steam pressure* supplied to the coil. The manufacturers' rating chart then indicates the temperature at which the air will leave the heating coil and gives the number of pounds of steam per hour per square foot of heating coil which will be required. Since these coils have been developed in recent years, their rating system is modern, logical, and based upon the exact Btu basis. In the case of coils within factory-assembled conditioning units, sometimes the entire unit is rated in terms of the cubic feet per minute of air handled and the Btu per hour which the unit can add to these various quantities of air, the ratings being again based upon different entering air temperatures and different steam pressures. (Hot-water ratings are also available for air-conditioning heating coils, although steam is the preferable heating medium.)

Air-conditioning heating coils are ordinarily installed in connection with two-pipe steam-heating systems of the vapor type, condensate-return type, or vacuum type. When installed in this manner, the heating coils are piped exactly the same as a radiator would be piped except that a high-capacity trap is required to pass the large amounts of condensate. Furthermore, instead of the hand-operated modulating valve used with a radiator, the steam flow to an air-conditioning coil is ordinarily controlled by some type of automatic valve. We shall not discuss these control valves at this time but will consider them in a future chapter on automatic-control mechanisms. It is also possible to pipe air-conditioning heating coils if the steam-heating plant available is a one-pipe system. The method of connecting heating coils to a one-pipe system is illustrated in Fig. 13.

Boilers. The obvious exact thermodynamic method of rating the capacity of boilers is to use the number of thousand Btu per hour which the boiler can transfer from the combustion chamber into the heating medium. Since boilers are frequently used for other purposes than residential, and for other low-pressure heating, manufacturers of even small boilers have come to provide ratings on the Btu per hour basis. Smaller boilers, however, are also rated on a basis of the number of square feet of E.D.R. to which the boiler output is equivalent. Since 1 sq ft E.D.R. equals a certain number of Btu, this is merely another way of expressing

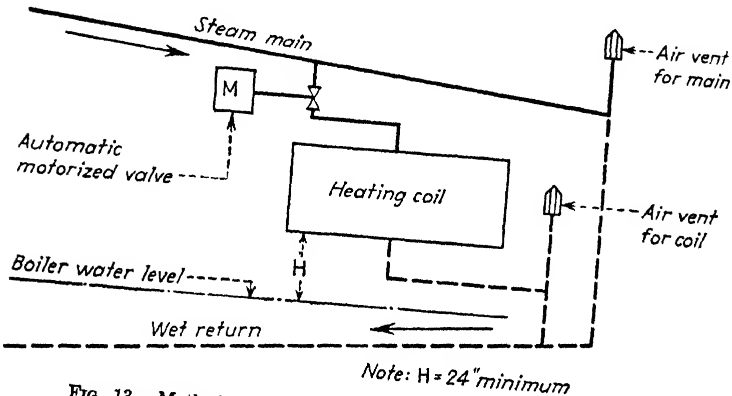


FIG. 13.—Method of connecting heating coils to a one-pipe system.

the boiler capacity. Remember that when a boiler capacity is expressed in square feet of E.D.R. the *standard* E.D.R. conditions are used. These are 240 Btu per sq ft for steam and 150 Btu per sq ft for hot water. From this you can see that the E.D.R. rating of a boiler would be higher (more square feet) for hot water than for steam. Calculate for yourself the E.D.R. rating, both hot water and steam, for a boiler capable of an output of 240,000 Btu per hr.

Heating boilers can be divided into two general classes, *viz.*, cast-iron sectional boilers and welded steam boilers. For many years, cast-iron heating boilers were the standard for all small-size boilers, only the largest boilers being of steel construction. The early steam boilers (and some of them today) were constructed without the use of welding. Only the high development of the welding art today has made possible the low-cost all-welded modern steel boiler in the smaller sizes.

Cast-iron sectional boilers for ordinary sized residences are usually of the round type made up by assembling several disk-shaped or "pancake" sections one above the other, the top section being called the "steam dome." Larger cast-iron boilers consist of *vertical* sections, a design that gives great flexibility in size. With one front section containing the door and one back section containing the flue connection, and with a standard intermediate cast section, a boiler could be built with as few as 3 or as many as 15 sections. You can see the flexibility in size which this permits. The round sectional boilers, on the other hand, are limited in height for the obvious reason of ceiling clearance. Many cast-iron sectional boilers are being manufactured and sold today. For hand firing or for stoker firing, where the fire is never out, these boilers do a satisfactory job. Because the technique of manufacturing them is old and highly developed, and because they consist of extremely rough castings with little machine-work, these boilers are the lowest priced. However, steel boilers have become lower and lower in cost, until today good steel boilers compete with the highest class most expensive cast-iron boilers. Steel boilers offer operating cost savings, and a boiler of approximately 10 per cent less capacity may be used if a steel boiler is selected. (You will see the reason for this shortly when we discuss selection of boiler size.)

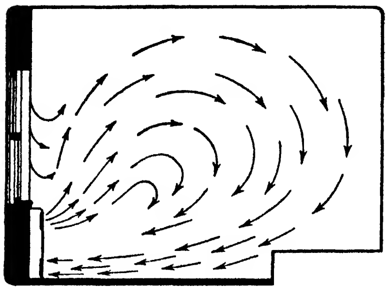


Fig. 14.—Demonstrates the efficiency of convection heat distribution. Note path of arrows carefully.

A few manufacturers of boilers also manufacture fuel-burning equipment, *i.e.*, stokers, oil burners, gas burners, etc. Ordinarily, however, fuel-burning equipment is purchased from a different manufacturer. Therefore, care must be exercised that both the boiler manufacturer and the manufacturer of the fuel-burning equipment agree that their respective equipment will operate in harmony and that both manufacturers guarantee the performance of their respective equipment when working together as a unit. Combustion is easiest to regulate in the case of gas, the problem being simplified by the fact that the gas pressure provided by the gas company supplies the energy necessary to draw the air into the combustion chamber. In the case of both gas and oil burners, less radiant heat is thrown from the flame to the heating surface of the boiler than in the case of a solid-fuel fire. The heat instead is taken from the hot gases, which requires more heating surface for the gas- or oil-fired boiler. Bituminous-coal stoker-fired boilers require a high firebox so that the volatile matter can be burned. Figures 14 to 17 illustrate modern steel boilers from large apartment house sizes down to small domestic boilers. The illustrations also cover boilers built for different types of fuel and fuel-burning equipment.

Selection of Boiler Size or Capacity. Before selecting the size of the boiler, the heating load for the building must be calculated according to the method in Chap. XV. Based upon the heat load for the individual rooms, the radiators are individually sized. The total heating capacity of the radiators should equal the total heat loss of the building. This quantity represents the major portion of what is called the "estimated design load" of the boiler. In addition, the boiler ordinarily supplies domestic hot water either by means of a storage or tank type of heater or by means of an instantaneous emersion coil heater which eliminates

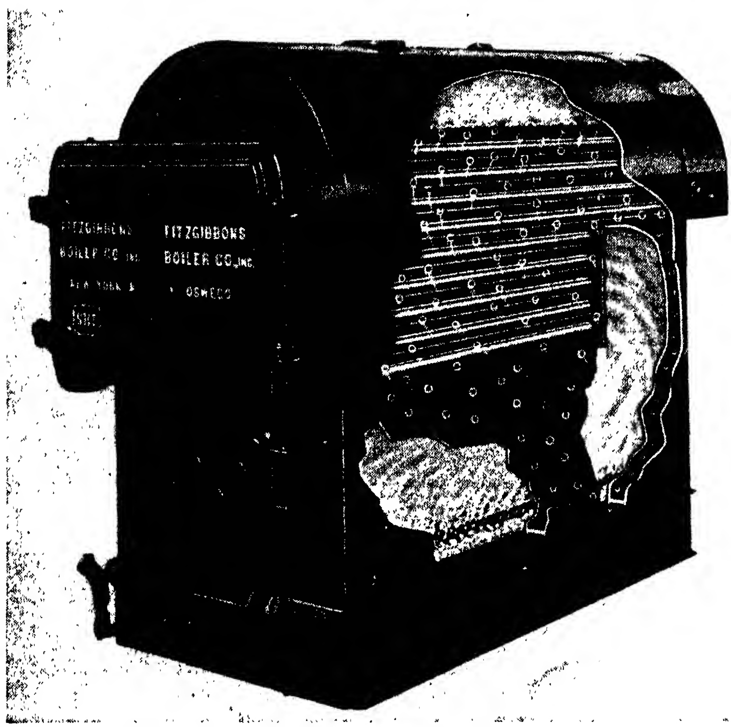


FIG. 15.—The Fitzgibbons D type commercial boiler, coal-fired. (*Fitzgibbons Boiler Co., Inc.*)

the tank. In any event, the boiler must be able to provide the maximum heating load for the building and at the same time heat domestic hot water at the maximum required rate. In the case of devices such as the Fitzgibbons Tank Saver, the boiler manufacturer can best instruct as to the proper allowance for domestic hot-water heating, because this allowance depends upon design and storage capacity of the particular storage-tank eliminating device.

A good approximate method for figuring hot-water supply load (based upon city water at 50°F) is to add 5 sq ft E.D.R. for each gallon of water heated per hour. In other words, if it is estimated that the maximum hot water heat demand will be 30 gal per hr, add 150 sq ft E.D.R. to the building heating load. This method of calculating hot-water supply load allows for heating the 50° city water to 175°F, which is extremely conservative. Actually, the cold water upon entering the heater will probably be at least 10° warmer than the cold water main temperature. It would ordinarily suffice to heat this water through a range of 100°F, or to a temperature of 150 to 160°. In this case, add 4 sq ft E.D.R. per gallon of water heated per hour. When using storage tank hot-water heaters, it is often assumed that the maximum rate of heating will be one tankful of water per hour. This rate applies for average size tanks; for oversize tanks less than one tankful per hour would be required, while with a small

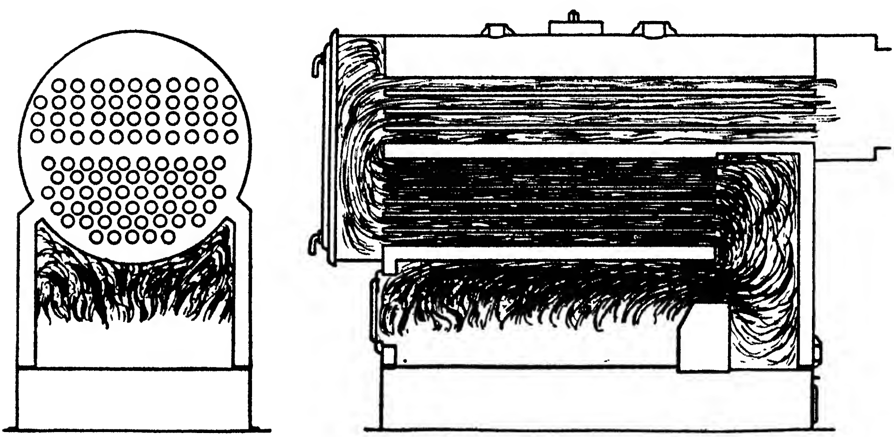


FIG. 16.—Path of flue-gas travel in R-Z-U boiler. (*Fitzgibbons Boiler Co., Inc.*)

tank the maximum demand might be only as much as two tankfuls per hour (on "wash day," etc.).

In order to arrive at the total estimated design load for the boiler we have now two items, the building-heating load and the hot-water supply load. These may be expressed either in square feet E.D.R. or in the more logical rating, simply, Btu per hour. To this total add the heat load required by any other auxiliary apparatus such as air-conditioning heating coils or steam tables in restaurants. The grand total is our *estimated design heating load* and represents the maximum output required of the boiler provided our outdoor design temperatures and other design factors are not exceeded. Although we call this our "estimated design load," if a boiler of just this capacity were selected the performance of the system would be unsatisfactory.

Two further allowances must be made and added to the estimated design load to ensure satisfactory operation. (1) There is the heat loss of the steam-supply piping (or hot-water piping) even though the piping may be insulated. As we mentioned in Chap. XIV, steam pipes are often deliberately left uninsulated in the basement, and in some cases steam risers are not insulated because of space conditions. It is possible to calculate the piping tax or piping loss accurately by a method developed by the A.S.H.V.E.[1]

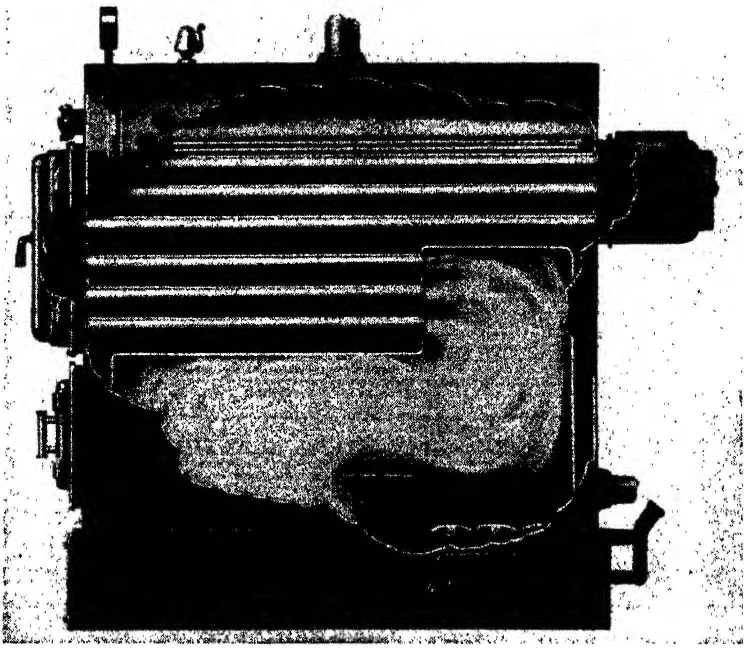


FIG. 17.—Fitzgibbons R-Z-U junior steel boiler for steam or hot water. (*Fitzgibbons Boiler Co., Inc.*)

(2) In addition to the piping tax, a certain allowance must be made in boiler capacity so that the boiler is capable of *warming up* a cold system and a cold building. Since this involves adding thousands of Btu to cold iron in piping and radiators and to cold machinery and building materials, the boiler output must be considerably greater than our estimated design load when the plant is starting operation from a completely cold condition. Even when a heating plant is called upon to warm up a building in the morning (especially on a Monday morning when buildings have been closed for the weekend), the building is usually 10 to 15° below the indoor comfort temperature. This allowance is called "warming-up" or "pick-up" allowance.

Piping loss, while it can be calculated, involves more work than it is

[1] See A.S.H.V.E., "Heating, Ventilating and Air Conditioning Guide," Chap. 39, 1939.

worth because it is a *small portion* of boiler capacity. Pickup loss is really impossible to calculate at all. Therefore flat allowances are ordinarily made to cover these factors, the percentages being based upon those which have worked out satisfactorily in experience. The allowances shown in Table 1 may be used for buildings for the heating loads indicated in the table. However, it is recommended that for heating loads above 5,000

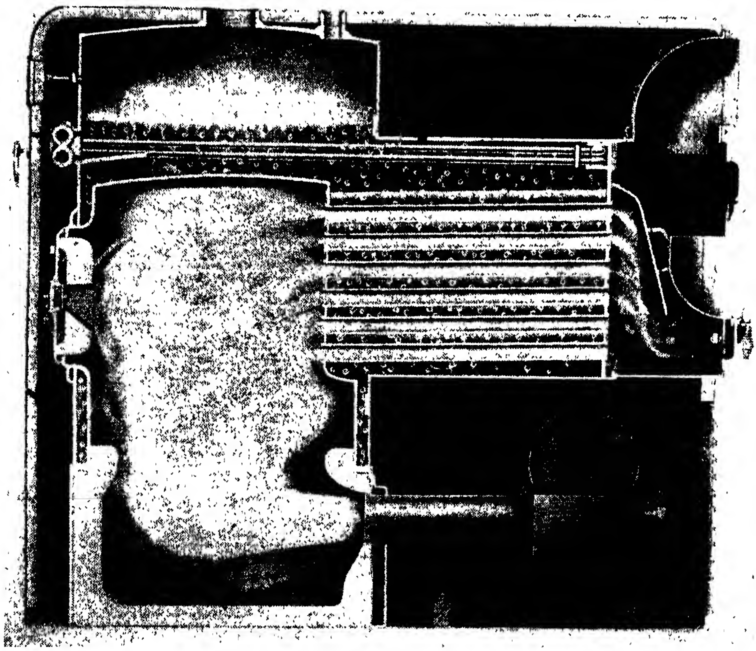


FIG. 18.—The Fitzgibbons oil-eighty automatic with enclosing jacket. (*Fitzgibbons Boiler Co., Inc.*)

sq ft E.D.R. some verification be made by comparing the plant to a satisfactory existing installation.

Radiant Heating. A type of heating system, not entirely new but holding promise, is radiant heating. In this application, panel radiators, heated pipes or flues, are imbedded in walls, baseboards, ceilings, or floors. This system has been used in England and other foreign countries for many years, and it is beginning to find greater use in the United States.

The object of panel or radiant heating is to offset the radiant-heat losses from the human body to colder surroundings. In this way, lower room dry-bulb temperatures may be carried without the attendant stuffiness of overheated rooms. With properly designed radiant-heating systems,

TABLE 1

Estimated design load, sq ft E.D.R.	Per cent to add for (insulated) piping loss and pickup allowance			
	Hand-fired boilers		Automatic fired boilers	
	Cast-iron boilers	Steel boilers	Cast-iron boilers	Steel boilers
Up to 800	60	50	50	35
800–2,500	55	45	40	30
2,500–5,000	50	40	20*	15*
5,000–7,500	45	35	20*	15*
Above 7,500	40	30	20*	15*

* Pickup allowance only—add 10 percent for piping if it cannot be calculated. It is better to calculate piping losses in these large sizes, as it may in some cases be the differences between one boiler size and the next larger.

room temperatures of 60 to 65°F have been found comfortable. Surface temperatures in the neighborhood of 80°F are common.

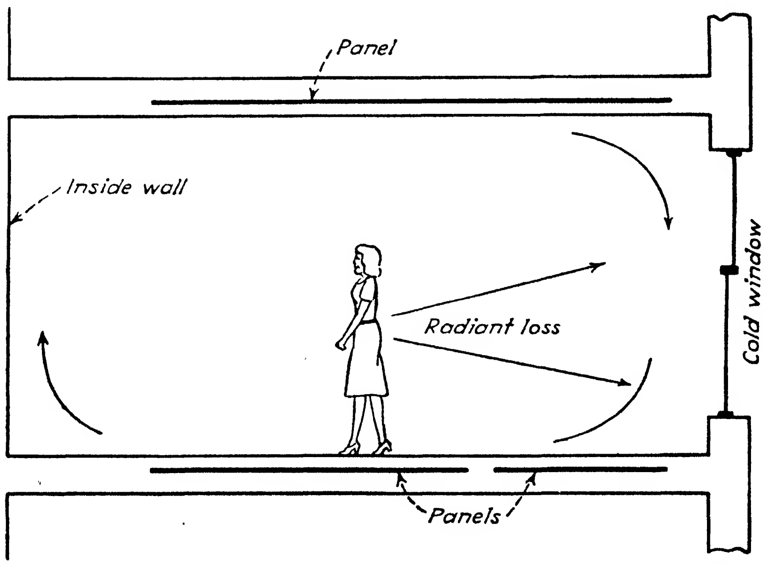


FIG. 19.—Unless the radiant effect is felt on all sides, discomfort will result.

A knowledge of the type of structure, climate, wall-surface temperature, and an evaluation of body-heat losses requires certain data which may be used properly only in an empirical way. The effect of weathering on structures, protracted cold spells, and surface temperatures all present problems. Room occupancy, change of occupancy, large window area, panel inertia, room exposure, and arrangement of furniture all affect the radiation properties of panels.

Since the A.S.H.V.E. comfort chart discussed in Chap. XIII is based on room air conditions with sensible heat control, it follows that this chart becomes void when radiant heating is considered.

Hot water at temperatures of 130°F may be circulated in pipes imbedded in walls, floors, and other structures composed of materials *unaffected by heat.* Circulating steam and hot air through adequate conduits also are used. However, these are only a few methods used in the application of radiant heating. Unless the radiant effect is felt on all sides of the human body, discomfort and poor performance will result (see Fig. 19).

The subject of warm-air heating systems and winter air conditioning, including selection of conditioner sizes, will be taken up in the next chapter. Since this equipment is relatively new, all ratings will be found to be on a Btu per hour basis. Even the expression of the heat required for any one room will be always in terms of Btu per hour. The complication of an arbitrary factor such as E.D.R. ratings will not be necessary in the case of warm-air heating. If you wish to work out an interesting problem before starting the next chapter, determine how much air at 130°F must be supplied to a room if the room is to be kept at 70°F, the heat loss being 13,000 Btu per hr.

PROBLEMS

1. Distinguish between simple air-vent valves and vacuum air-vent valves. What is the purpose of an adjustable simple air-vent valve?

2. We have float traps and thermostatic traps. What actuates each type, and what is the purpose of any steam trap?

3. What are the limitations and disadvantages of a simple one-pipe steam system as illustrated in Fig. 1? What is the only advantage of such a heating system?

4. Explain how a two-pipe vapor-heating system may operate with steam at subatmospheric pressures. What operating advantage does this make possible with this system? This advantage may also be secured with two other types of heating systems. Name them.

5. Two-pipe steam-heating systems, where condensate is returned to the boiler by gravity, should always be piped with a Hartford Loop return connection at the boiler. Why is this essential? Hot-water heating systems never require Hartford Loop return connections. Why not?

6. What is the function of an expansion tank in a hot-water heating system? Give one disadvantage of the closed-type expansion tank and one advantage of this type of tank. Why are gravity circulation hot-water heating systems considered obsolete today?

7. Define "one square foot of equivalent direct radiation" for steam and for hot-water radiators. Explain the difference between convectors and ordinary standing radiation.

If a certain steam-heated convector is rated as 60 sq ft E.D.R., what size hot-water heated convector would be required to have the same output?

8. What type of heating coils are used for air-conditioning units either of the central-system or the factory-assembled ("package unit") type? How are these coils ordinarily rated?

9. A residence has a heat loss of 80,000 Btu per hr. Hot-water supply involves heating 30 gal per hr through a temperature range of 100°F. Add an allowance for piping and pickup to the foregoing, and state the correct *name* for your answer. Assume an automatically fired steel boiler. If boilers are available (in the type you plan to use for this installation) in sizes of 96,000, 130,000, and 163,000 Btu per hr, which size will you select for this installation?

10. Do Prob. 9 over again stating the entire problem, including the boiler sizes, in square feet E.D.R. for steam and in square feet E.D.R. for hot water.

CHAPTER XVII

WARM-AIR HEATING AND WINTER AIR CONDITIONING

General. Warm-air heating systems were perhaps the first type of central-heating plant to be widely used for heating residences. In those early installations the circulation of the warm air was by gravity or convection, with no fan at all being used. Then followed a period of about 15 years when warm-air heating systems were unpopular and were considered cheap and inferior to steam-heating systems. All the disadvantages of warm-air heating systems were traceable to the fact that no fan was used and to limitations in the technique of manufacturing warm-air furnaces. Within the past 10 years warm-air heating, with the added factors of humidity control and air cleaning, has again come into prominence because it offers real advantages.

With the gravity-circulation systems, filters could not be used because filters offer a definite resistance to the flow of air. The flow of air was not positive and certain in amount, so that heat distribution to different rooms was often poor. Because the rate of air circulation was undependable, humidification, properly regulated and always in the desired amount, was impossible. Finally, the design of conditioners was such that, using hand-fired coal-burning furnaces in most cases, dust and in some cases products of combustion mixed with the warm air stream supplied to the heated spaces. The first three of these disadvantages are overcome as soon as a fan is employed in the conditioner. The disadvantage of dirt and possible products of combustion in the warm-air stream has been overcome by the development of steel all-welded conditioning units. The use of filters even makes possible the elimination of the ordinary *atmospheric dust* so that modern winter air-conditioned homes are much more dust-free than homes heated with ordinary steam or hot-water heating systems.

In air-conditioning work today, the gravity warm-air heating system is obsolete. The only need to be familiar with it arises from the fact that old gravity warm-air furnaces are sometimes converted into modern winter conditioners. In other cases, the old-style furnace and all basement duct work is removed and replaced by a new conditioner with modern duct work which can be flush with the ceiling. Although not a disadvantage from an operating standpoint, a minor reason for the unpopularity of the gravity warm-air furnace was the obstruction and spoiling of basement space. The furnace ordinarily was located near the center of the basement so that the duct runs would all be about the same length. Furthermore, the duct runs had to pitch upward away from the furnace, which destroyed headroom at all points near the furnace.

A fan unit consisting of a fan, fan housing, motor, and filters may be installed in connection with an old-style gravity warm-air system. With the air in the ducts then being circulated by means of power, dampers must be installed at every warm-air supply grille so that the proper amount of air will reach each room. Since the air travels through the furnace (no longer a furnace—we are converting it to a conditioner) much more rapidly than under gravity circulation, baffles must be installed to create turbulence and direct the air over the hot surfaces. Provided the old *furnace* itself is in good condition, the above described procedure converts it to a modern warm-air conditioning system. We assume that the humidifying device in the furnace is in good order and that means for controlling it are provided when the installation is modernized.

Modern Warm-air Conditioning systems. In general, we may consider warm-air heating or winter air-conditioning systems in two general classes. In the direct-fired warm-air conditioner no piping or radiators are installed in the building. The fuel burns in the combustion chamber of the conditioner, the chamber having metal walls. The warm air to be heated passes over the other side of these metal walls and is thus heated. With this type of conditioner, it is not feasible to generate domestic hot water by an auxiliary connection to the main furnace. A separate gas or oil-fired domestic hot-water heating unit is required. The reason for this limitation is that it is desirable to *operate the fan* of the conditioner *all year around* in order to ventilate and provide air motion. At times when no heat is required, a fire in the furnace, to provide domestic hot water, would add heat to the circulated air. This would make the house objectionably warm and would waste fuel. It is not possible to operate the fuel burner in a direct-fired conditioner for any length of time without operating the fan that circulates the conditioned air. The metal heat-transfer walls of the combustion chamber would overheat if the conditioned air were not circulating over them.

Figure 1 illustrates the other type of winter air-conditioning system. Note that in Figure 1 there is a complete steam boiler in addition to the conditioning unit. Ordinary steam radiation is provided in the bathroom, kitchen, and garage. In large houses, any servants' quarters are ordinarily also heated by direct radiation. The conditioner contains an extended surface coil heated by steam from the steam boiler. In addition to the steam coil, the conditioner contains simply the circulating fan, humidifying device, and air filters. A split-system installation offers definite advantages compared with a direct-fired warm-air conditioner. As is usually the case, these advantages are found to cost a nominal amount in increased first cost. Current practice is to use direct-fired warm-air conditioners in most residences under approximately $12,000. The split system represents the best winter conditioning, but it is at present in the luxury class.

Direct-fired warm-air conditioner installations can be installed at no greater cost than a first-class one-pipe steam system equipped with a wet return and with relief drips on risers having three or more radiators. With the split-system installation, no independent domestic hot-water unit is

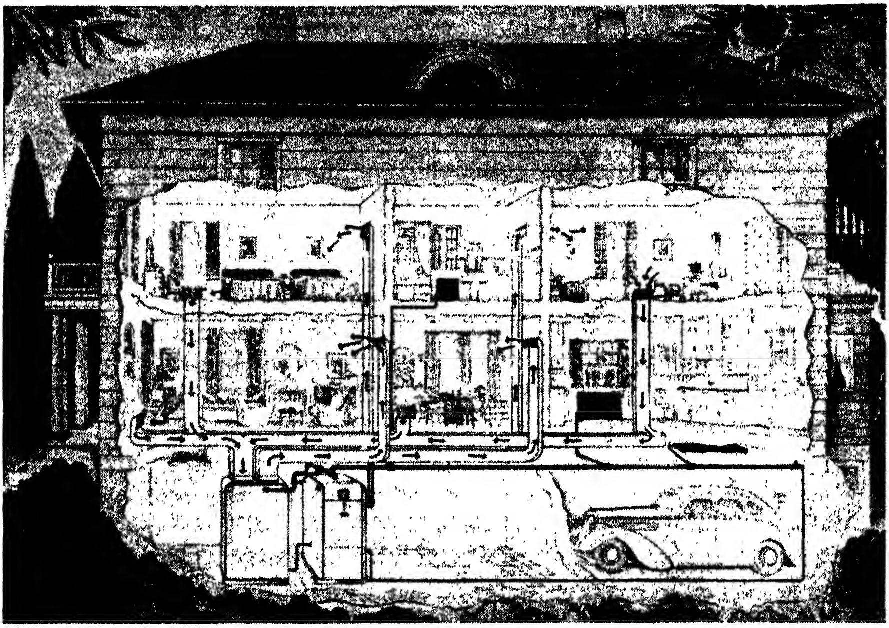


FIG. 1.—Split system—separate boiler with steam coil in conditioner. (*Fitzgibbons Boiler Co., Inc.*)

required, as the steam or hot-water boiler can provide domestic hot-water service. This point is merely incidental, the only saving being space saving in the basement. The hot water still costs money in summer as fuel is burned for this sole purpose.

The duct system, as may be seen from Fig. 1, may be identical for the direct-fired installation. The only difference is that ducts to the kitchen and bathroom are eliminated in the split system and radiators are used instead (Fig. 2, 3, 4).

The design problems of the system then, including duct work, air distribution within rooms, grille locations, heat output of the conditioner, etc., are identical for the two systems. The split system involves the additional steps of selecting a boiler and laying out the piping and radiators for the portion to be heated by steam or hot water.

Oil enters the Superfex oil burner 3 through the oil feed line 1, which is a sturdy half-inch copper pipe with no small openings to clog or out-of-the-way connections to leak. The pipe is welded directly to the burner.

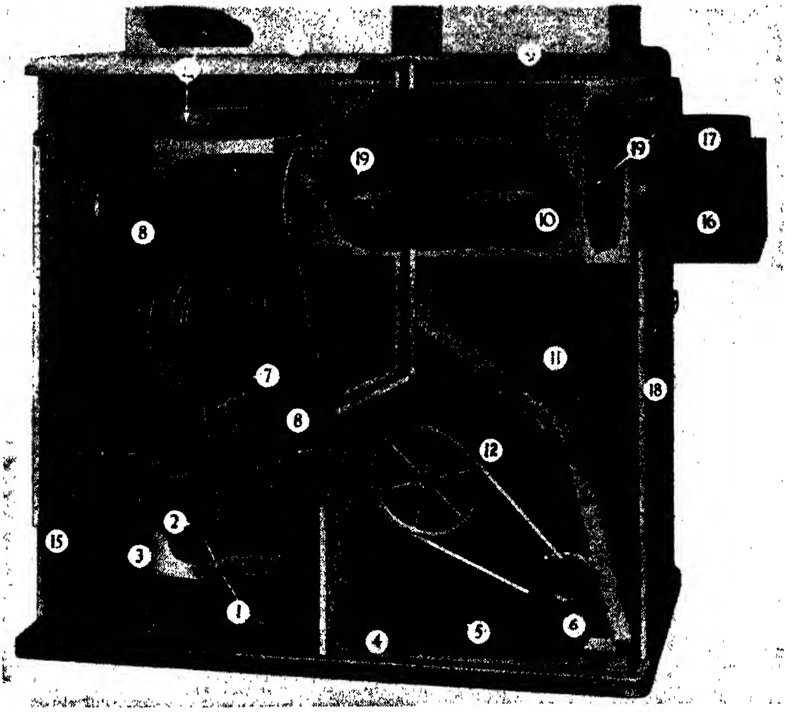


FIG. 2.—Superfex oil-fired conditioner. (*Perfection Stove Co.*)

Air for combustion is supplied through the tube 4 by the small blower 6 attached directly to the shaft of the same motor that drives the conditioned air blower 12. The motor is a double-duty two-speed capacitor motor of extra sturdy construction. The V-type belt drive to blower is silent. As this combination blower is regulated by the automatic high-low control, the combustion air, as well as the conditioned air, always is in correct proportion to the fire. Adjustment to individual draft conditions is made at the time of installation by means of control 15.

The Superfex principle of combustion gives a large "suspended" radiant flame. The baffle 7, which is made of long-life stainless steel, directs the flame toward the walls of the combustion chamber 8. The flame does not actually touch the walls, however, and cannot burn them out. This makes it possible to use comparatively thin walls of highly conductive steel, which quickly and efficiently transfer the heat to air passing around the outside of the chamber walls. The combustion chamber is shaped to provide the maximum heat radiating surface.

Combustion gases on the way to the flue pipe 17 pass through the several baffled sections of the welded steel heat economizer, which utilize heat that otherwise would be wasted, to reheat the return air coming through return air intake 9. The heat-economizer sections are easily cleaned through cleanout openings 19. The draft is regulated by an automatic draft control 16.

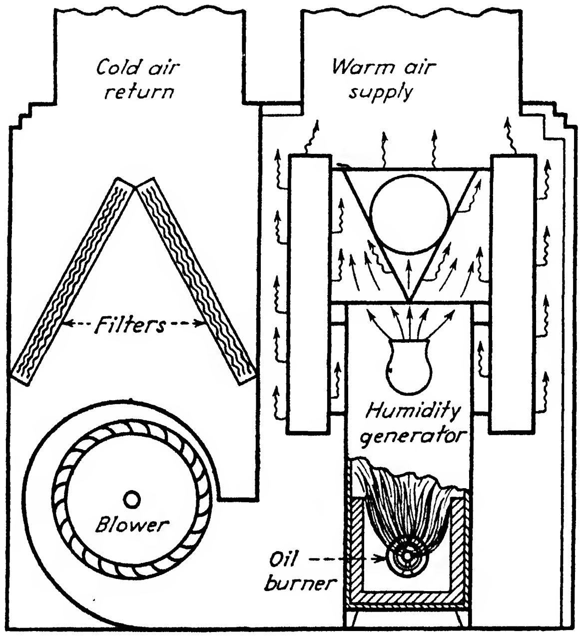


FIG. 3.—Arrows indicate flow of air. Those with straight tails are products of combustion, while those with wavy tails, conditioned air.

The preheated air is drawn through the filters 11 by the blower 12 then passed over the surface of the combustion chamber and the wiping surface of the economizer. The filtered and warmed air receives its final conditioning as it passes through the vapor created by the humidifier 13. The humdifier pan is in direct contact with the top of the combustion chamber. Water supplied through the automatic valve 13 drops upon the hot surface of the pan and is rapidly vaporized. The double-wall casing is indicated at 15. Filters and blowers are readily accessible through doors 18.

The conditioned air—filtered, warmed, and humidified—is delivered through outlet 14 every minute of the day and night to all parts of every room where the gentle but continuous air movement positively prevents stratification (layers of air at varying temperature).

Air Distribution within Individual Rooms. We shall first consider air distribution within a room in a general way, *i.e.*, the path or travel of the

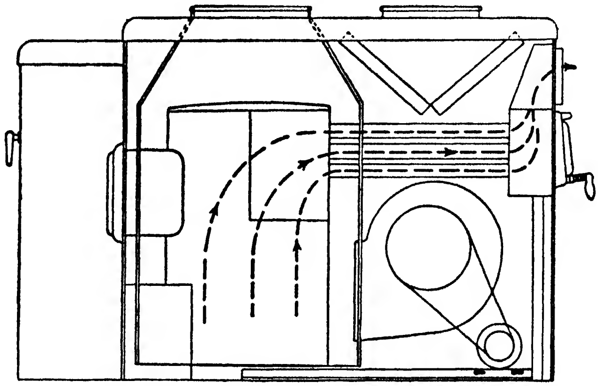


FIG. 4*a*.—Flow of flue gases through standard Directaire. Air is drawn by fan through filters and then across tubes containing hot gases. (*Fitzgibbons Boiler Co., Inc.*)

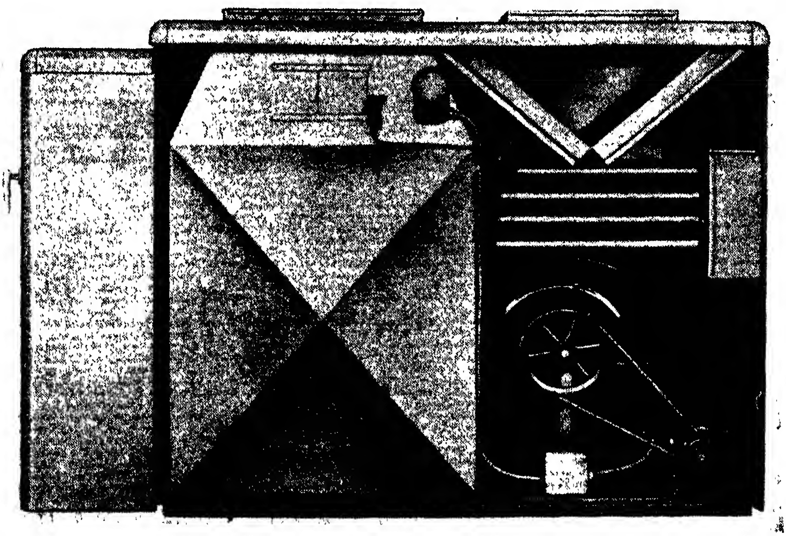


FIG. 4*b*.—Standard Directaire conditioner. (*Fitzgibbons Boiler Co., Inc.*)

warm air within the room, regardless of the exact location of the supply and return grilles. The question of the exact location of supply and return grilles deserves separate consideration. There are two means of providing general air distribution within a room which have worked out satisfactorily in practice.

One method consists of locating supplies on inside walls, as shown by the duct system in Fig. 1. This is done because the heat loss from the ducts will be less if they are located on the interior of inside walls. Furthermore, any heat lost from the duct goes toward heating the house when the ducts are located in this manner. With this supply-duct arrangement, the returns are located on outside walls. This situation is desirable because the return-air duct does not contain *hot* air; therefore heat losses are less because the duct is within a cold wall. Since the air enters the room at an inside partition, coverage of the entire room by the warm air is assured if the return air is removed from the room at an outside wall.

A newer method of distributing air within rooms, which has received considerable favorable comment recently, consists of locating warm-air supply grilles on inside partitions *near an outside wall.* In this manner the warm air blows down or sweeps along the cold wall. In this case the return-air grilles must be located in a manner that will ensure air distribution throughout the entire room with no dead pockets.

Grille Locations, Height above Floor. Warm-air conditioning systems will function with what are called "high-supply grilles" and "low-return grilles," or with low-supply grilles and high-return grilles. They will also function if all grilles are low. By low grilles we mean grilles located within 24 in. of the floor. By high grilles we mean those located so that the bottom of the grille is at least 5 ft above the floor and more likely 6 ft or more above the floor. These two methods of locating grilles, using high-supply grilles and low grilles, are directly opposite. In spite of this fact both will function, and there is considerable argument in heating circles as to which is best.

If high-supply outlets are used, the velocity of the air leaving the grille must be at least 600 fpm in order that this air can be *thrown across* the room, dropping lower as it crosses the room, ensuring no cold layer of motionless air near the floor. This high velocity causes more noise than a velocity of 300 to 400 fpm. Extreme care must be exercised in locating the return grilles to avoid the warm air shooting across the room and entering a return grille without having circulated throughout the entire room. The main advantage of high-supply outlets is that they are definitely superior if summer-cooling facilities are added to the system. Considered solely from the *winter* air-conditioning viewpoint, high-supply outlets are *not so good* as low-supply outlets in providing uniform heating throughout the entire room and avoiding stratified (motionless) cool air layers near the floor.

With low-supply outlets, three different types of grilles may be used. The first type is a straight-throw grille from which the air leaves in a horizontal somewhat fanlike pattern. As these supply outlets might blow their air directly at persons in the room, the grille velocity must be kept very low and no obstruction (furniture or occupants) may be placed too near the grille. A somewhat better type of grille is the downward deflecting grille which deflects its stream of air downward toward the floor. With

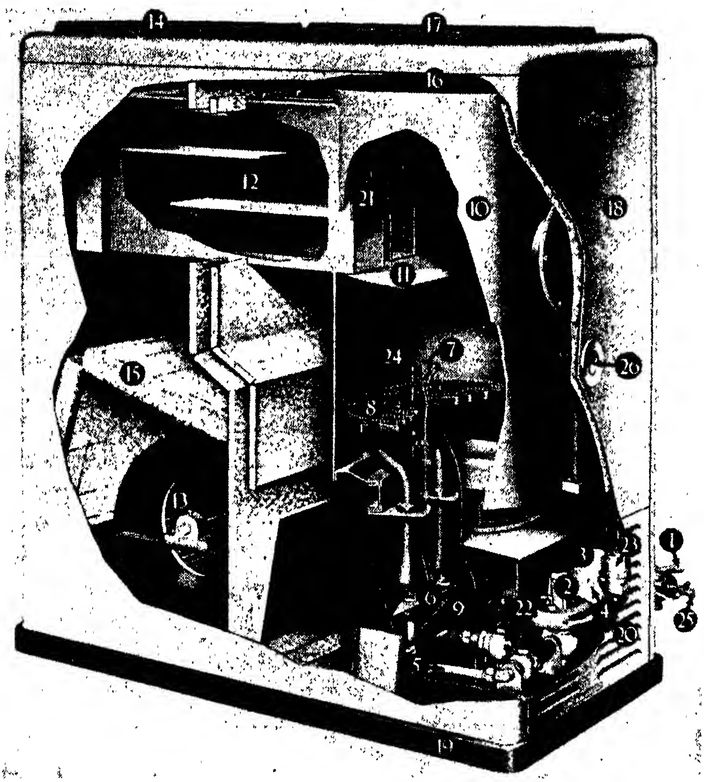


FIG. 5.—Superflex gas-fired conditioner. (*Perfection Stove Co.*)

1. Main gas shutoff valve
2. Pressure regulator
3. Factory-wired control panel
4. Pilot-lighter button
5. Gas lines to burners
6. Adjustable primary air shutters
7. Thermal control element
8. Main burners
9. Protecting baffle
10. Combustion chamber
11. Combustion chamber baffle
12. After-passes of combustion chamber
13. Circulating blower
14. Return-air connection
15. Air filters
16. Humidifier pan
17. Warm-air plenum chamber
18. Removable front door
19. Airtight base frame
20. Removable control panel
21. Flue-chamber cover
22. Electric solenoid valves
23. Fire-stage controller
24. Pilot-fire tube
25. Pilot-light valve
26. Burner observation window

this type of supply grille, higher velocity may be used, as the warm air stream breaks up and mixes with room air as soon as it hits the floor. Finally, if there is a large quantity of air to be supplied through a grille of limited size, such that the velocity must be very high, a third type of grille may be used. This type *deflects air downward* toward the floor and *also diffuses it sideways* in a wide low velocity fanlike pattern.

Using any of the low-supply outlet grilles, but preferably a grille that at least deflects the air somewhat downward, it is possible to stir up the air near the floor and ensure that no cold strata exist adjacent to the floor. Return registers may be located directly across the room from the supply outlets in the case of low-type supply outlets. Two supply outlets, one at each end of a wall, used in conjunction with one return register in the middle of the opposite wall, provide ideal air distribution within a room. Naturally, structural conditions often make it necessary to depart from this arrangement, but many arrangements as to grille location will provide satisfactory distribution of the warm air.

Humidification and Automatic Control. Humidification devices are ordinarily built into modern warm-air conditioning units. Hence the type of humidifying device will depend upon the manufacturer of the conditioner selected. The control of the humidification equipment will depend upon the type of humidifier, so that control also may vary depending upon the manufacturer of the conditioner. Control may be manual in some cases and automatic in others. We shall discuss the question of control, not only of humidification but of temperature and with regard to safety of operation, in a future chapter on Controls.

Humidifiers may be of the pan or nozzle type, the names of which are self-explanatory. Some humidifiers consist of a device from which water drips slowly on hot surfaces within the conditioner. Pan-type humidifiers are arranged so that the water is at various temperatures from 100 to 212°F, depending upon the design. Nozzle-type humidifiers are most sensitive to control because they can be *completely shut off instantly* and because they are operating at *full capacity as soon as they are turned on.* Offsetting this advantage, nozzle humidifiers have the disadvantage of being more expensive and of offering the most possibilities for operating troubles and maintenance.

Duct Design; Air Circulating Fan. At the present time, we shall not give the complete details regarding duct design. For purposes of this chapter, confine yourself to selecting the locations of the supply and return grilles in rooms and drawing a single line to indicate where the duct might be run from each grille to the trunk duct in the basement. The size of ducts and grilles may be determined (approximately) based on Table 2.

Duct work, like water pipes or any other passage conveying a moving fluid, offers resistance to the flow of air. The fan must put sufficient energy into the air so that it can overcome this resistance. The two most widely accepted methods of duct design, by which exact duct sizes are selected and the resistance to flow is calculated, will be taken up in a future chapter regarding the flow of air, water, and other substances through ducts and pipes. For the present, it is sufficient to recognize that ducts *do cause resistance* and that if the ducts are small and restricted the resistance is greater. In any event, resistance of duct work may be calculated accurately and is to some extent controllable by the designer of the system.

We have said that filters cause a resistance to air flow. In addition, the zigzag, direction-changing path of the air through the conditioner itself offers a certain amount of resistance to flow. In the case of direct-fired conditioners and many conditioners heated with steam coils, the fan is built into the conditioner. In this case, the manufacturer of the conditioner calculates the *internal* resistance of the conditioner, including filters. He then states how much *external* resistance, caused by duct work and grilles, can be imposed on the conditioner without reducing the air flow seriously. If the duct system is extensive and the external resistance unavoidably high, it is always possible to change the operating conditions of the fan so that this higher resistance can be overcome.

In the case of some large conditioners, the fan is not furnished as part of the conditioner. In this event, the resistance caused by the conditioner must be determined from data published by the manufacturer of the conditioner. The fan must be chosen to operate against not only the duct-work resistance but *also* the *resistance of the conditioner.* In the case of converting old-style gravity warm-air heating systems, the fan must take care of the resistance of the conditioner, but the filter resistance is part of the resistance of the fan unit itself. A resistance of 0.25 in. of water is a fair average for the resistance of a converted baffled warm-air conditioner. Approximately this amount should be added to the resistance of the duct work in selecting the operating conditions for the fan unit. Manufacturers of fan and filter units for conversion purposes have good data available on the proper resistance allowance for the converted conditioner itself.

Design of Winter Air-conditioning System. *Calculation of Heat Load.* The heating load for the building is calculated by the method developed in Chap. XIV. These calculations should be made room by room. Be sure to include a calculation of the infiltration air in cubic feet per hour, this calculation being made for each of the rooms subject to air leakage. In addition to the heat load calculated thus far, any fresh air that is introduced by means of the conditioner fan must be heated by the conditioner. Very little fresh air is introduced in this manner in residences, but the quantity should not be ignored or there is danger that the conditioner may be undersized.

Location of Grilles. Keeping in mind the discussion regarding air distribution and grille location, *tentatively* locate the supply and return registers in each room. Then plan risers or wall stacks to connect the grilles and registers to the trunk ducts in the basement. Your planning in this connection probably will have to be revised after you select the size of your supply grilles and ducts.

Temperature of Air at Registers. Any reasonable temperature may be selected at which air is to leave the supply registers in the heated rooms. Temperatures from 110 to 160° may be selected, bearing in mind that the temperature will affect the air quantity circulated. Air must leave the *conditioner* at a temperature *higher than* the desired *register temperature.* Experience has shown that between the conditioner and the registers heat losses from uninsulated duct work cause the air temperature to drop 0.3 to 0.6°F per ft of travel through the ducts. We suggest using a temperature drop of 0.5°F per ft of duct as an average factor for uninsulated duct work.

Selection of Air Quantity per Room. The air quantity supplied naturally depends upon the temperature of the air leaving the supply grille. There is less variation in the air temperature at the return-air registers, but this is also a factor. It is common practice to take the recirculated air temperature as 65°F. The supply-air temperature may be anywhere from 110 to 160° or even slightly higher. The supply-air temperature selected depends somewhat upon the number of air changes or complete air recirculations desired per hour. The higher the supply-air temperature, the smaller the quantity of air to be supplied and therefore the smaller the duct work. Hence the highest temperature at the registers should be selected which will still ensure at least *five recirculations of the air in the heated building.* In other words, if a house has an internal volume of 14,000 cu ft, the total air circulated in the heating system should be at least five times this amount, or 70,000 cu ft per hr. The highest register temperature that will provide a quantity of air circulation conforming to the above standard should be selected, but 170° should be set as a maximum not to be exceeded.

In our fundamental heat-transfer equation, Chap. V, let us substitute H for Q as the quantity of heat. Rewriting that equation we then would have

$$H = cW\,(T_2 - T_1) \tag{1}$$

If we wished to rewrite this in order to show the heat transferred to a certain air quantity in cubic feet per minute Q and if we wished the heat to be expressed in Btu per hour, we could modify Eq. (1) to the form shown in Eq. (2).

$$H = 60\,(0.24)\,W\,(T_2 - T_1) \tag{2}$$

where H = heat transferred *per hour*
60 = number of minutes in 1 hr
0.24 = c = specific heat of air
W = weight of air, lb per min
T_2 = initial air temperature (at register)
T_1 = final air temperature (at return grille)

The number of pounds per minute of our air, W in Eq. (2), is equal to the number of cubic feet per minute of air multiplied by the specific volume of the air. Explaining this as an equation, we have

$$\text{cfm} = Q = W(v) \tag{3}$$

where Q = air quantity, cfm
W = same as in Eq. (2)
v = specific volume of the air

If we divide both sides of Eq. (3) by v, we then have

$$W = \frac{Q}{v} \tag{4}$$

Now go back to Eq. (2) and make the following substitutions: for W, substitute Q/v; for T_2, substitute t_r, meaning temperature of air at register; for T_1, substitute just 65°, this being the temperature of the recirculated air, which we have assumed. We now have an equation that looks quite different, so we shall give it a new number.

$$H = Q\ (60)\ (0.24) \left(\frac{1}{v}\right) (t_r - 65) \tag{5}$$

If we solve Eq. (5) for Q, *the air quantity in cubic feet per minute,* we get Eq. (6).

$$Q = \frac{H}{60\ (0.24) \left(\frac{1}{v}\right) (t_r - 65)} \tag{6}$$

where Q = air quantity, cfm, measured at temperature t_r
H = heat quantity, Btu per hr
v = specific volume of the air (disregarding water-vapor content) at temperature t_r.
t_r = temperature of the air at the supply grille or register

Equation (6) is to be used to calculate the cubic feet per minute of air measured at the register temperature which must be supplied to each room in order to take care of a heat loss of H Btu per hr. Study the denominator of the fraction, which comprises the right side of Eq. (6). Obviously the figures 60, 0.24, and 65 are constant and do not change. If we select a value for t_r, the temperature of the air leaving the supply grille, we also fix the specific volume v of the air, because specific volume is a function of temperature. For instance, if we pick a register temperature t_r of 140°F we find the specific volume of dry air at this temperature to be 15.1 cu ft per lb (see the Psychrometric Tables in the Appendix). For a register temperature such as 140°F, or for any other register temperature, the *denominator of the fraction* comprising the right-hand side of Eq. (6) becomes a simple number, or factor. Equation (6) may be rewritten thus

$$Q = \left[\frac{1}{60\ (0.24)\left(\dfrac{1}{v}\right)(t_r - 65)}\right](H) \tag{7}$$

In this rewritten form of Eq. (6), the entire large fraction enclosed in the brackets is a constant factor for any fixed register temperature. Suppose that we call this constant factor k.

We now have

$$k = \left[\frac{1}{60\ (0.24)\left(\dfrac{1}{v}\right)(t_r - 65)}\right] \tag{8}$$

If we choose a register temperature of 140°F, v is 15.1 cu ft per lb. Substituting these numbers in Eq. (8), we have

$$k = \frac{1}{60\ (0.24)\dfrac{1}{15.1}(140 - 65)}$$

$$k = \frac{1}{71.4} = 0.0140$$

In a similar manner we may calculate values for this factor k based upon any other register temperatures desired. We have calculated these values

TABLE 1.—k FACTORS FOR FORMULA 8

Register Temp. t_r	Factor k
110	0.0221
120	0.0184
130	0.0158
140	0.0140
150	0.0125
160	0.0114

and present them in Table 1. Table 1 is to be used in connection with a simplified form of Eq. (6) which we now write as Eq. (7a).

$$Q = k \times H \tag{7a}$$

where Q = same as in Eq. (6)
H = same as in Eq. (7)
k = factor to be taken from Table 1

Selection of Air Velocities in Ducts and through Grilles. In the preceding section, we have learned the method for determining the *air quantity* to be supplied to each heated room. Once this *quantity* is fixed, it is obvious that if we use a *small-size duct* the air will travel at a *high velocity,* while if we use a much *larger duct* the *air will travel more slowly* through the duct. The same is true of grilles and registers; a small grille for a fixed quantity of air means a high velocity, while a larger grille would mean a lower velocity.

In the case of ducts, the air may travel through the *entire area* within the duct. In other words, if you have a duct 6 by 12 in. in cross-sectional area, this duct has a cross-sectional area of ½ sq ft. Air will travel through this entire ½ sq ft of area since there are no obstructions inside an ordinary duct. In the case of a grille or register, there are various bars and deflecting vanes which *partly obstruct the grille opening.* Hence if you have a grille that measures 12 by 6 in. as the over-all dimensions of the opening, it is correct to say that the grille has a gross area of ½ sq ft. However, a portion of this area is obstructed by the bars or deflecting vanes making up the grille face. The air can travel only through the opening between these obstructions. It is necessary to know the total area of these *actual available openings.* This quantity is called the *net free area* of the grille.

Frequently net free area of grilles is expressed by what is known as a "grille coefficient." Grilles and registers are made having net free areas varying from about 65 to 85 per cent of the gross area. When you purchase a grille, the grille coefficient is stated by the manufacturer either as a decimal or as a per cent. If the manufacturer does not present the information in this manner, he presents the actual *net free area* of the grille *in square inches* (or in square feet). Let us suppose that you purchase grilles for which the over-all dimensions of the opening are 12 by 12 in. Suppose that the manufacturer states that these grilles have a net free area of 108 sq. in. Since the over-all area of the grille is 144 sq in., or 1 sq ft, we may determine the net free area in square feet as follows:

$$\left(\frac{108}{144}\right) 1 = 0.75 \text{ sq ft}$$

If, on the other hand, the manufacturer had stated that these grilles have a grille coefficient or a free area coefficient of 75 per cent (sometimes expressed 0.75), we would have determined the net free area of the grilles as follows:

$$0.75\ (1) = 0.75 \text{ sq ft}$$

We may express the information we have just discussed as an equation.

$$\text{Net free area} = k(A) \tag{9}$$

where A = gross area of the opening of the grille as measured, sq ft
k = grille coefficient or free area coefficient as a decimal

It is clear that $k = 1.00$ for ordinary ducts having no internal obstructions. We may then set up a simple equation that gives the relation between air quantity in cubic feet per minute, free area through which the air may travel, and the velocity at which air will travel.

$$Q = k(A)(V) \tag{10}$$

where Q = air quantity, cfm [same as in Eqs. (6) and (8)]
k = free area coefficient
A = cross area (or over-all area) of the opening, sq ft
V = velocity of the air, *feet per minute*

From Eq. (10) dividing both sides by the quantity $k(V)$, we get another useful equation.

$$A = \frac{Q}{kV} \tag{11}$$

It is Eq. (11) in which we are really most interested. We have developed a method for getting the air quantity in cubic feet per minute, which is Q in Eq. (11). k is a quantity that we shall obtain from the grille manufacturer in the case of grilles or registers, and which we know to be equal to 1.00 in the case of ducts. [In the case of ducts, therefore, k may be simply dropped out and forgotten in Eqs. (10) and (11).] We shall also soon learn that the air velocity, which is V in Eq. (11), must be limited according to different design standards, which we are about to present. This means that we shall use Eq. (11) to obtain the necessary duct area or grille size that must be selected for satisfactory performance. We shall calculate Q and shall obtain V from data in a table to be presented shortly. Then we simply select a proper value for k, and we *know the entire right-hand side of Eq.* (11). We substitute these know numbers and solve for A which gives us the *grille area* or *duct area* in *square feet.*

Table 2 presents standards regarding the air velocities which may be permitted through ducts, grilles, and registers. In the case of ordinary ducts, the first three items in Table 2, a wide variation in velocities is given. The lower velocities are those which produce a noise level so low as to be scarcely audible. The higher velocities are such that a definite air noise will be audible. This air noise will not be objectionable provided the register and grille velocities are lowered as indicated by Table 2. Note that there are three cases in Table 2 where only *maximum* permitted velocities are stated. In these cases the reason for setting these maximums is not so much noise as the necessity of preventing the warm-air blast from the register from striking persons near the registers. (As we have stated once before in this chapter, the details regarding the design of duct systems and grilles, including connection of grilles to duct systems, is a subject of itself and will be covered in a future chapter.)

Selection of Conditioner Size. The winter air-conditioning unit selected must be properly sized with regard to several considerations. (1) It must have the *desired heat output* or capacity to supply heat at a sufficient rate properly to heat a building.

TABLE 2.—DUCT, GRILLE, AND REGISTER VELOCITIES

Type of Duct, Grille, or Register	Velocity, fpm
Main trunk duct	500–1,000
Branch duct	450– 750
Risers (wall stocks)	350– 600
Supply grilles:	
Low, plain outlet*	300 maximum
Low, deflecting toward the floor*	500 maximum
Low, deflecting and diffusing*	800 maximum
High, side wall†	600 minimum
Return registers	300–500

* Less than 2 ft above floor.
† Not less than 5 ft above floor.

(2) The conditioner must be *physically large enough* to handle the desired air quantity. In some cases this means that a conditioner must be purchased which is oversized in heat-output capacity, in order that it may be large enough to handle the necessary air quantity. This could occur only when low register temperatures are used, resulting in a large quantity of air circulated. (3) The *fan* within the conditioner must be able to deliver and move the desired quantity of air against the resistance that will be caused by the duct work and grilles. We shall not be concerned with the fan capacity because we have not yet learned how to calculate the resistance of the duct work.

The second factor mentioned in the preceding paragraph, regarding the physical capacity of the conditioner to handle the desired air volume, rarely influences conditioner selection because higher register temperatures are usually used to keep the cost of the duct work reasonably low. Hence we must concern ourselves at this time with the selection of the conditioner *with regard to its capacity in heat output.* Selection in this respect is much simpler than in the case of the selection of steam or hot-water heating boilers. In the first place, we do not have to contend with any appreciable pickup or warm-up allowance. In normal operation, the duct work probably cools from approximately register temperature to a temperature only 30 or 40° lower. The quantity of air handled is so large and the total weight of the duct work so relatively small that pickup is not a factor in *continuously heated buildings.*[1]

In selecting steam and hot-water boilers an allowance is added for loss because of uninsulated piping. In the case of warm-air heating, there is a small loss due to uninsulated duct work, mainly that duct work which is located along the basement ceiling. We ordinarily use the following simple equation in making this allowance:

$$R = \frac{H_t}{E} \tag{12}$$

where R = the rated conditioner output of heat, Btu per hr

H_t = the total heat loss of the building, *i.e.*, the sum of the H values for the individual rooms as used in Eqs. (6) and (8).

$$E = \frac{\text{heat delivered to rooms}}{\text{heat delivered to rooms} + \text{duct heat losses}}$$

Of course it would be possible to calculate the duct heat losses, thus determining a value for E by calculation. For continuously heated buildings however, it is common practice to use standard estimated values for E. These arbitrary values for E are based upon experience and upon numerous calculations of duct losses. They may be used for all ordinary systems, and E need only be investigated in more detail for systems having unusually extensive and lengthy duct systems.

For gas-fired conditioners, the American Gas Association recommends using 0.90 for E for residential work. In this case, the official A.G.A. output rating for the conditioner must be used for the value R. In the case of oil or stoker-fired conditioners, the manufacturers' output ratings may be used for R and 0.85 to 0.90 used for E. If the duct system is simple and has few hot wall stacks (supply risers) in cold outside walls, use 0.90 for E. If the duct system is more extensive and has some supply risers in cold outside walls, such exposed supply risers should be covered with at least a thin layer of insulation and 0.85 used for E.

Winter Air Conditioning for Larger Buildings. In the case of larger buildings such as city office buildings, a duct system to *supply all heat* by means of warm air often would be too expensive and would occupy too much valuable space. In such buildings, obviously all the air could not

[1] In the case of intermittently heated buildings such as country houses which are only occupied week ends, it is necessary to provide for a warm-up allowance. This allowance is not for the heating of the duct work itself but is instead to allow for heating up the entire building. The building will comprise at least several tons of solid materials, all of which may be as cold as 35° or colder when the plant is started. These materials must be heated up to about 70° before the house can be comfortable. Warm-up allowances for intermittently heated buildings vary from 50 to 100 per cent, depending upon the building and cannot be calculated accurately.

be supplied for a warm-air conditioning system from a single unit in the basement, as this would mean supply risers larger than elevator shafts. Therefore steam-heated conditioners are used on various floors, each conditioner usually caring for a total of three floors, at most, its own floor and the floors directly above and below. Thus steam must be piped through the building to supply the steam coils in these conditioners.

Steam may be piped throughout the building at high pressure in small pipes and reduced to heating pressures of less than 10 psi at each floor. Therefore office buildings being built today, even though equipped with complete summer and winter air conditioning, have convectors or concealed radiators of a sufficient capacity to take care of the transmission-heat losses of the building. The duct system then provides clean conditioned air with a sufficient fresh outdoor air percentage to take care of ventilation. This outdoor air percentage is usually high enough to create a slight positive pressure within the building, thus minimizing infiltration. The winter humidification is of course provided by adding moisture to the air circulated by the duct system.

Conclusion. With winter air-conditioning units now on a mass production basis, prices have been greatly reduced and these units are for the most part well engineered. Winter air conditioning has thus been brought within the reach of the most inexpensive homes. Large real-estate developments offering complete homes are today frequently offering winter air conditioning as the *standard* means of heating. For those persons desiring heating by radiation, an extra charge is often made in real-estate developments of this type. Even though modern convectors and concealed radiators have become highly developed, their use in the better class of work is to *supplement* a *winter air-conditioning* system, resulting in installation of the split-system type. Large buildings, such as the air-conditioned parts of Rockefeller Center and the Metropolitan Life Insurance Company home-office buildings are examples of split-system installations. Because this chapter contains important material, it should be reviewed after you become familiar with the design of duct systems.

PROBLEMS

1. What is necessary in order to convert an obsolete gravity warm-air furnace into a modern winter air-conditioning unit? What are the advantages of the converted unit compared with the obsolete gravity unit?

2. What is the difference between a direct-fired winter air-conditioning installation and a split-system installation?

3. Warm air may be supplied through "high" supply outlets (5 to 7 ft. above the floor) or through low supply outlets (not over 2 ft above the floor). Compare these two methods as to advantages and disadvantages from the standpoint of the persons living in the air-conditioned house.

4. How is domestic hot water provided when a direct-fired winter air-conditioning unit is used to heat the building?

5. There is considerable difference in the type of installation to winter air-condition a large city building compared with that used to winter air-condition a residence. What are the three major points of difference?

6. What are the steps involved in the design of a winter air-conditioning system?

7. You have the following information regarding a small residence:

	Living room	Dining room	Kitchen	Bedroom *A*	Bedroom *B*	Bath-room
Heat load, Btu per hr.	16,000	12,000	10,000	16,000	13,000	9,000
Number of grilles needed for the proper distribution in rooms	2	2	1	2	1	1

The heat loads shown above include heat required for hallways, etc. Using a register temperature of 140°F, determine the air quantity necessary per room.

8. Assume that we use one return air grille per room in the preceding problem, with no return grille in the bathroom or kitchen. Assume that the supply air is divided equally between the two grilles in those rooms which have two grilles per room. Based upon grilles having a free area coefficient of 80 per cent, determine the size grilles if low supply outlets of the deflecting and defusing type are used.

9. If fresh outdoor air is introduced having the *same weight* as the *weight of the warm air supplied to the kitchen*, determine the capacity of the conditioner required, in Btu per hour. Assume the residence continuously heated, and use $E = 0.90$ in formula (12).

10. Assume that the residence described in Prob. 8 is to be heated with a split system, radiators heating the kitchen and bathroom. Assume that a radiator is added for the garage requiring 8000 Btu per hr. Assuming 10 *lb per min* of fresh outdoor air for ventilation, determine the following: (*a*) Capacity of steam heated conditioner. (Express your answer in Btu per hour and square feet E.D.R.) (*b*) The size of the kitchen, bathroom, and garage radiators, in square feet E.D.R. (*c*) The size of the boiler required, assuming an automatically fired steel boiler. (Refer to Chap. XVI, page 294, for the method of selecting the steam boiler. Consider the steam-heated conditioner the same in its effect upon the steam boiler as one large radiator.)

CHAPTER XVIII

USE OF PSYCHROMETRIC CHART

General. At the end of Chap. VII we presented some information regarding how psychrometric data could be shown on a graph. In this chapter, we shall continue this discussion and show *how* different types of psychrometric *charts are drawn* from tabular data. After learning how the charts have been compiled and drawn, we shall learn how to *locate* any sample of *air on a chart*, starting with a knowledge of any *two independent items* of psychrometric data about the sample of air. We shall then learn how to *find all* the *useful psychrometric properties* of air, starting from any sample of air located anywhere on the psychrometric chart. Finally, we shall illustrate on the psychrometric chart various heating, cooling, humidifying, and dehumidifying processes, including mixtures of two different amounts of air, each having different psychrometric properties.

Before proceeding, it is essential that you understand the theory of psychrometry and the use of the Psychrometric Tables. If you do not have a thorough knowledge of all the material in Chap. VII, you should review it before proceeding further. As a test problem, take air at 90° dry bulb, total heat, sensible heat, latent heat, specific volume, and vapor pressure. You must be able to solve this problem easily, and with certainty as to your method, before studying this chapter.

Construction of Psychrometric Charts. There are probably two dozen different psychrometric charts in general use, although there are only about a half a dozen that are widely used. Most charts fall into one of two general classes, depending upon their method of construction, although some charts are a combination of the two general types. The chart of the Carrier Engineering Corporation is an example of the latter combined type. One general class of psychrometric charts is drawn in a manner such that the relative-humidity lines are pronounced *curved lines*. Charts of the other general class are drawn in such a way that the relative humidity lines are *straight lines*.

It is easier to understand the *method of construction* resulting in *curved* relative-humidity lines. Let us, therefore, undertake to construct a simplified chart of this nature, after which we shall add other information to make it more nearly like a complete chart. From the Psychrometric Tables,[1] look up values for the moisture content of saturated air, expressed

[1] See Tables IV and V in the Appendix.

in grains of water per pound of dry air, at saturation temperatures in steps of 10° from 0° to 100°F. Set up this information as the first two columns of a table. Divide each moisture content in the second column by 2, and tabulate this "50 per cent moisture content" values as column (3) of your table. Your table should look like Table 1.

TABLE 1.—ABSOLUTE HUMIDITY, GRAINS PER LB

Dry-bulb temp (1)	To saturate fully (2)	To saturate 50% (3)
0	5.5	2.75
10	9.2	4.6
20	15.0	7.5
30	24.0	12.0
40	36.4	18.2
50	53.4	26.7
60	77.2	38.6
65	92.4	46.2
70	110.2	55.1
75	131.1	65.6
80	155.5	77.8
90	217.0	108.5
100	300.0	150.0

Now take a sheet of 8½- by 11-in. graph paper or ordinary paper ruled in squares about ⅜ in. apart. Along the bottom horizontal axis of the graph locate dry-bulb temperatures from 0 to 100°F. Along the left-hand vertical axis plot grains of moisture per pound of dry air from 0 to 160. You are now ready to plot the first curve of your graph. Using columns (1) and (2) from Table 1, plot a curve and label it "100% saturated." Make sure that your curve is fairly smooth through the points that you have plotted.

Now use columns (1) and (3) from Table 1 and plot a second curve. Label this curve "50% saturated." You now have a simple psychrometric chart which should closely resemble Fig. 1. From this chart you can read the moisture content of air at any dry-bulb temperature from 0 to 100°, at either 50 or 100 per cent relative humidity. Starting from the left-hand scale you can determine the required dry-bulb temperature for air having any given moisture content, provided its relative humidity is to be 100 or 50 per cent.

The construction of the psychrometric chart shown in Fig. 1 should have constituted a review for you of material in Chap. VII. Let us now proceed to make additions to the psychrometric chart, shown in Fig. 1, so that we may have a skeleton of an *actual complete* chart. Take another sheet of graph paper, and draw upon it the same scales used to construct Fig. 1. About ¾ in. to the right of the 100° dry-bulb vertical line, draw another vertical line which we shall soon use for a scale. Draw the two curves the same as they were drawn in Fig. 1.

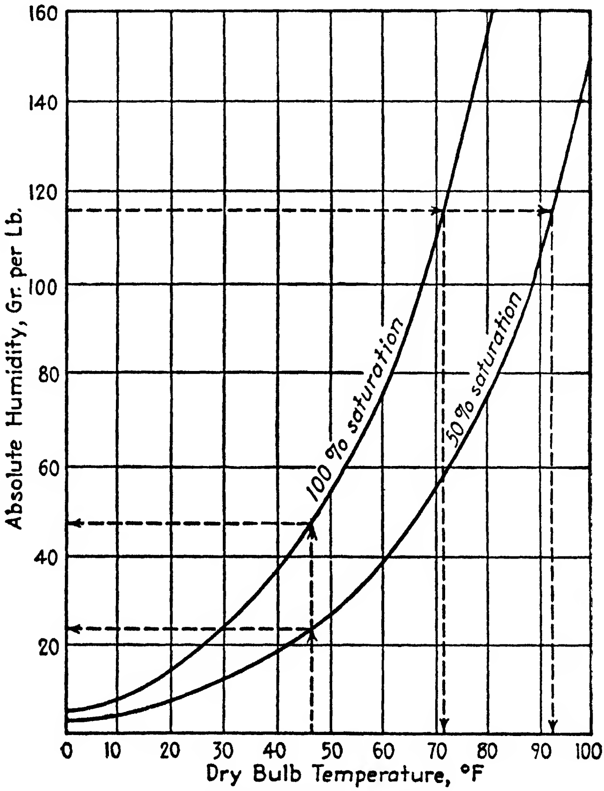


FIG. 1.—Simple psychrometric chart showing 50 per cent and 100 per cent saturation lines.

Mark each point where a dry-bulb temperature line intersects (crosses) the 100 per cent saturation curve. From each of these marked points, draw a horizontal line to the right as far as the 100° dry-bulb line. Consider now the horizontal line you have just drawn to the right from the *intersection of the* 50° *dry-bulb line* and *the saturation curve.* Label the *right-hand end* of this line "50°." In a similar manner, label the right-hand end of all the horizontal lines you have just drawn. This will result in a vertical scale along the right-hand side of your chart. Mark this scale "dew point." The chart you are drawing, although not yet complete, should resemble Fig. 2.

Either on your chart or on Fig. 2, mark two points about 1½ in. apart anywhere along the 50° dew-point line. Determine approximately the absolute humidity for air at these two different conditions. You will find that, upon reading horizontally to the left, the absolute humidity for the two points *is identical.* Since the two points are at two different dry-bulb temperatures but at the very *same dew point,* you know from your previous

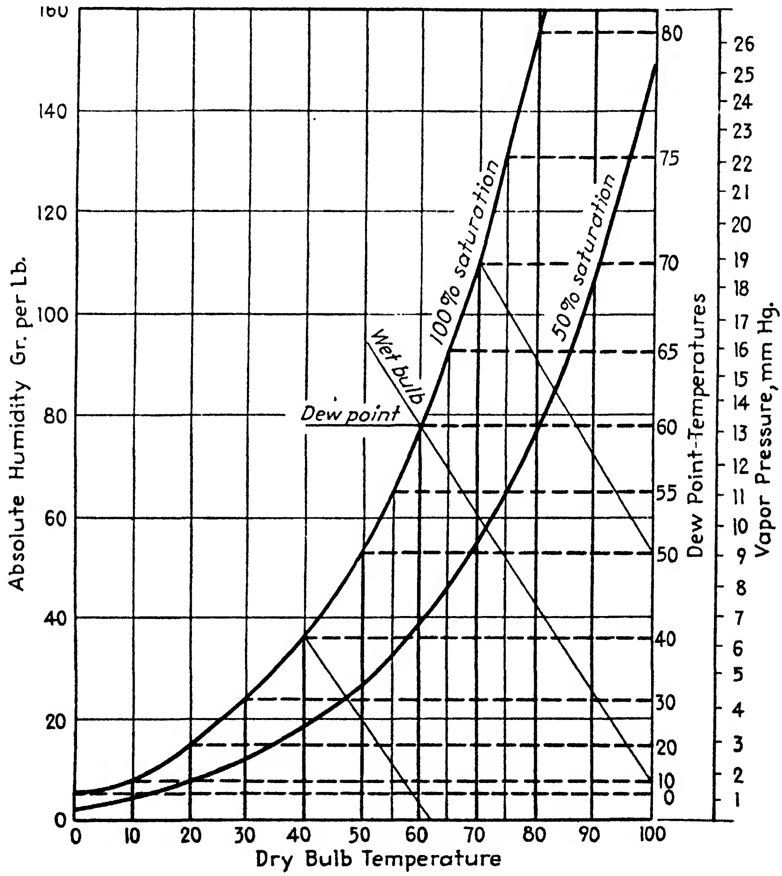


FIG. 2.—Simple psychrometric chart with wet-bulb and dew-point lines added.

knowledge of psychrometry that the absolute humidity must be the same. Therefore, our horizontal dew-point lines have been drawn correctly. You also know that for every dew point and its corresponding absolute humidity one and only one corresponding water-vapor pressure exists and this vapor pressure can be determined from column (2) of the Psychrometric Tables. Look up these vapor pressures in inches of mercury for dew-point temperatures of 0, 10, 20°, and so on up to 80°. From the conversion information on page 110, express these vapor pressures in *millimeters of mercury* instead of inches of mercury. Plot these vapor pressures *very* *lightly* in pencil on the vertical scale ¾ in. off to the right of your chart. Your vapor pressures will not be in even millimeters of mercury—1.00, 2.00, 3.00, etc. For an actual chart it is necessary to determine from the psychrometric tables the dew points corresponding to different *vapor pressures in even steps.* These vapor pressures in even steps are then plotted on the vertical "ladder scale" to the right of your chart. This has been done and is illustrated in Fig. 2.

To complete Fig. 2, we now need to draw some wet-bulb temperature lines. We know that any wet-bulb temperature line, say the 50° wet-bulb line, starts from the saturation curve at the dry-bulb line of the same temperature. In other words, the 50° wet-bulb line meets the saturation curve at the same point where the 50° dry-bulb line intersects the saturation curve. We need to know the point at which these wet-bulb lines intersect our 50 per cent relative humidity in order to draw the lines. This is one of the most difficult steps in drawing a psychrometric chart as these points can be found only by trial and error in ordinary psychrometric table practice. These points can be determined, however, and wet-bulb lines drawn as shown in Fig. 2.

The wet-bulb temperature lines can be extended to the left of the saturation curve in Fig. 2, and a total heat scale can be plotted along the ends of the wet-bulb lines. This method of showing total heat is used in the Peerless Psychrometric Chart, shown on page 340, and is also used on the copyrighted psychrometric chart of the General Electric Company.

Now we shall examine the construction of psychrometric charts having straight lines instead of curves for the relative-humidity lines. The mathematical nature of the relative-humidity curves on the type of chart we have already discussed is such that, by the use of a special type of graph paper having unequally spaced lines, these relative-humidity curves from the chart we have just drawn turn out to be perfectly straight lines when the points are plotted. These unequally spaced lines are scaled off in a manner similar to the divisions on the scale of a slide rule. Turn now to Fig. 3. Since you do not have the logarithmic type of graph paper, we suggest that you plot scales on a sheet of plain white paper, making your scale spacings similar to those shown on the vertical and horizontal axes on Fig. 3. You can then construct for yourself a chart similar to Fig. 3 and really understand how this is done.

In this chart, vapor pressure in millimeters of mercury is plotted on the left-hand vertical axis. You must make a table similar to Table 1, showing the vapor pressure in *millimeters of mercury* corresponding to saturation temperatures of 0, 10, 20, etc., to 100°F. Plotting these vapor pressures will result in the *straight-line* 100 per cent saturation curve, as shown in Fig. 3. The dew-point lines may next be drawn from the intersection points of the dry-bulb temperature lines on the saturation curve. The equation giving the most exact definition of relative humidity is

$$\frac{\%\ \text{R.H.}}{100} = \frac{P_{va}}{P_{vs}} \tag{1}$$

where P_{va} = actual vapor pressures of the water vapor

P_{vs} = vapor pressure of the water vapor, for a saturated vapor at the dry-bulb temperature of the air

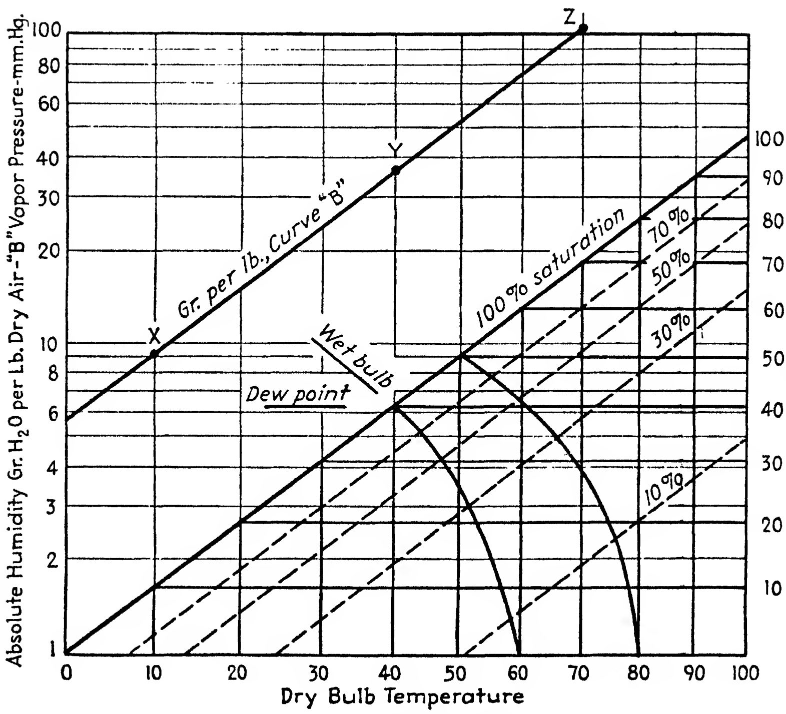


FIG. 3—Psychrometric chart based on points plotted on log-log graph paper. Compare with Fig. 1.

Transposing the above equation, we have

$$P_{va} = P_{vs}\left(\frac{\%\ \text{R.H.}}{100}\right) \tag{2}$$

By means of Eq. (2) we can determine the vapor pressure at any dry-bulb temperature and at any relative humidity desired. By this means, other relative-humidity lines can be plotted on the chart in Fig. 3.

To draw wet-bulb temperature lines on this type of chart requires locating at least four or five points *representing the same wet-bulb temperature.*

When this is done, the wet-bulb temperature lines are found to be curved lines, as shown in Fig. 3.

The relation between the two types of charts that we have constructed may appear clearer if you place Fig. 2 before you and imagine it to be made of extremely elastic rubber. Imagine that you can stretch this rubber psychrometric chart out of shape in such a way as to straighten out all the relative-humidity lines. You can see that, provided the dew-point lines are kept horizontal straight lines, the wet-bulb temperature lines *must bend.* The most prominent psychrometric chart of the type shown in Fig. 3 is the Bulkeley chart, which was presented to the A.S.H.V.E. in 1926. This chart is copyrighted by the A.S.H.V.E., and individual copies may be obtained from the society for a nominal charge. A Bulkeley chart may be found on the inside back cover.

On charts of this type, the left-hand scale, which we have already drawn for vapor pressure, is ordinarily used in addition for other purposes. Place your pencil at 50° dry bulb, 50° dew point, 100 per cent saturated on Fig. 3. You can readily see that if you read to the left you do *not* read absolute humidity in grains per pound as was the case in Fig. 2. It is true that we can erect a ladder scale to the right of the chart showing grains per pound. It happens that the space to the right of the chart is desired for another purpose on the Bulkeley chart and on some others of this type. Also there is a convenient and neat method of using the scale we already have at the left as a means for showing grains per pound. From the Psychrometric Tables or from Fig. 2, determine the absolute humidity at three saturation temperatures such as 10, 40, and 70°. For these three points, the absolute humidities are 9.2, 36.4, and 110.2 grains per lb. Consider the data in the preceding two sentences as data for a graph; from it plot three points such as X, Y, and Z in Fig. 3. Point X is plotted *vertically directly above the corresponding dew-point temperature* on the saturation line and *horizontally opposite* its *corresponding absolute humidity in grains per pound.* Points Y and Z are similarly located. Our curve through points X, Y, and Z is practically a straight line and corresponds to curve B on the Bulkeley psychrometric chart. By means of curve B, absolute humidity may be determined as illustrated in Fig. 11*b*. Two other curves, similar to curve B, may be drawn above and to the left of the saturation line for use in reading *total* heat and moisture content in *grains per cubic foot,* the values being read on the same numerical scale at the left of the chart.

How to Locate a Sample of Air on the Chart. In using psychrometric tables, we have seen that it is easily possible to determine all desired properties of the air, provided we know two *independent* properties. If we know total heat and wet bulb, we cannot find the other properties, because total heat and wet-bulb temperature are properties that vary one with the other—they are *dependent,* not *independent.* Similarly, if we know two independent properties of air, we can locate the sample of air *on the psychrometric chart* and mark the point representing the particular air we have. Let us start with the two properties of air that we most frequently measure, *viz.*, dry-bulb temperature and wet-bulb temperature. Figure 4 illustrates how to locate air on both of the types of charts which we have discussed, provided the dry-bulb and wet-bulb temperatures are known. All the following figures illustrate both types of charts. Sketch *a* illustrates the type of chart we developed in Fig. 2, having curved relative-humidity lines. Sketch *b* in each case will represent the type of chart having straight relative-humidity lines.

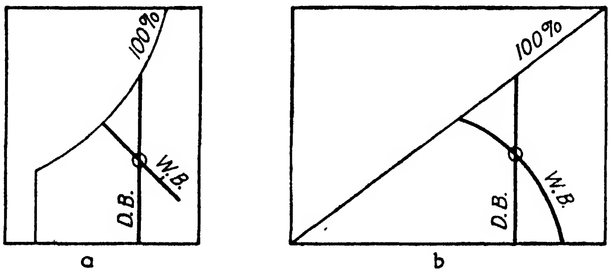


FIG. 4.—Dry-bulb and wet-bulb relationships shown for two types of psychrometric charts.

The dry-bulb temperature line may be located, on either chart, along the *bottom axis* or *up on* the *saturation curve.* The dry-bulb temperature line illustrated in Fig. 4 is definitely located in this way. The wet-bulb temperature line corresponding to the wet-bulb temperature of the air being located is found either by looking up the intersection of this wet-bulb line with the saturation line, or simply by means of the identifying numbers located adjacent to the various wet-bulb temperature lines. The dry-bulb and wet-bulb lines which you locate will intersect at *one and only one point,* regardless of the type of chart. This point locates your air. Figure 5 illustrates how to locate a sample of air on the chart, when the dry-bulb temperature and the relative humidity are known. The dry-bulb temperature line is located in the same manner as in Fig. 4. The proper relative-humidity line is easily identified in either chart. If the actual relative humidity of the air is 43 per cent, interpolate by eye between the 40 per cent relative-humidity line and the 50 per cent relative-humidity line in the case of most charts. With the Bulkeley chart, relative-humidity lines are shown for 42, 44, 46%, etc., so that you simply have to interpolate between the 42 and 44 per cent line.

Figure 6 illustrates how to locate a sample of air on the chart when the wet-bulb temperature and relative humidity are known. We rarely measure relative humidity and wet-bulb temperature as our two measured and known properties. It is sometimes necessary to determine the other properties of air when only these two are known, so we must be able to work starting from these two properties. In Fig. 6 note that on either

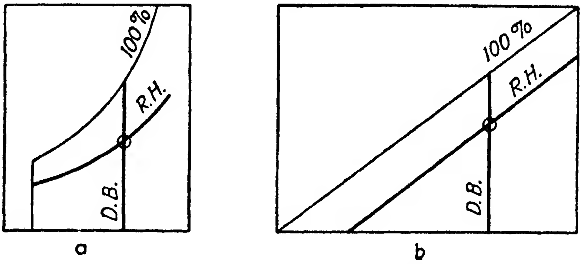


FIG. 5.—Dry-bulb temperature and per cent relative humidity relationships for two types of psychrometric charts.

chart the wet-bulb temperature line and the relative-humidity line intersect at one and only one point. This point locates the air, with the wet-bulb line and relative-humidity lines determined as described in the preceding paragraphs.

Figure 7 illustrates how to find the location on the chart of a sample of air when the dry-bulb temperature and dew point are known. In the construction of our first chart in Fig. 1 and 2, we saw that the dry-bulb

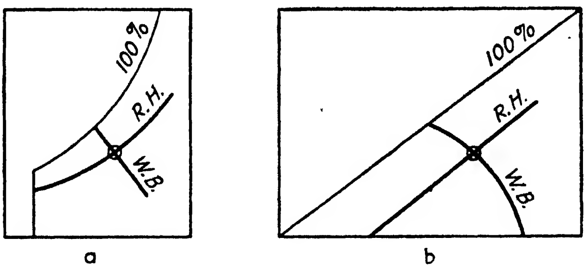


FIG. 6.—Method showing how to locate a sample of air on two types of psychrometric charts when wet-bulb temperature and relative humidity are known.

temperature lines are vertical while the dew-point lines are horizontal. Hence it is obvious that these two types of lines are *perpendicular* to one another. Take any dry-bulb temperature between 30 and 100°F, and take a dew-point temperature at least several degrees *lower* than the dry-bulb temperature you have picked. Following the method illustrated in Fig. 7, locate the dry-bulb and dew-point lines that correspond to the temperatures you have chosen, and you will see that they intersect. Again, this point of intersection locates the air definitely.

Figures 8 and 9 illustrate how to locate your air on a chart under two more sets of conditions. In Fig. 8 the intersection point of a relative-

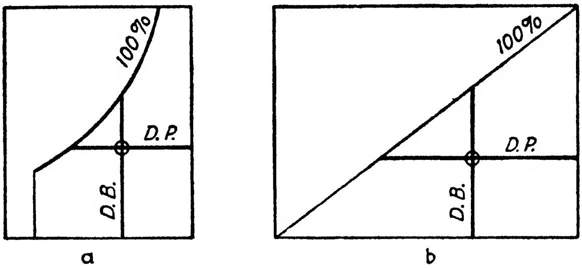


FIG. 7.—Method showing how to locate a sample of air on two types of charts when the dry-bulb and dew-point temperatures are known.

humidity line and a dew-point line is shown. When the relative humidity and dew point of a sample of air are known, you simply locate the proper

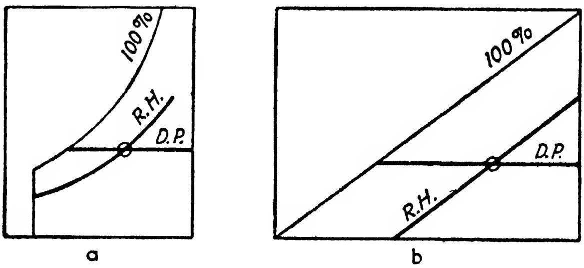


FIG. 8.—Method of locating a sample of air when dew point and relative humidity are known.

dew-point and relative-humidity lines and follow them to the point where they cross. This point locates the air on the chart. Figure 9 illustrates how

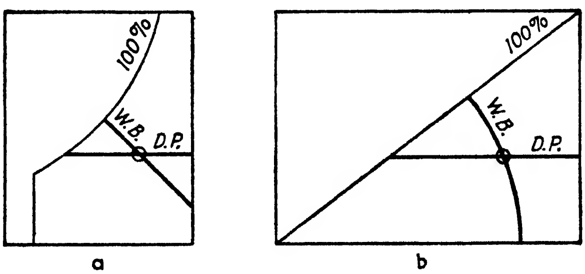


FIG. 9.—Method of locating a sample of air when wet-bulb temperature and relative humidity are known.

you would find air when the dew point and the wet-bulb temperature are known. You are now thoroughly familiar with finding dew-point and wet-bulb lines: simply find the proper lines corresponding to your known data, and their point of intersection locates the air.

Practice in Locating the Air on Charts. From the foregoing section, if you have followed the discussion with actual psychrometric charts before you, you have seen that any sample of air, once located on the chart lines, is the intersection of a total of four lines. Lines representing dry-bulb temperature, wet-bulb temperature, dew-point temperature, and relative humidity may all be drawn through any point plotted on a psychrometric chart. What is to be done if the point you have on a psychrometric chart

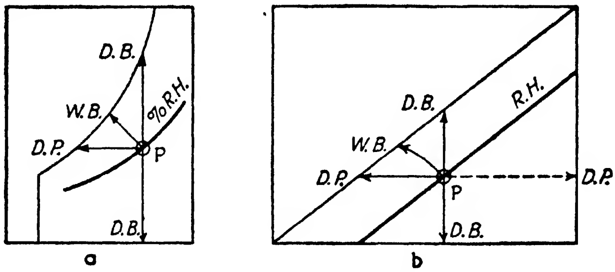


FIG. 10.—Read proper values in direction of arrows.

lies in clear white space with none of the chart lines passing through it? Look at the skeleton psychrometric charts shown in Fig. 10. This illustrates the direction to take with your eye to read dry bulb, wet bulb, dew point, and relative humidity from any point on the chart.

Interpolation between the various lines, if your point does not lie exactly on any line, consists of estimating the location of your point between the two lines giving the values of the psychrometric property that you are trying to read. On actual psychrometric charts, relative-humidity lines are the only ones with an interval of 10 points between lines. The chart lines representing other properties are spaced at smaller intervals, spacing between lines being from one to not exceeding five points. Observe the following general procedure when you are forced to interpolate by eye between lines on any type of chart or graph.

If the spacing between two marked lines represents five points—5°, 5 per cent or 5 of whatever the scale may be—you know that a point exactly *halfway* between the two marked lines is at a distance of 2.5 *points* from each of the marked lines. Let us say that the marked lines represent 40 and 45; the mid-point would represent 42.5; then 43 would be represented by a point *slightly above* the mid-point, while 42 would be represented by a point slightly *below* the mid-point. If we were required to locate 41, first locate 42 mentally, and then 41 lies exactly *halfway* between 40 and 42. Similarly, 44 lies exactly *halfway* between 43 and 45.

If you have to interpolate between two scale divisions or between two lines on a chart where the space is 10 *points* or 10 *divisions* between the two known markings, first mentally divide this space into *four* equal parts. Of course, until you become expert through practice you may make light pencil marks on charts instead of trying to create a *mental picture* of the divided space. Let us estimate with our two marked points 10 spaces apart, saying that the points represent 40 and 50. Divide the space into four equal parts, and you then have points at 40, 42.5, 45, 47.5, and 50. Now treat the space between 40 and your mid-point of 45, the same as any *five-space intervals*, as discussed above. Similarly, the space between 45 and 50 constitutes another simple five-point interval.

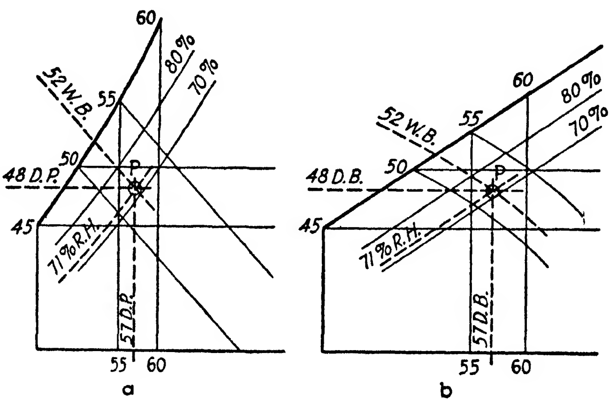


FIG. 11.—Method showing how to interpolate between major values when reading psychrometric charts.

Figure 11 shows a point on each of the two types of psychrometric charts, the point not lying directly on any chart line. Note that the charts sketched in Fig. 11 are not drawn to a true correct scale; in other words, they would not correspond to a photograph of an actual section of a real chart. Also the spacing between lines on Fig. 11 is greater than the spaces between lines on an actual chart. Figure 11 shows dry bulb, wet bulb, and dew point in steps of 5° each, whereas these properties are ordinarily shown in steps of 1 or 2° each. If real psychrometric charts were constructed with as widely spaced lines as our sketches in Fig. 11, interpolation on real psychrometric charts would be more difficult than it actually is. Therefore, if you can clearly understand the interpolation indicated in Fig. 11, you should have no trouble interpolating on a real psychrometric chart. Point P represents our point on both sketches in Fig. 11. The solid lines in Fig. 11 represent actual lines shown on a psychrometric chart. The dotted lines are guidelines which you would use in interpolating. At first draw these guidelines lightly in pencil, but after some practice you will merely lay a ruler along without actually drawing the line, or simply trace the line with your pencil point without actually marking the chart. In both sketches in Fig. 11, point *P* represents the same psychrometric conditions of air. Drop from point *P* vertically to the bottom axis (or you may go vertically upward to the saturation line) and you will see that you are between 55° dry bulb and 60° dry bulb. If you estimate carefully you will see that you are two-fifths of the way between 55 and 60°. Hence point *P* represents a dry-bulb temperature of 57°. As to dew point, extend a horizontal line to the left from point *P* to the saturation curve, as indicated by the dotted lines in Fig. 11. You will see that your dotted line thus drawn is three-fifth of the way between 45° dew point and 50° dew point. Therefore, point *P* represents a dew point of 48°.

To locate wet bulb on sketch *a*, draw a dotted wet-bulb line through point *P* parallel to the solid wet-bulb lines representing 50 and 55° wet bulb. On sketch *b*, the Bulkeley type chart, your dotted wet-bulb line through point *P* must be a curve in between and "parallel" to the wet-bulb curves for 50° wet bulb and 55° wet bulb. It is then clear that point *P* lies two-fifths of the way between 50° wet bulb and 55° wet bulb. Therefore, point *P* represents a wet-bulb temperature of 52°.

Finally, we must determine the relative humidity at point *P*. Point *P* lies between 70 per cent relative humidity and 80 per cent relative humidity and is, in fact, *very slightly* more than 70 per cent relative humidity. Perhaps you have become sufficiently practiced to merely look at point *P* and realize that it represents about 71 per cent relative humidity. If not, draw the dotted relative-humidity curve in sketch *a* and the dotted relative-humidity line in sketch *b* and you will see that these lines lie about one-tenth of the way from the 70 per cent relative-humidity line to the 80 per cent relative-humidity line. Thus it is clear that *P* does represent 71 per cent relative humidity.

Now for practice locate a point *P*, representing 57° dry bulb and 52° wet bulb, on the Bulkeley, Peerless, and Trane psychrometric charts. You will find that on all three charts this point will correspond to a dew point of 48°. The relative humidity appears to be 72 per cent on the Peerless and Bulkeley charts, while it seems more nearly 71 per cent on the Trane chart. Do not be disturbed by this apparent slight difference; remember the difference between the vapor pressures of water as shown in the Steam Tables and in the Psychrometric Tables. The difference is only in relative humidity and is so small that it is negligible.

Determination of Other Psychrometric Properties. *Absolute Humidity—Grains per Pound.* We now want to learn how to determine moisture content in grains per pound, total heat in Btu per pound, and specific volume, from various psychrometric charts. Let us first consider finding grains per pound, as this property is one of the most important and because most charts have a vertical scale of some sort from which grains per pound may be read.

On charts such as the Peerless, Trane, and General Electric, the general method is the same for determining moisture content in grains per pound. On the Peerless chart, read horizontally from point P, as indicated in sketch a, Fig. 12, *to the right* and read grains per pound on the scale so marked. On the General Electric chart, read *to the left* horizontally from point P and read grains per pound on the scale so marked. The Trane chart does not show grains per pound printed on the chart, but this value can be determined with the aid of the Trane psychrometric ruler, which we shall discuss later in this chapter. On the Research Corporation chart,

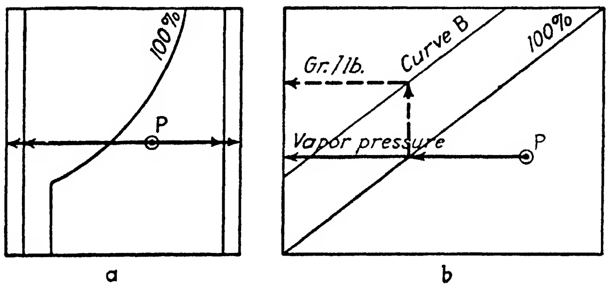


FIG. 12*a*.—Skeleton of Peerless psychrometric chart.
FIG. 12*b*.—Skeleton of General Electric chart.

read horizontally from point P and determine the dew point of the air. Then on the left hand of the two ladder scales in the upper left-hand portion of the chart, record this dew point; opposite this dew point on the ladder scale, the grains per pound value is shown.

Finally, we want to determine the moisture content in grains per pound from the Bulkeley chart. Refer to Fig. 3. Note the straight line $x = y = z$, also marked curve B, in the upper left-hand portion of the figure. If you recall how this curve was drawn, you know how to determine grains per pound from the Bulkeley chart. Figure 12b gives direction lines on a skeleton Bulkeley chart showing how to read grains per pound (and also vapor pressure). Read horizontally to the left from point P to the saturation curve, follow the vertical line straight upward till you strike curve B, then go horizontally to the left and read grains per pound on the scale at the left side of the chart. (If you wish to read vapor pressure, simply go straight horizontally from point P to the vapor-pressure scale at the left side of the chart, as indicated by the solid guideline in sketch b.)

Total Heat in Btu per Pound. Figure 13a shows the method for finding total heat on the Peerless or General Electric psychrometric chart. Simply extend the wet-bulb line representing the wet-bulb temperature of your air as a straight line beyond the saturation line on the chart. This extended wet-bulb line will intersect the total heat scale on either the Peerless or the General Electric chart. On the Research Corporation chart, the right hand of the two ladder scales shows the relation between wet-bulb temperature and total heat. Simply place your pencil on the wet-bulb temperature of your air on this ladder scale and read the total heat in Btu per pound.

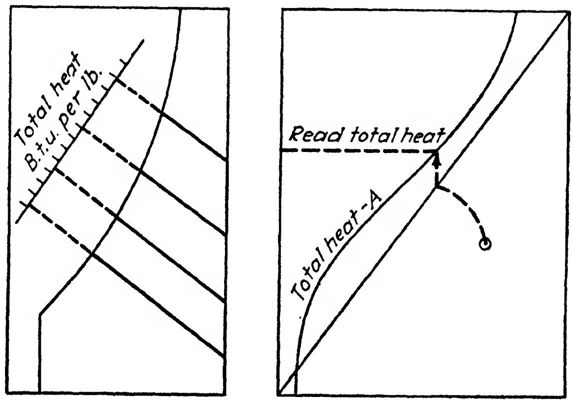


FIG. 13a.—Skeleton of Peerless chart for finding total heat.
FIG. 13b.—Skeleton of Bulkeley chart for finding total heat.

Figure 13*b* illustrates finding the total heat on the Bulkeley chart. You will note that there is a curve *A* above and to the left of the saturation line on the psychrometric chart. Unlike curve *B*, curve *A* is not a straight line. To find total heat on this chart, determine the wet-bulb temperature of your air. Follow this wet-bulb line up to the saturation curve, as indicated by the guide line in Figure 13*b*, then go vertically upward until you meet curve *A*, then horizontally to the left and read total heat on the scale at the left side of the chart. (This scale at the left side of the Bulkeley chart is very cleverly designed. Note that we have already found that we can determine vapor pressures, grains per pound, and now finally total heat on this *same* scale.)

Specific Volume in Cubic Feet per Pound. Again we find the difference in determining these psychrometric properties depending upon the chart that is used. We shall now consider the Bulkeley chart illustrated in Fig. 14*b*. Suppose that we take point *B* on the Bulkeley chart as 80° dry bulb, 50 per cent relative humidity, 60° dew point. The Bulkeley chart does not offer a method that is either easy or accurate for obtaining the specific volume of air that is at any other relative humidity than 100 per cent. Since most of the air we are interested in is not at 100 per cent relative humidity, the Bulkeley chart is a poor one for use in finding specific volume. Read to the left from point P horizontally to the saturation curve as indicated by the guidelines in Figure 14*b*, then read vertically upward until you meet curve E, which is one of the several curves above and to the left of the saturation curve. Then read horizontally *to the right*, reading a value on the scale "weight of 1 cu ft of air in pounds." By doing this as indicated by the solid guideline on Figure 14*b*, the Peerless

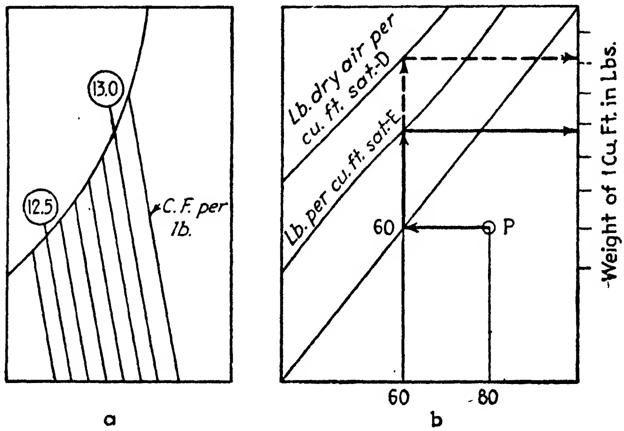


FIG. 14*a*.—Method for finding specific volume using Trane chart.
FIG. 14*b*.—Method for finding various values using Bulkeley chart.

chart will show the result to be 0.0757 lb per cu ft. This value is not our usual *specific volume* which we represent by the letter v; instead it is $1/v$. This result obtained as discussed above from air at 80° dry bulb, 50 per cent relative humidity *gives* $1/v$ *for air at* 60° *dry bulb* 100 *per cent saturated* but does not *give* $1/v$ *for our actual air at point* P. To get $1/v$ for our actual air at point P, we would have to apply our temperature-volume gas law from Chap. IV, which was

$$\frac{T_1}{V_1} = \frac{T_2}{V_2}$$

where $T_1 = 60° + 460°$
$T_2 = 80° + 460°$

It is not necessary that you understand the method of determining specific volume of partly saturated air from the Bulkeley chart as discussed above. In reading horizontally to the right from curve E, it is easy to make a mistake in reading your $1/v$ value. Inaccuracies are then quite possible, and the need for applying the temperature-volume law makes the method lengthy and difficult rather than easy and quick. (By following the dotted guidelines on Fig. 14*b* using Curve *D* instead of curve *E* on the Bulkeley chart, it is possible to read the weight of dry air per cubic foot of saturated air and vapor, not including the weight of the water vapor.)

Precautions Regarding Units. From the Psychrometric Tables, determine the number of pounds of dry air and the number of pounds of water vapor in 10,000 cu ft of moist air at 95° dry bulb and 75° wet bulb. Your answer should be as follows:

714.1 lb of dry air
7.9 lb of water vapor
722.0 lb of mixture

These values were obtained as the solution of a problem in Chap. VII, but we present them again to remind you that air as we ordinarily find it is a *mixture*, which is *by weight* mostly *dry air*, plus a little *water vapor*. On psychrometric charts we have seen that we can read the following properties:

Absolute humidity	Grains per *pound*
Total heat	Btu per *pound*
Specific volume	Cubic feet per *pound*

For each of these values, the psychrometric-chart data are based upon a quantity of *air plus water vapor* such that the *dry air alone* has a *weight of 1 lb.* In other words, when we read a property of air "per pound" from the chart, we read that property "per *pound of dry air, plus* whatever *vapor* it may be carrying." We may state this fact in still another way to make sure that it is clear.

Total heat per pound gives the total heat of a quantity of air and water-vapor mixture that would weigh 1 lb. if all the water vapor were removed; but the latent heat of the water vapor is of course included in the total-heat value given.

Absolute humidity in grains per pound gives the weight of water vapor that is contained in a quantity of air-vapor mixture that would weigh exactly 1 lb. if the water vapor were removed.

Similarly, specific volume in cubic feet per pound gives the volume in cubic feet of a quantity of air and water vapor mixture, the weight of which would be 1 lb. if the water vapor were removed.

The information we have presented in the preceding paragraphs should not be new to you if you understand thoroughly the principles of psychrometry from your study of the Psychrometric Tables. These tables also present properties based upon 1 *lb of dry air plus its vapor.* Let us take a numerical example to make this matter definitely clear. The chart or tables tell you that air having a dew point of 70° contains 110 grains of

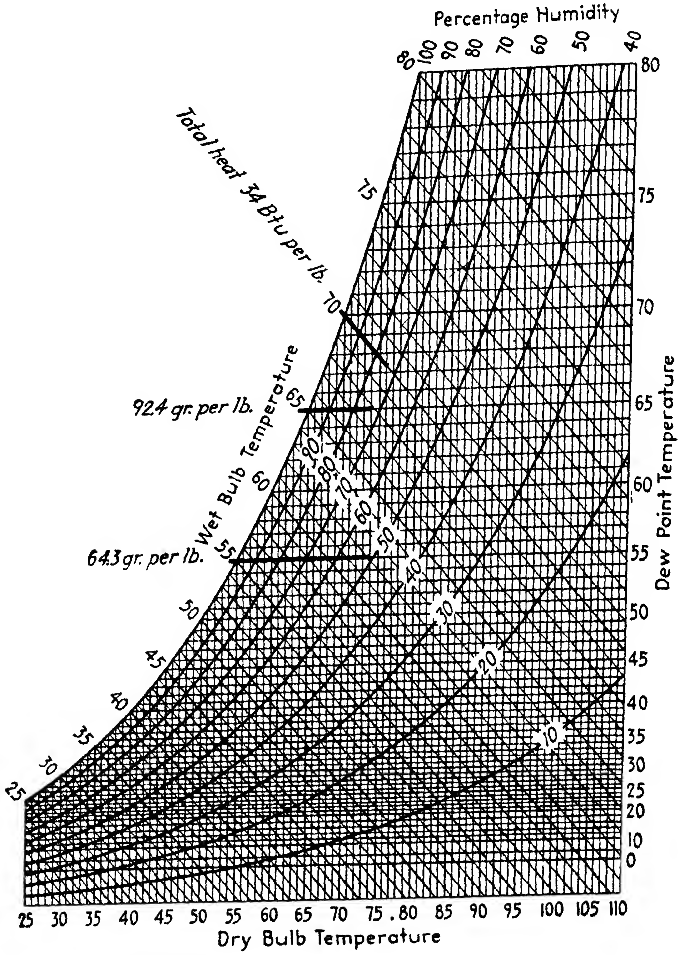


FIG. 15.—Trane psychrometric chart. (*The Trane Co.*)

water vapor per pound of dry air. Therefore you actually have

Dry air	1.000 lb	= 7,000 grains
Water vapor	0.0157 lb	= 110 grains
Total mixture	1.0157 lb	= 7,110 grains

Since the three properties of total heat, absolute humidity, and specific volume are *all* based upon 1 lb of dry air plus its vapor, no complications are introduced into our calculations. It is simply important to realize and understand throughout your work that the air *volumes* that we measure

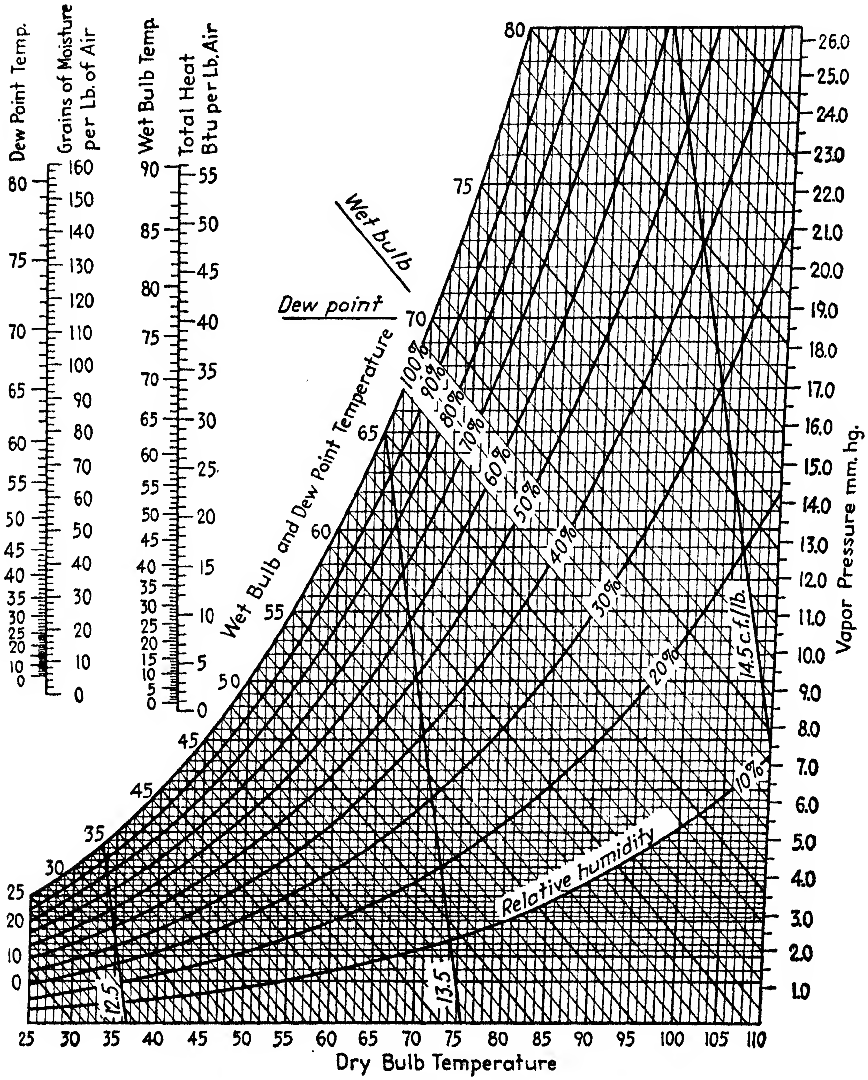


FIG. 16.—Research Corporation psychrometric chart. (*Research Corporation.*)

and talk of in cubic feet per minute are volumes of actual moist air, *i.e.*, the volume of dry air plus vapor. The *weights*, however, will include only the weight of dry air, the weight of the vapor being excluded from our calculations.

Charts Referred To in This Chapter. *Bulkeley Chart.* A Bulkeley psychrometric chart of the size that is used by the A.S.H.V.E. is attached to the back cover. This chart conforms to the general construction of

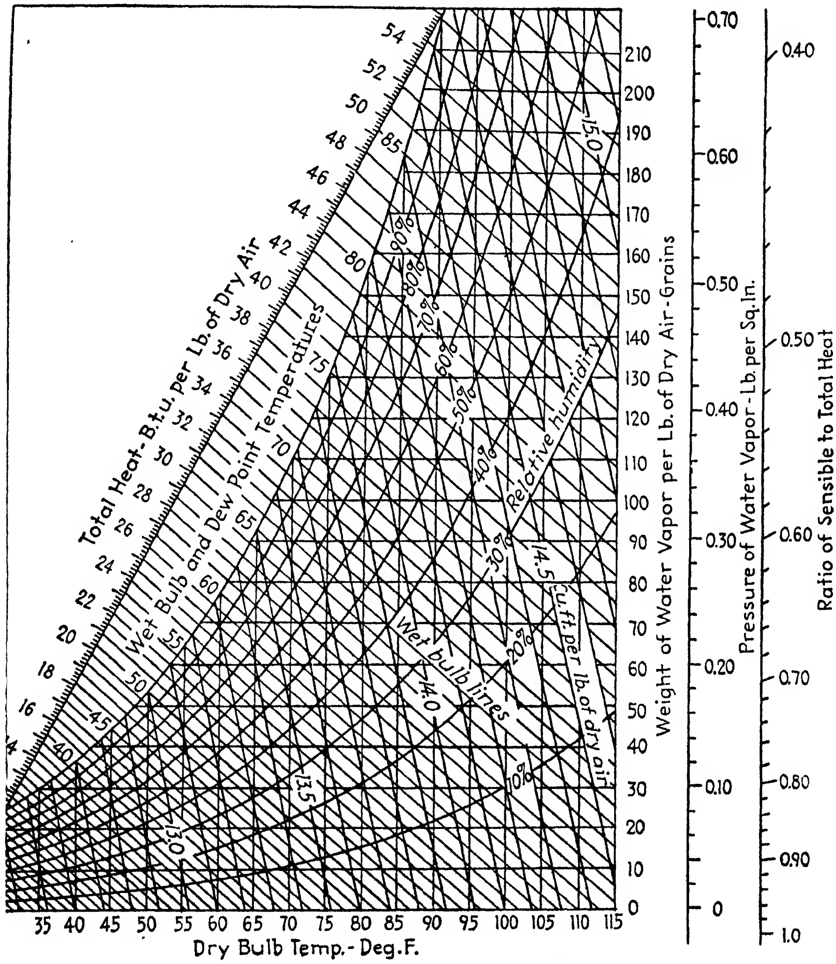


FIG. 17.—Peerless psychrometric chart. *(Peerless Co. of America.)*

the chart illustrated in Fig. 3, and it is explained by the sketches marked *b* in Figs. 4 to 14.

Trane Psychrometric Chart. The Trane psychometric chart is illustrated in Fig. 15.

Research Corporation Psychrometric Chart. The Research Corporation psychrometric chart is illustrated in Fig. 16. This chart is similar in general construction to those illustrated in Figs. 1 and 2 and in the sketches marked *a* in Figs. 4 to 14. The use of the ladder scales to the left in the

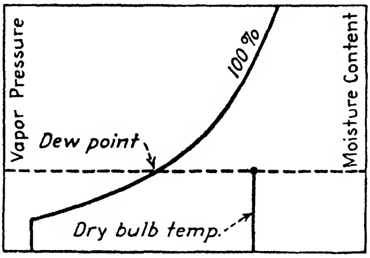


Fig. 18*a*.—Method for finding dew point, vapor pressure, and moisture content on General Electric chart Fig. 19.

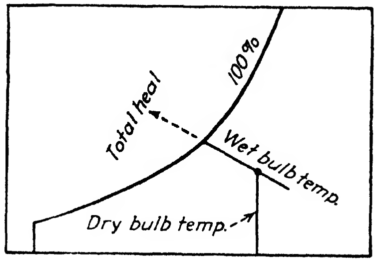


Fig. 18*b*.—How to find wet-bulb temperature and total heat using General Electric chart Fig. 19.

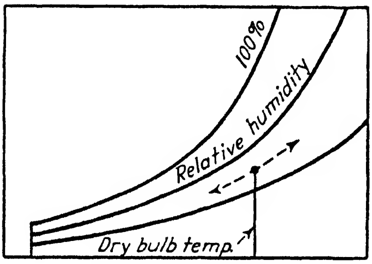


Fig. 18*c*.—How to read relative humidity on General Electric chart Fig. 19.

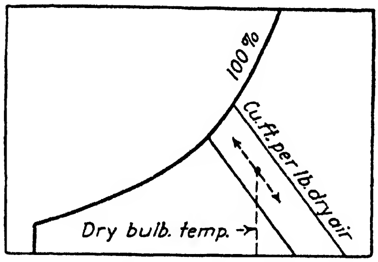


Fig 18*d*.—How to read specific volume from lines marked cu ft per lb of dry air, using General Electric chart in Fig. 19.

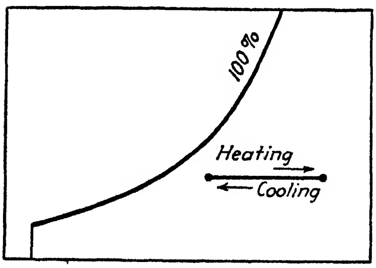


Fig. 18*e*.—Sensible heating and cooling of air as shown on General Electric chart in Fig. 19.

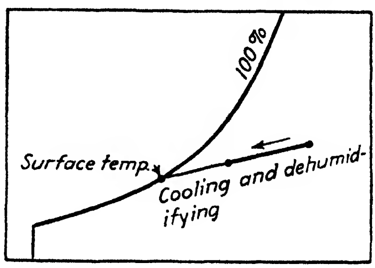


Fig. 18*f*.—Cooling and dehumidifying as shown on General Electric chart in Fig. 19.

Research chart is explained on page 334. Specific volume must be interpolated on the Research chart, using the general method illustrated in Fig. 14*a*.

Peerless Psychrometric Chart. The Peerless psychrometric chart was illustrated in Chap. VII and for your convenience is shown again as Fig. 17.

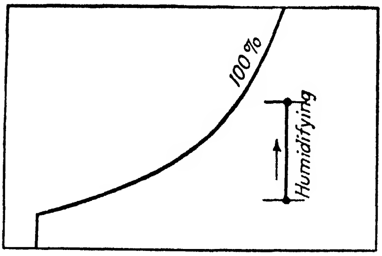


FIG. 18*g*.—Humidifying of air as shown on General Electric chart in Fig. 19.

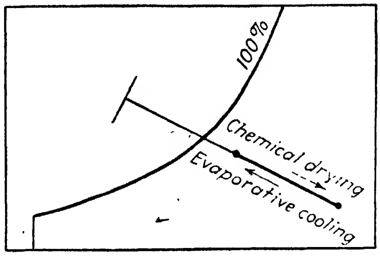


FIG. 18*h*.—Chemical drying and evaporative cooling as shown on General Electric chart in Fig. 19.

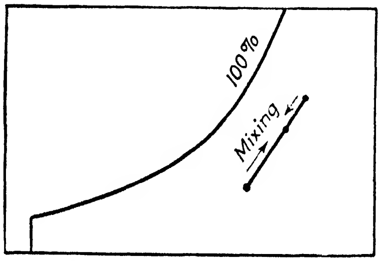


FIG. 18*i*.—Mixing of air of two initial conditions. Condition of resultant mixture will fall on this line at a point determined by the relative weights of air being mixed. Refer to Fig. 19.

General Electric Psychrometric Chart. The General Electric psychrometric chart and instructions for using it are shown in Figs. 18 and 19.

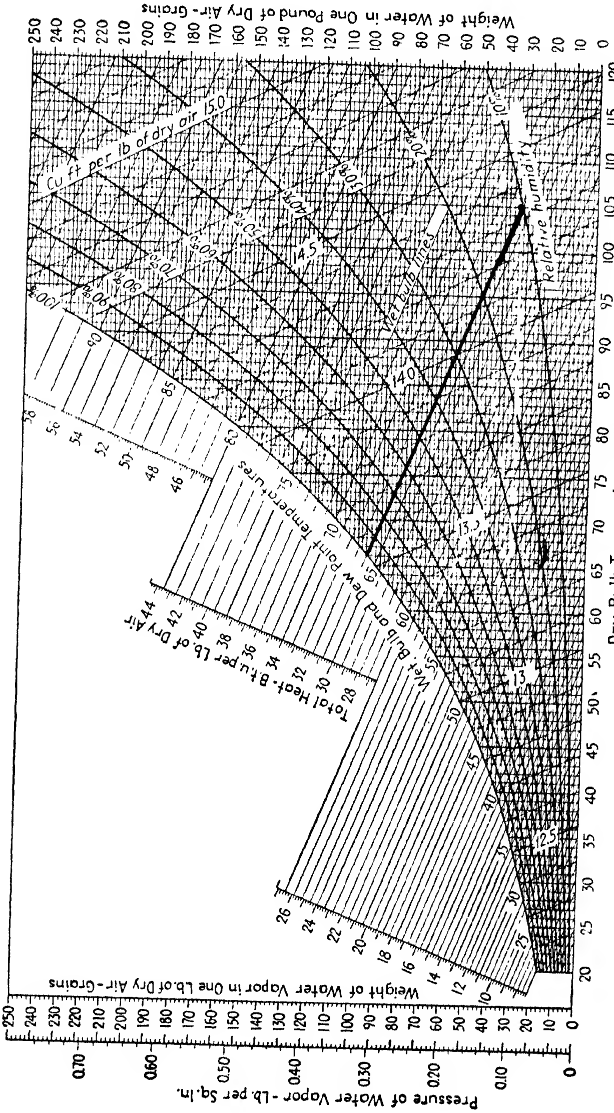


FIG. 19.—General Electric psychrometric chart. (*General Electric Company.*)

CHAPTER XIX

CALCULATION OF COOLING LOAD

General. In Chap. XIII we examined the interior conditions that affect human comfort and we became familiar with the A.S.H.V.E. comfort chart for summer and winter. In Chap. XIV we learned the calculation of transmission-heat losses through various types of building construction and also learned a method for calculating the amount of air leakage likely to occur with various types of building and window construction. In Chap. XV we developed a method of calculating winter heating load based upon the information in Chap. XIII and XIV.

A good deal of the work of computing summer cooling load is identical with the calculation of winter heating load except that the heat flows *into* instead of flowing *out of* the building. The direction of heat flow is reversed in summer because the indoor temperature is held (by the air-conditioning system) lower than the outdoor temperature. Although the method is basically the same for calculating summer heat load, there are minor differences and exceptions so that we cannot tell you to reverse the procedure learned in Chap. XV. Furthermore, there is one important new factor, which we have not as yet mentioned, that of "solar heat" (or radiant heat from the sun) which is of tremendous effect in determining summer cooling requirements. We shall, therefore, learn in this chapter an organized procedure for calculating summer cooling load. Since solar heat effect is new we shall discuss this item in detail. We shall also cover the other components that make up the total summer cooling load.

Ordinary Transmission Heat Load. Even on a cloudy summer day when there is no sun effect to contend with, we have heat flow by transmission into any building that is cooled. This ordinary transmission flow of heat occurs because of the temperature difference between indoors and outdoors. This transmission heat load is calculated in a manner similar to that described in Chap. XV except that the direction of heat flow is reversed. We may use a formula that is substantially the same as formula (2) in Chap. XIV, rewriting it slightly for our purpose here.

$$H_t = AU(t_2 - t_1) \qquad (1)$$

where H_t = heat transmitted through the material of wall, glass, floor, etc., *Btu per hr*

A = net inside area of wall, glass, floor, etc., *sq ft*

t_1 = inside temperature, °F
t_2 = outside temperature, °F
U = coefficient of transmission of wall, glass, floor, etc., Btu (hr) p (sq ft) (°F) difference in temperature (Tables XIX to XXIX)[1]

For the inside temperature t_1 we shall use the temperature decided on after a study of the summer comfort chart in Chap. XIII. As stated in Chap. XIII, 80°F is the usual dry-bulb temperature used as a basis for summer air-conditioning design in New York City when the outdoor-design dry-bulb temperature is at its maximum value. For values of t_2 (outdoor dry-bulb temperature) upon which designs are ordinarily based see Table 1.

TABLE 1

State	City	Design dry bulb	Design wet bulb	Summer wind velocity, mph	Prevailing summer wind direction
Conn.	New Haven	95	75	7.3	S
Del.	Wilmington	95	78	9.7	SW
D.C.	Washington	95	78	6.2	S
Ill.	Chicago	95	75	10.2	NE
Maine	Portland	90	73	7.3	S
Mass.	Boston	92	75	9.2	SW
Md.	Baltimore	95	78	6.9	SW
N.J.	Trenton	95	78	10.0	SW
N.Y.	Albany	92	75	7.1	S
	New York	95	75	12.9	SW
Pa.	Philadelphia	95	78	9.7	SW
	Pittsburgh	95	75	9.0	NW
R.I.	Providence	93	75	10.0	NW

SOURCE: Abstracted from "Heating, Ventilating and Air Conditioning Guide," Chap. 8, 1939.

You will recall that in calculating winter heating loads we chose an outdoor temperature 10 to 15° above the lowest outdoor temperature ever recorded by the Weather Bureau. The choice of summer outdoor dry-bulb design temperatures is not quite so simple. In the first place, outdoor *wet-bulb* temperature is *just as important* as outdoor dry-bulb temperature in the complete over-all design of a summer air-conditioning system. (Outdoor wet-bulb temperature of course does not affect transmission heat flow into a cooled building.) The temperatures given in Table 1 are conditions that will not be exceeded more than 5 to 8 per cent of the time during an average summer cooling season (June through September) of

[1] Tables with roman numerals are given in the Appendix.

120 ten-hour days, or 1,200 hr.[1] In choosing outdoor-design conditions for a location where this information is not known, it is necessary to obtain Weather Bureau or other records of conditions during past years. As you design dry-bulb and wet-bulb temperatures, select *values that have not been exceeded more than about* 5 *per cent of the time* in previous average summers. Be sure to use *day*time data of past years as *night*time data will show approximately 10° lower in dry bulb and 5 to 7° lower in wet bulb.

Transfer into Buildings of Heat from Solar Radiation. Solar heat makes itself felt in different degrees of intensity, depending upon the type and thickness of the surface that it strikes and upon the location of that surface with respect to the sun. In other words, sun striking on a horizontal skylight and an ordinary side-wall window would have a different interior heating effect per square foot of glass area. Likewise, sun striking a smooth light-colored roof would have a different heating effect inside from the same sun striking a rough dark-colored roof. Finally, heat from the sun striking on a very thin wall or roof would be felt inside *sooner* than this same solar heat if it were to strike on a very thick wall or roof of the same construction material. In spite of the difference and complications implied by the foregoing, calculation of the effect of solar radiation has been reduced to an orderly and not difficult procedure by the A.S.H.V.E.

Intensity of Heat from Solar Radiation. Figure 1 gives curves showing the solar-heat intensity upon various surfaces of buildings. The top curve, labeled "surface *normal* to sun," refers to a surface held in such a manner that the sun's rays are always striking *at right angles*.

To obtain test data for a curve such as the type shown in Fig. 1, the surface would have to be moved every few minutes so that the sun's rays would keep striking *perpendicular* to the surface as the sun moved. The word "normal" is a mathematical word meaning "perpendicular to" or "at right angles to."

The value in Btu per square foot per hour as read in Fig. 1 is called "solar intensity" and is represented by I. Since Fig. 1 shows solar intensity for only one latitude, and since curves are shown only for walls facing

[1] For New York City the outdoor design conditions in Table 1, 95° D.B., 75° W.B. result in 67° dew point. In our experience with actual outdoor weather data in New York City in 1936, 1937, 1938, and 1939, outdoor dew points as high as 70° have occurred 10 to 15 per cent of the daytime cooling season hours. Outdoor wet-bulb temperatures of 76 to 78° have occurred 8 to 15 per cent of the daytime cooling hours, *depending upon the location in the city* where the reading was taken. We have similar experience regarding temperature readings in Stamford, Conn., and throughout portions of New Jersey along the sea coast and in the metropolitan area. Other air-conditioning designers have corroborating data, and there is a growing feeling that 95° dry bulb, 77° wet bulb is a more conservative outdoor-design condition for New York City. Opposed to using the latter values is a group that feels that three out of the last four summers have been abnormally high in wet-bulb and dew-point conditions in this area.

north, east, south, and west, the data from Fig. 1 is not complete enough for the full range of calculations. Therefore, the A.S.H.V.E. Research Laboratory has prepared Tables 2 to 5. These give solar intensity I for various hours of the day, on walls facing various directions, and also upon horizontal surfaces. Each table is for a different degree of latitude. (See any good geography map to determine the approximate latitude of any particular city.)

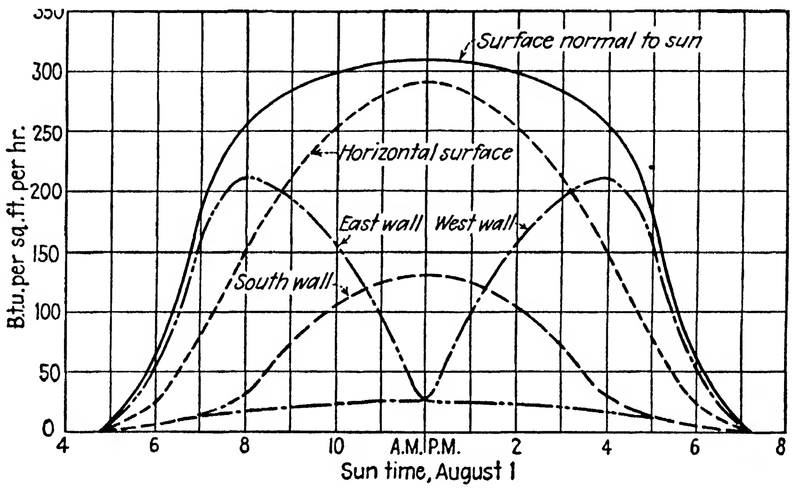


FIG. 1.—Curves giving solar intensity normal to sun, on horizontal surface and on walls for August 1.

Portion of Solar Heat Passing through Walls. Under this heading we shall consider walls of different construction and will also, of course, include roofs. Glass skylights and windows are not included, but will be taken up in a separate section. The solar intensity I discussed in the preceding paragraph and shown in Tables 2 to 5, gives the rate at which radiant heat from the sun *strikes upon the outside surface* of the wall or roof. Only a portion of this heat actually reaches the interior of the building. The amount of heat reaching the interior of the building depends upon two things: (1) the heat-transmission coefficient U for the wall or roof and (2) the color and type of *surface* of the wall or roof. In general, the higher the heat-transmission coefficient, the greater the portion of the solar heat that passes through the wall. As to the surface of the wall or roof, smooth-surface light-colored materials reflect the most heat and permit the least to enter the wall, while dark rough surfaces reflect very little heat and permit almost all of it to enter the wall.

To determine the amount of solar heat actually transmitted into the interior through a wall or a roof exposed to the sun, we may use the following formula

$$H_R = AFaI \tag{2}$$

where H_R = radiant solar heat transmitted inside, *Btu per hr*
A = area of wall or roof, *sq ft*
F = percentage of the absorbed solar radiation that is transmitted *through the wall or roof to the inside*, expressed as a *decimal*. (Read this from the left-hand vertical scale of Fig. 2; this factor varies depending upon the heat-transmission coefficient U for the wall or roof.)
a = percentage of the solar radiation that is *absorbed into the surface*, expressed as a *decimal*. (Read this value from Table 6.)
I = intensity of solar radiation impinging on or *striking* the surface, *Btu per hr per sq ft*. (Read this value from Tables 2 to 5, depending upon the location of the wall, the time of day, and the direction that the wall faces.)

TABLE 2.[1]—SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS, AND A HORIZONTAL SURFACE
For 30 Deg Latitude on July 21

Sun time	Intensity of solar radiation, Btu per sq ft per hr							
	North-east	East	South-east	South	South-west	West	North-west	Horizontal surface
4:59	0	0	0	..	...	...	...	0
5:00	1	1	0.3	..	...	...	...	0.01
6:00	47	51	24	..	...	...	...	9
7:00	136	160	90	..	..	...	...	68
8:00	151	205	136	..	...	...	...	147
9:00	127	189	140	8	...	...	...	214
10:00	79	141	122	31	...	...	...	265
11:00	21	78	85	45	..	...	...	296
12:00	...	...	36	50	36	...	...	305
1:00	...	...	...	45	85	78	21	296
2:00	...	...	...	31	122	141	79	265
3:00	...	...	...	8	140	189	127	214
4:00	...	...	...	..	136	205	151	147
5:00	...	...	...	..	90	160	136	68
6:00	...	...	...	..	24	51	47	9
7:00	...	...	...	..	0.3	1	1	0.01
7:01	...	...	...	..	0	0	0	0

[1] SOURCE: Reprinted by permission HOUGHTON, F. C., and GUTHERLET, CARL, Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, *A.S.H.V.E. Research Rept.* 853 (*A.S.V.H.E. Trans.*, vol. 36, 1930).

With all the values in Eq. (2) and with the information presented in Tables 2 to 6 and in Figs. 1 and 2, it might seem that the calculation of the actual solar heat transmitted *to the inside* of the building (H_R) is difficult. As a matter of fact it is simple because you select only a few definite items of data to substitute in Eq. (2). Let us illustrate this by a specific example

TABLE 3 [1].—SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS, AND A HORIZONTAL SURFACE
For 35 Deg Latitude on July 21

Sun time	Intensity of solar radiation, Btu per sq ft per hr							
	North-east	East	South-east	South	South-west	West	North-west	Horizontal surface
4:46	0	0	0	..	...	...	...	0
5:00	9	9	3	..	...	...	...	0.01
6:00	67	72	35	..	...	...	...	15
7:00	142	174	103	..	...	...	...	77
8:00	150	209	145	..	...	...	...	151
9:00	118	191	154	26	...	...	...	214
10:00	60	143	139	55	...	...	...	264
11:00	2	75	103	72	...	...	...	291
12:00	...	...	55	78	55	...	...	300
1:00	...	...	...	72	103	75	2	291
2:00	...	..	...	55	139	143	60	264
3:00	...	...	...	26	154	191	118	214
4:00	...	...	...	..	145	209	150	151
5:00	...	...	...	..	103	142	142	77
6:00	...	...	...	..	35	72	67	15
7:00	...	...	...	..	3	9	9	0.01
7:14	...	...	...	..	0	0	0	0

[1] SOURCE: Reprinted by permission HOUGHTON, F. C., and GUTBERLET, CARL, Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, *A.S.H.V.E. Research Rept.* 853 (*A.S.H.V.E. Trans.*, vol. 36, 1930).

Example. Suppose that you have a building having a 4-in. brick veneer on an 8-in. hollow tile backing, the interior being ¾-in. plaster on metal lath. This would be wall 38–D from Table XX, and from this table we see that $U = 0.25$ for this wall. Let us assume that the brick is red in color and 100 sq ft in area. Let us further assume that the wall faces southwest and that it is located in New York City, which has a latitude of 40°. Find the maximum rate of solar heat transmission H_R in Btu per hour.

Solution. We would use formula (2), and the values we would substitute would be as follows:

A = 100 sq ft, from statement of problem
F = 0.06, from Fig. 2 for a wall with a U factor of 0.25

a = 0.7, from Table 6, because a red brick has a medium dark surface
I = 168, from Table 4, wall facing southwest, time 3 P.M. (We use the value at 3 P.M. in this calculation because it is the maximum solar heat effect on a southwest wall.)

Then substituting in Eq. (2), we have

$$H_R = 100(0.06)(0.7)(168) = 706 \textit{ Btu per hr}$$

TABLE 4 [1].—SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE
For 40 Deg Latitude on July 21

Sun time	Intensity of solar radiation Btu per sq ft per hr							
	North-east	East	South-east	South	South-west	West	North-west	Horizontal surface
4:31	0	0	0	..	...	...	...	0
5:00	14	14	5	..	...	..	...	1
6:00	72	80	40	..	...	...	...	19
7:00	143	180	112	..	...	...	...	82
8:00	143	211	155	8	...	...	...	152
9:00	104	192	168	46	...	...	...	213
10:00	46	143	156	77	...	...	...	258
11:00	...	75	121	95	15	...	...	284
12:00	...	...	73	103	73	...	...	293
1:00	..	..	15	95	121	75	...	284
2:00	...	..	...	77	156	143	46	258
3:00	...	...	...	46	168	192	104	213
4:00	...	...	...	8	155	211	143	152
5:00	...	...	...	..	112	180	143	82
6:00	..	...	...	..	40	80	72	19
7:00	...	.	..	..	5	14	14	1
7:29	..	...	...	..	0	0	0	0

[1] SOURCE: Reprinted by permission HOUGHTON, F. C., and GUTBERLET, CARL, Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, *A.S.H.V.E. Research Rept.* 853 (*A.S.H.V.E. Trans.*, vol. 36, 1930).

In making calculations similar to the above, the personal judgment factor enters only in the selection of the proper absorption coefficient a from Table 6. In this connection, the smallest value shown for a, 0.4, should be used only for a building that is to be kept very clean at all times. Compare a new white-stone surface building to an identical building five or six years old, and you will see that the older building is really a dark gray. (Compare one of the newest Rockerfeller Center buildings to one of the older buildings.) Therefore, it is unsafe to use a value smaller than 0.5

for *a* unless the building is to be painted or cleaned every year. Ordinary values used for *a* will be between 0.7 to 0.9.

Time Lag in Transmission of Solar Heat. In the typical problem that we have just solved, we obtained our value for *I* from Table 4 at 3 P.M. This does not mean that the full impact of this solar heat load would be

TABLE 5[1].—SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE
For 45 Deg Latitude on July 21

Sun time	Intensity of solar radiation Btu per sq ft per hr							
	North-east	East	South-east-	South	South-west	West	North-west	Horizontal surface
4:26	0	0	0	...	...	...	...	0
5:00	25	24	9	...	...	...	...	2
6:00	89	99	52	...	...	...	...	26
7:00	149	194	125	...	...	...	...	90
8:00	140	219	171	22	...	...	...	156
9:00	92	194	183	65	...	...	...	210
10:00	33	144	171	98	...	...	...	251
11:00	...	75	139	121	32	...	...	274
12:00	...	...	91	128	91	...	...	282
1:00	...	...	32	121	139	75	...	274
2:00	...	...	...	98	171	144	33	251
3:00	...	...	...	65	183	194	92	210
4:00	...	...	...	22	171	219	140	156
5:00	...	...	...	...	125	194	144	90
6:00	...	...	...	...	52	99	89	26
7:00	...	...	...	...	9	24	25	2

[1] SOURCE: Reprinted by permission HOUGHTON, F. C., and GUTHERLET, CARL, Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, *A.S.H.V.E. Research Rept.* 853 (*A.S.H.V.E. Trans.*, vol. 36, 1930).

felt *inside the building* at 3 P.M., or even at 3:30 or 4 P.M. There is a definite *time lag* involved in the flow of this heat through various types of walls. This time lag depends *approximately* on the *U* factor for the particular wall or roof. In many buildings, having rather heavy walls, the solar heat that impinges upon the southwest and west walls late in the afternoon does not reach the inside of the building until after the air-conditioning plant is shut down for the day. In buildings such as theaters, which are occupied in the evening, the full impact of the afternoon heat load on the walls and roof often does not reach the interior, to make itself

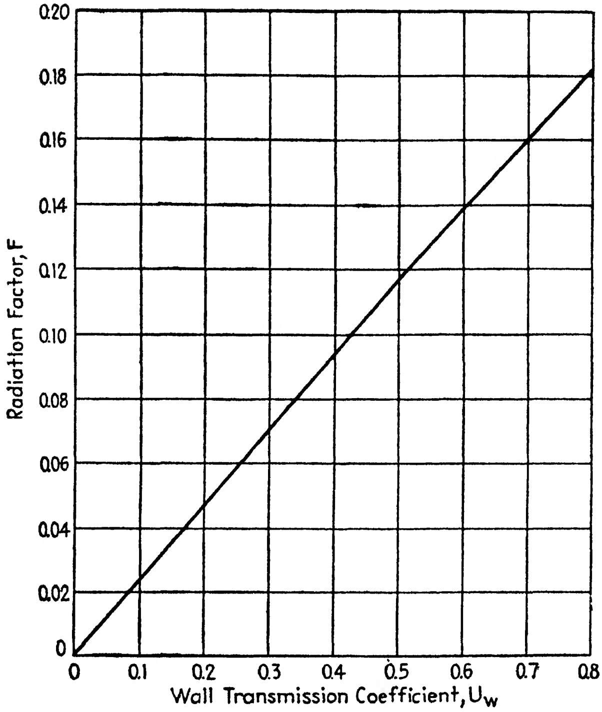


Fig. 2.—Solar radiation factors.

Table 6.[1]—Solar Absorption Coefficients for Different Building Materials

Surface	Material	Absorption coefficient, a
Very light colored . . .	White stone Very light colored cement White or light cream-colored paint	0.4
Medium dark.	Asbestos shingles Unpainted wood Brown stone Brick and red tile Dark-colored cement Stucco Red, green, or gray paint	0.7
Very dark colored . . .	Slate roofing Tar roofing materials Very dark paints	0.9

[1] Source: Reprinted by permission Faust, F. H., Levine, L., and Urban, F. O., A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, *A.S.H.V.E. Trans.* vol. 41, 1935.

felt by an increase in the cooling load, until late in the evening when the outdoor temperature is below maximum and when the theater is perhaps partly empty.

TABLE 7[1].—TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

Type and thickness of wall or roof	Time lag, hr
2-in. pine	1½
6-in. concrete	3
4-in. gypsum	2½
3-in. concrete and 1-in. cork	2
2-in. iron and cork (equivalent to ¾-in. concrete and 2.15-in. cork)	2½
4-in. iron and cork (equivalent to 5½-in. concrete and 1.94-in. cork)	7¼
8-in. iron and cork (equivalent to 16-in. concrete and 1.53-in. cork)	19
22-in. brick and tile wall	10

[1] SOURCE: From *A.S.H.V.E. Research Repts.* 853 and 923.

Table 7 gives data taken from *A.S.H.V.E. Research Papers* regarding time lag in transmissions of solar radiation through walls and roofs. These data result from a study of wall and roof slabs rather than actual buildings, but they give an approximate idea of the time lag to be expected in various structures. In estimating time lag, choose (in Table 7) the wall nearest

TABLE 8[1].—SOLAR RADIATION TRANSMITTED THROUGH SHADED WINDOWS

Type of appurtenan	Finish facing sun	Per cent delivered to room
Canvas awning	Plain	28
Canvas awning	Aluminum	22
Inside shade, fully drawn	Aluminum	45
Inside shade, one-half drawn	Buff	68
Inside Venetian blind, fully covering window, slats at 45 deg	Aluminum	58
Outside Venetian blind, fully covering window, slats at 45 deg	Aluminum	22

[1] SOURCE: BLACKSHAW, J. L., and HOUGHTEN, F. C., Radiation of Energy Through Glass, *A.S.H.V.E. Research Repts.* 974–975 (*A.S.H.V.E. Trans.*, vol. 40, 1934); HOUGHTEN, F. C., GUTHERLET, CARL, BLACKSHAW, J. L., *A.S.H.V.E. Trans.*, vol. 40, 1934.

in construction to your actual wall, and *assume* that time lag varies approximately with U factor in comparing your actual wall to the wall selected in Table 7.

Transmission through Glass of Solar Radiation. Windows and skylights being made of thin transparent glass instead of thick opaque masonry, etc., permit a greater percentage of solar heat to pass into the room. The thickness of the glass does not greatly affect the amount of solar heat transmitted to the inside; so the same general method can be used for ordinary $\frac{1}{8}$ in. residential window glass and for thick plate-glass windows in commercial buildings. The main variable that may affect the proportion of solar heat that passes through the window and into the interior is the *degree of shading* present to interfere with the passage of this solar heat. Table 8 gives the necessary information in this connection.

We shall calculate solar heat transmission through windows or skylights by means of formula (3).

$$H_G = A_G f I \tag{3}$$

where H_G = solar radiation transmitted through the glass, *Btu per hr*
f = percentage of solar radiation transmitted to the inside, expressed as a *decimal.* (Determine this value from Table 8 using $f = 1.00$ for a completely bare unshaded window having no awning, Venetian blind, window shade.)
I = intensity of solar radiation striking the glass, Btu per hr per sq ft. [Read this value from Tables 2 to 5, just as it was read for use in formula (2).]

Problem. Let us take a window having a *net glass area* of 20 sq ft, having a light-colored inside shade fully drawn. Windows in ordinary office buildings have approximately this area. Let us assume that this window faces south and that the outdoor temperature is 95° and the interior temperature is 80°. Let us now calculate the heat that would enter the room through this window, first by means of normal *transmission heat flow* and then because of radiant solar heat.

Solution.

Transmission heat flow

Using Eq. (2) from Chap. XIV,

$$H = AU(t_2 - t_1)$$

where $A = 20$
$U = 1.13$
$(t_2 - t_1) = 15°$

we have

$$H = 20(1.13)(15) = 339 \text{ Btu per hr}$$

Solar heat transmitted

We use Eq. (3) from this chapter, the values being

$A_G = 20$ sq ft
$f = 0.45$, from Table 8
$I = 103$ at 12 o'clock, from Table 4

then

$$H_G = 20(0.45)(103) = 927 \text{ Btu per hr}$$

From the above you can see that the amount of heat that enters the room because of solar radiation is approximately *three times* that which we would calculate as flowing inward by means of ordinary heat transmission. If a higher factor for f from Table 8 had been used, the proportion of radiant heat would be found greater. What actually happens is that the entire pane of glass becomes warmer owing to the radiant heat from the sun. In fact, the glass may become considerably higher in temperature than the outdoor dry-bulb temperature of the air. In this case, no ordinary transmission heat flow occurs; instead some heat flows from the warm glass to the outdoor air. Therefore when the solar heat transmitted through the glass, as calculated by Eq. (3), is greater than the heat that would be calculated as flowing inward due to ordinary transmission, *the transmission heat may be neglected and the solar heat alone may be used.* Conversely, when the heat gain due to ordinary transmission exceeds that calculated because of solar intensity, the latter may be neglected and the entire heat gain may be considered as that resulting from ordinary transmission.

Heat Load Due to Human Occupancy. In Chap. XIII we saw that human beings gave off both sensible and latent heat to the air. This heat load is an important factor in summer cooling load, especially when the space to be air-conditioned has a high human occupancy.

Infiltration. Outdoor air that enters the conditioned space because of infiltration or air leakage is higher in temperature and higher in dew point than the conditions that are being maintained inside. This air which leaks in is, therefore, the same as an internal sensible heat load by the amount that its sensible heat exceeds the sensible-heat content desired indoors. We may calculate the amount of this sensible heat by Eq. (4).

$$H_s = 0.24\left(\frac{Q}{v}\right)(t_2 \times t_1) \tag{4}$$

where H_s = sensible heat load to lower the temperature of the air from the outdoor temperature to the indoor temperature, Btu per hr

0.24 = specific heat of the air, Btu per lb

Q = infiltration, cu ft of air per hr

v = specific volume of the outdoor air, cu ft per lb

t_2 = outdoor dry-bulb temperature

t_1 = indoor dry-bulb temperature

In Eq. (4), the quantity Q/v will be subject to only slight variations because of changes in v. For outdoor summer air, v will ordinarily be between 13.8 and 14.3 cu ft per lb. Check the outdoor conditions given in Table 1, and you will find that this is true. Therefore, v is often taken arbitrarily as 14.0 cu ft per lb for use in Eq. (4) and also in Eq. (5).

We have just calculated the contribution to the internal sensible heat load caused by infiltration air. In addition, we ordinarily want the latent heat load caused as well as the total heat load caused. We obtain the total heat load by means of Eq. (5).

$$H_t = \frac{Q}{v}(H_{to} - H_{ti}) \tag{5}$$

where H_t = total heat load caused by the infiltration air, Btu per hr

$\frac{Q}{v}$ = same as in Eq. (4), *i.e.*, pounds per hour of infiltration air

H_{to} = total heat of the outdoor air, Btu per lb

H_{ti} = total heat of the indoor air, Btu per lb

If we let H_L equal the latent heat load in Btu per hour due to infiltration air, we have Eq. (6)

$$H_L = H_t - H_s \tag{6}$$

Of course it is possible to determine the latent heat in terms of grains of water per minute or per hour introduced by the infiltration air; in some cases this is necessary, while in others it is not. If needed, it can easily be determined from the difference in moisture content per pound between the indoor conditions and the outdoor conditions.

Heat from Machinery and Appliances. When we calculated winter heating load, we stated that we could not deduct the heat caused by all heat-producing equipment within the heated space because some of this equipment might be idle and therefore not contributing heat. In summer, for which we are now calculating cooling load, we have the opposite situation. We must consider any heat-generating machinery or appliance that is *likely to be operating* at the same time that the air-conditioning plant is operating. If there is a question as to whether a certain appliance will be operating, we should consider that *it will be* operating. If we do not do this, at times when such appliances are operating, the heat load might be sufficient to add excessive sensible or latent heat to the air inside the conditioned space.

Looking at this another way, we may say that *in winter*, heat-generating appliances inside are *an advantage* but may be counted on only *if they are certain*. In summer, such devices are *a disadvantage* because we must supply cooling capacity to counteract them. They must be figured in calculating the cooling load if there is a *good possibility that they will be operating*, because their effect when operating is a hindrance rather than a help to the problems of the air-conditioning system.

Table 11 shows the heat added by miscellaneous electric, gas, and steam-heated equipment. Considerable judgment is required in using values

TABLE 9.—RELATION BETWEEN METABOLIC RATE AND ACTIVITY*

Activity	Hourly metabolic rate for avg person or total heat dissipated, Btu per hr	Hourly sensible heat dissipated, Btu per hr	Hourly latent heat dissipated, Btu per hr	Moisture dissipated per hr	
				Grains	Pounds
Average person seated at rest[1]	384	225	159	1,070	0.153
Average person standing at rest[1]	431	225	206	1,390	0.199
Tailor[2]	482	225	257	1,740	0.248
Office worker moderately active	490	225	265	1,790	0.256
Clerk, moderately active, standing at counter	600	225	375	2,530	0.362
Bookbinder[2]	626	225	401	2,710	0.387
Shoemaker[2]; clerk, very active standing at counter	661	225	436	2,940	0.420
Pool player	680	230	450	3,040	0.434
Walking 2 mph[3,4]; light dancing	761	250	511	3,450	0.493
Metalworker[2]	862	277	585	3,950	0.564
Painter of furniture[2]	876	280	596	4,020	0.575
Restaurant serving, very busy	1000	325	675	4,560	0.651
Walking 3 mph[2]	1050	346	704	4,750	0.679
Walking 4 mph[3,4]; active dancing, roller skating	1390	452	938	6,330	0.904
Stone mason[2]	1490	490	1000	6,750	0.964
Bowling	1500	490	1010	6,820	0.974
Man sawing wood[2]	1800	590	1210	8,170	1.167
Slow run[4]	2290				
Walking 5 mph[3]	2330				
Very severe exercise[5]	2560				
Maximum exertion different people[4]	3000–4800				

* Metabolism rates noted based on tests actually determined from the following authoritative sources:
[1] A.S.H.V.E. Research Laboratory.
[2] Becker and Hamalainen.
[3] Douglas, Haldane, Henderson and Schneider.
[4] Henderson and Haggard.
[5] Benedict and Carpenter.

Metabolic rates for other activities estimated. Total heat dissipation integrated into latent and sensible rates by actual tests for metabolic rates up to 1250 Btu per hr, and extrapolated above this rate Values for total heat dissipation apply for all atmospheric conditions in a temperature range from approximately 60 to 90°F dry bulb. Division of total heat dissipation rates into sensible and latent heat holds only for conditions having a dry-bulb temperature of 79°F. For lower temperatures, sensible-heat dissipation increases and latent heat decreases, while for higher temperatures the reverse is true.

SOURCE: From "Heating, Ventilating and Air Conditioning Guide, Chap. 3, 1939.

for equipment where the sensible and latent heat values are marked with an asterisk; see footnote at the bottom of the table.

Electric motor heat may be figured by formulas (8) and (9) of Chapter XV. We are *not* reprinting these because they are such basic equations, going back to the fundamentals in Chap. V, that you should be able to

TABLE 10.—HEAT DISSIPATED BY AVERAGE ADULT PERSONS

Activity	Total heat dissipated, Btu per hr	Hourly sen heat dissipated, Btu per hr	Hourly latent heat dissipated, Btu per hr	Moisture dissipated	
				Grains per hr	Grains per min
Average person seated at rest	400	220	180	1,210	20.2
Average person standing at rest	430	225	205	1,380	23.0
Office worker moderately active	490	225	265	1,790	29.9
Store clerk moderately active	600	225	375	2,530	42.0
Walking 2 mph, or slow dancing	750	250	500	3,380	45.3
Waiters, very busy	1000	325	675	4,560	76.0
Walking 4 mph, active dancing, active bowling	1390	450	940	6,330	105.3
Very vigorous exercise	2500	*	*	*	
Maximum exertion (varies)	3000–4800	*	*		

* Proportion varies—to assume 35 per cent sensible and 65 per cent latent would not be badly in error.

Values apply for atmospheric conditions of 65 to 85°F with moderate humidities.

Proportion of sensible and latent heat is correct for air with a dry-bulb temperature of 80°. For lower temperatures, the sensible-heat portion increases and the latent-heat portion decreases, and vice versa.

derive these equations at any time. If you fail to derive them you may always look them up.

Electric lights, whether ordinary Mazda incandescent lamps, fluorescent lights, or arc lights, produce heat at the rate of 3415 Btu per hr for each kilowatt of electric power consumption. In other words, if a restaurant uses ninety 100-watt lamps, the power consumption is 9,000 watts, or 9 kw; the heat produced is 9(3415) = 30,700 Btu per hr. Obviously the heat from electric lights is *sensible* heat.

Heat Load from Transmission Heat Flow into Ducts. If the recirculated air from the conditioned space, which is about 80°F, travels through a duct that passes through a *room warmer than the air in the duct,* heat will flow from the warm air through the duct and the temperature of the air in the duct will be raised. The duct carrying the *cold* conditioned air to the conditioned space is liable to suffer in this manner to a much greater extent, because of the low-temperature air being carried. To calculate this heat loss,[1] use the regular transmission heat-flow formula

$$H = AU(t_2 - t_1)$$

where the terms in the equation *for ducts* are

A = area of the duct (*all four sides*), sq ft
U = over-all coefficient for the duct (see Table 12)
t_2 = air temperature outside duct
t_1 = air temperature inside duct

Summary of Components of Cooling Load. Thus far we have discussed all components of cooling load which are in effect heat sources *within* the conditioned space. As given in this chapter, these components are as follows:

1. Ordinary transmission heat load
2. Solar heat load on
   a. Walls and roofs
   b. Glass
3. Human heat load
4. Infiltration load
5. Machinery and appliances
6. Duct heat losses

Upon determining the total cooling load by adding the components listed above, we would have a load that should be called "total *internal* cooling load." In other words, if an air-conditioning system were installed using 100 per cent recirculated air with no fresh outdoor air, the capacity of the plant would be that determined by totaling the items listed above. Such a plant should be capable of maintaining the desired interior psychrometric conditions.

Ventilation Load or Fresh-air Load. As we saw in Chap. XIII a certain amount of fresh outdoor air must be continually introduced in order to provide proper ventilation. This outdoor air must of course enter through the conditioning unit. Imagine that a *separate conditoning unit* is provided for handling the fresh outdoor-air component of the air supply and that this conditioning unit delivers its air at the precise dry-bulb and wet-bulb temperatures selected as the *inside design conditions.* This conditioned outdoor air would then have no heating or cooling effect upon

[1] For a more exact but much more complicated method of heat flow through ducts, see A.S.H.V.E. "Guide," Chap. 39, pp. 746–748, 1939.

TABLE 11.—HEAT GAIN FROM VARIOUS SOURCES

Source	Btu per hour		
	Sensible	Latent	Total
Electric heating equipment			
Electric equipment, dry heat, no evaporated water	100%	0%	100%
Electric oven, baking	80%	20%	100%
Electric equipment, heating water, stewing, boiling, etc.	50%	50%	100%
Electric lights and appliances per watt (dry heat)	3.4	0	3.4
Electric lights and appliances per kilowatt (dry heat)	3413	0	3413
Electric motors per horsepower	2546	0	2546
Electric toasters or electric griddles	90%	10%	100%
Coffee urn, large, 18 in. diameter, single drum	2000	2000	4000
Small, 12 in. diameter, single drum	1200	1200	2400
Approx. connected load per gallon of capacity	600	600	1200
Electric range, small burner	*	*	3400
Large burner	*	*	7500
Oven	8000	2000	10000
Warming compartment	1025	0	1025
Steam table, per square foot of top surface	300	800	1100
Plate warmer, per cubic foot of volume	850	0	850
Baker's oven, per cubic foot of volume	3200	1300	4500
Frying griddles, per square foot of top surface	*	*	4600
Hot plates, per square foot of top surface	*	*	9000
Hair dryer in beauty parlor, 600 watt	2050	0	2050
Permanent-wave machine in beauty parlor, 24–25 watt units	2050	0	2050
Gas-burning equipment			
Gas equipment, dry heat, no water evaporated	90%	10%	100%
Gas-heated oven, baking	67%	33%	100%
Gas equipment, heating water, stewing, boiling, etc	50%	50%	100%
Stove, domestic type, no water evaporated, per medium-size burner	9000	1000	10000
Gas-heated oven, domestic type	12000	6000	18000
Stove, domestic type, heating water, per medium-size burner	5000	5000	10000
Residence gas range, giant burner (about 5½ in. diameter)	*	*	1200
Medium burner (about 4 in. diameter)	*	*	10000
Double oven (total size 18 x 18 x 22 in. high)	*	*	18000
Pilot	*	*	250
Restaurant range, 4 burners and oven	*	*	100000
Cast-iron burner, low flame, per hole	*	*	100
High flame, per hole	*	*	250
Simmering burner	*	*	2500
Coffee urn, large, 18 in. diameter, single drum	5000	5000	10000
Small, 12 in. diameter, single drum	3000	3000	6000
Per gallon of rated capacity	500	500	1000
Egg boiler, per egg compartment	2500	2500	5000
Steam table or serving table, per square foot of top surface	400	900	1300
Dish warmer, per square foot of shelf	540	60	600
Cigar lighter, continuous-flame type	2250	250	2500
Curling-iron heater	2250	250	2500

TABLE 11.—HEAT GAIN FROM VARIOUS SOURCES—*Continued*

Source	Btu per hour		
	Sensible	Latent	Total
Gas-burning equipment—*Continued*			
Bunsen-type burner, large, natural gas	*	*	5000
Large, artificial gas	*	*	3000
Small, natural gas	*	*	3000
Small, artificial gas	*	*	1800
Welsbach burner, natural gas	*	*	3000
Artificial gas	*	*	1800
Fishtail burner, natural gas	*	*	5000
Artificial gas	*	*	3000
Lighting-fixture outlet, large, 3 mantle 480 cp	4500	500	5000
Small, 1 mantle, 160 cp	2250	250	2500
1 cu ft of natural gas generates	900	100	1000
1 cu ft of artificial gas generates	540	60	600
1 cu ft of producer gas generates	135	15	150
Steam-heated equipment			
Steam-heated surface not polished, per square foot of surface	330	0	330
Steam-heated surface polished, per square foot of surface	130	0	130
Insulated surface, per square foot	80	0	80
Bare pipes, not polished, per square foot of surface	400	0	400
Polished, per square foot of surface	220	0	220
Insulated pipes, per square foot	110	0	110
Coffee urn, large, 18 in. diameter, single drum	2000	2000	4000
Small, 12 in. diameter, single drum	1200	1200	2400
Egg boiler, per egg compartment	2500	2500	5000
Steam table, per square foot of top surface	300	800	1100
Miscellaneous			
Heat liberated by food per person, as in a restaurant	30	30	60
Heat liberated from hot water used direct and on towels per hour—barber shops	100	200	300

* Per cent sensible and latent heat depends upon use of equipment; dry heat, baking or boiling.

the conditioned space, as its dry-bulb and dew-point temperatures would be the same as those in the conditioned space.

When fresh outdoor air is introduced through the main conditioning unit, as it usually is, it is of course brought *lower in dry-bulb* temperature and *dew point* than the inside design conditions for the conditioned space. Therefore, if 100 lb per min of fresh air is introduced, 100 lb *less recirculated* air is used than would be used if the installation were 100 per cent recirculation. As far as the heat load on the conditioner is concerned, this outdoor air is worse than recirculated air *by the difference in sensible- and* *latent-heat content between fresh outdoor air and recirculated air.* If we assume recirculated air to be at the exact dry bulb and wet bulb maintained inside the conditioned space, the net effect of outdoor air on the *conditioner heat load* is the same as that described in the preceding paragraph. In other words, outdoor air introduced for ventilation gives *a net increase* to the internal heat load we have calculated, depending upon the quantity of outdoor air and the sensible- and latent-heat content of this outdoor air compared with the sensible- and latent-heat content of the indoor air.

The internal heat load should be first calculated, assuming no outdoor air, by the method summarized above. The amount of fresh outdoor air to be used should then be determined, and the increase in load which it

TABLE 12.—HEAT TRANSMISSION THROUGH DUCTS

Type of Duct Construction	U Factor
Uninsulated sheet metal	1.3
Sheet-metal insulated with ½-in. rigid insulating board	0.4
Sheet-metal with 1-in. Corkboard, ⅜-in. thick gypsum plaster finish	0.25

(Based on tight ducts, still air outside, up to 2,000 fpm velocity inside)

causes should be calculated; this additional load is ordinarily called "ventilation load" or "fresh air load." We may use Eq. (4), (5), and (6) to calculate fresh air load, using Q in these equations as the cubic feet *per hour* of fresh outdoor air that is to be used.

Fresh air load may be thought of in this manner. No fresh air is needed to maintain the desired psychrometric conditions inside the conditioned space. If a 100 per cent recirculation job were installed the conditioner would be of the capacity indicated by the calculation of total *internal* cooling load. A refrigeration machine required in connection with this conditioner would also be of this capacity. As far as overcoming the leakage of heat into the conditioned space, plus any source of heat located inside the conditioned space, such an installation would operate and would produce the inside design conditions. As fresh air is added, the capacity of the conditioner as to heat removal ability must be increased, although its capacity in cubic feet per minute of air probably does not change. Similarly, since the refrigeration equipment provides a means of removing heat from the conditioner, the capacity of the refrigeration plant must be increased by the amount of fresh air or ventilation load. You will see in a later chapter (perhaps advanced students can already demonstrate this to themselves) that a certain air-conditioning job requiring a 50-ton installation *based on zero fresh air* might require a 100-ton installation *using 100 per cent fresh air.* If these were actual figures this plant would require 75 tons of refrigeration using 50 per cent fresh air, 62.5 tons of refrigeration using 25 per cent fresh air, etc.

At the present time do not be disturbed if you do not have a thorough and complete understanding of the calculation of fresh-air load as discussed above. For the present, it will be sufficient if you learn the mechanics of calculating this load. We are sure that you will learn this as you complete the problems at the end of this chapter. At that time you will have no difficulty in completely understanding the effect of fresh air load upon the capacity on a summer air-conditioning plant.

Maximum Cooling Load—When Does This Occur? Thus far in this chapter you have learned the procedure and the nature of the calculations involved in calculating summer cooling load. Suppose that you were to take a certain building and calculate summer cooling load as outlined in this chapter. For the air conditioning of the top floor of a building liberally lighted with skylights, would you include the solar-heat load due to all the skylights, plus the *maximum* interior electric lighting load? Would you calculate solar-heat load on the east, south, and west wall and *add all three* of these values *to your total* cooling load? Think seriously about the answers to these questions, and you will see that if a top floor is liberally lighted by skylights all electric lights will *not be lighted* at a time when the *sun effect upon the skylights is great enough to be an important source of heat load.* Furthermore, the maximum sun effect on the east wall occurs at about 8 A.M., at which time there is zero sun effect on the west wall, and almost zero sun effect on the south wall. From Fig. 1 it is obvious that the sun effect on the east and west walls occurs at totally different times.

Now consider the question of the effect of *time lag* in the flow of solar heat through walls. Let us assume a room having one wall with windows *facing east* and another wall of the same size with windows *facing south*, the other two walls of the room being inside walls. At 8 A.M. this room is subject to maximum sun effect on the east wall. The portion of this 8 A.M. solar-heat radiation that impinges on the east *windows* enters the conditioned space *immediately.* The portion of the solar heat striking the masonry or solid portion of the east wall may not reach inside until several hours later, sometime between 10 A.M and 1 P.M., depending upon time lag. By the time this heat reaches the inside of the conditioned space, the sun effect on the south wall is at a maximum and the portion striking the *windows* of the south wall is entering the conditioned space *immediately.* Hence it is likely that the maximum solar-heat effect *inside this room* might occur sometime between 11 A.M. and 1 P.M. This maximum would *not* include the solar heat striking the *solid portion* of the *south wall,* because time lag would delay this heat and it would finally enter the conditioned space between 3 and 4 P.M. By this time most of the sun effect on the south windows would have disappeared and since the room does not have an exposed west wall, the solar heat on the room would diminish as the afternoon progressed.

What does this mean in the way of an orderly procedure by which we can determine the time of day at which the maximum effect on solar heat is *actually reaching the inside of the conditioned space* and, hence, the time at which our internal cooling load is a *maximum*? In the case of certain buildings, especially those in which the glass area is a very large percentage of the total wall area, *it may* be obvious that the total cooling load for the building will be at a maximum at a certain time of day. Take for instance an office building whose east and west walls are not exposed at all, the area of the south wall being 50 per cent glass and 50 per cent masonry. If the windows are to have inside shades half drawn or fully drawn inside Venetian blinds, Table 8 indicates that 58 to 69 per cent of the solar heat on the windows will be delivered into the room immediately. For a building such as this, assume 2,000 sq ft of glass area and 2,000 sq ft of medium dark-colored masonry wall having an over-all U factor of 0.3 Btu/(hr) (sq ft) (°F), and assume the building in New York City at a 40° latitude. From Tables 4, 6, 7, and 8 and from Fig. 2, calculate the solar radiation transmitted through the glass and through the solid walls at the *maximum* intensity, which will be from 11 A.M. to 1 P.M. You will find that the solar heat actually transmitted to the inside *through the glass* is greatly in excess of that transmitted through the solid walls. Furthermore, the heat passing through the glass enters the inside *immediately*, while that entering through the wall is subject to a time lag of 2 to 4 hr. Because of this time lag, the solar heat passing through the solid walls would reach the inside after the peak or maximum sun effect on the glass is passed. Therefore, it is possible to state immediately, for the building with the south exposed wall which we are discussing, that the maximum load from solar heat would occur from 11 A.M. to 1 P.M. We *can be sure* that the maximum would occur during this time interval *before* we make any exact calculations.

There are many buildings having more complicated exposures, including those in which the interior lighting load and human-occupancy load may change from one portion of the building to the other. With such buildings even a highly experienced air-conditioning technical man *cannot* state definitely, on a quick inspection, the time of day at which various components of cooling load might be at a maximum. Consider a night club which may operate as a bar and grill in the afternoon. The occupancy load will be small in the afternoon, and the lighting load will probably also be small at this time. Solar heat will be felt during the afternoon, reaching the inside immediately, with part of it not reaching inside until perhaps 7 to 9 P.M. This night club will be subject to no sun effect except that solar heat which may be entering during the early evening because of time lag. After 9 P.M. the heat from people and lights (plus the latent heat load from foods served) may be four or five times greater than in the afternoon. It is not possible to examine these general conditions and quickly state the exact time at which the cooling load will be a maximum. Remember outdoor dry-bulb temperature may be at least 5° lower at 9 P.M. than at 3 P.M. Therefore, it is necessary to calculate cooling loads for this establishment at *several different times of day*. These times should be chosen such that some components of cooling load are at a maximum at each time you choose. For instance, if the night club had an exposed west wall, 5 to 6 P.M. should be chosen because sun effect on the west wall is high at this time and the occupancy load (cocktail hour) is probably always high. A second time of day should be chosen at the time at which it is estimated that the lagging solar load through solid walls will reach the interior. The load should be figured for a *third set* of conditions *late at night* when the establishment is most crowded.

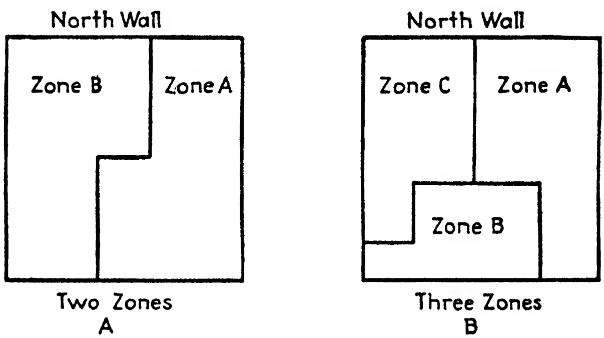


FIG. 3.—Method of zoning a single floor.

Calculation of Cooling Load by Zones. When entire floors of large buildings are to be cooled, it is usually found that these floors have exposed walls with windows on *three* or *four sides*. Furthermore, the interior is usually broken up somewhat by partitions. In this case, we sometimes divide the building into two or three zones, depending upon floor area and height. Figure 3 illustrates how such a floor might be divided into two or three zones.

In buildings in which separate zones are used, the cooling load is calculated, *separately for each zone*, at two or three different times of day. Your personal judgment must determine the times of day and also whether you should choose *two* or *three* different *times*. Let us suppose that for one floor of an office building, divided in three zones, we have calculated cooling loads (in terms of tons of refrigeration) as follows (see Fig. 3*b*):

From Tables 13 and 14 we see that zone *A* must have a conditioner of 60 tons capacity, zone *B* a conditioner of 40 tons capacity, and zone *C* a conditioner of 60 tons capacity, *if three separate conditioners are used.* Even though these three conditioners total 160 tons capacity, as shown in Table 14, from Table 13 we see that 125 tons is the maximum capacity being used at any one time. Therefore, 125 tons of refrigeration can be installed, operating to chill water which would be pumped to the three different conditioners. This arrangement is particularly satisfactory if at least two refrigeration machines are installed to operate the same evaporator. Under these conditions, if it were desired to hold an evening meeting in the building, *one* refrigeration machine, the chilled-water pump,

TABLE 13.—TABULATION OF COOLING-LOAD DATA FOR ZONED BUILDING

Zone	Cooling load, tons		
	8–10 A.M.	11 A.M.–1 P.M.	3–5 P.M.
A	60	40	30
B	25	40	35
C	30	40	60
Total	115	120	125

and *one* conditioner could be operated to cool the zone in which the meeting was to be held.

In some cases a zoned building, such as we have been discussing, might be served from one single large conditioner. In this case, the large conditioner and its refrigeration plant would both have a capacity of 125 tons. Dampers would be used such that zone *A* would receive a maximum of

TABLE 14.—COMPARISON OF PEAK LOADS IN VARIOUS ZONES

Zone	Maximum load, tons	Time maximum load occurs
A	60	Morning
B	40	11 A.M.–1 P.M.
C	60	Late afternoon
Total	160	

conditioned air in the morning during its period of peak load. As its load dropped and the load in the south zone (zone *B*) increased, automatic controls would reduce the amount of cold conditioned air going to zone *A* and increase the amount of air going to zone *B*. (The details of operation of automatically controlled air-volume dampers such as these will be discussed in a later chapter. At this time simply realize that there are such automatic dampers available for installations where the conditioned air quantity to different zones must be varied.) In late afternoon, zone C would get the greatest share of the conditioned air.

From the preceding paragraphs you can see that in the case of a zoned installation it is necessary to treat each zone separately in calculating cooling load, two or three different times of day are chosen, and the coolin load for all three zones is calculated for the three times of day selected. If 9 A.M. is selected as one time of day, the load for all zones must be calculated as of 9 A.M. Similarly, for any other time of day, the cooling loads for all zones must be calculated at these same times. It will then be found that the maximum cooling requirements for each zone occurs at a different time than for the other zones. We shall then always find the following general conclusions, which were illustrated for a specific case in Tables 13 and 14:

1. The peak load for zones will depend upon the exposure of their outside walls. East zones *generally* have a maximum load in the morning, south zones around noon, and west zones from 3 to 4 P.M.[1]

2. Arrangements must be made to supply the *maximum cooling requirements* for each zone *at the time it occurs,* but at this time the cooling-load requirement of other zones will be far below their respective maximums.

3. If we *total* the *maximums* for each zone, we obtain a much higher figure than the total cooling capacity that we actually need to install.

4. We should total the cooling requirements for *each zone* at *each of the various times* of day for which this information has been computed. These totals will probably be different, and the largest of these totals determines the *total cooling capacity that must be installed.* (Table 13 illustrates this point.)

Conclusion. Problems in calculating cooling load are solved sometimes with mathematical exactness and sometimes by quick rough approximate methods. How you calculate a cooling load depends upon the purpose for which you are to *use your result.* We may consider three general purposes for which cooling-load calculations are made. First we have the least accurate type of calculations, those which are used to give a rough offhand "guess" as to the proper size of plant required for a given installation. If a customer wants to know whether his establishment will require a 25- or 50-ton plant, you may calculate a rough cooling load, using approximate short-cut methods. After you become practiced, you can often make such a calculation without looking up exact U factors, just guessing at the dimensions of the building. *Rough cooling calculations made in this manner are not to be used for any other purpose.*

When a price quotation is to be made for a proposed installation, it is possible to make calculations for cooling load in which inaccuracies may be permitted in different components of cooling load. For instance you

[1] Not always true, especially for walls with no windows.

may not have accurate information regarding lighting load, and you may take a rough figure that you believe a little *too low*. In this case, *add* a little sensible heat load in calculating transmission or sun effect to counterbalance the low lighting figure. Care should be exercised that the total load figure is accurate within 5 per cent for installations over 100 tons, within 10 per cent for 40- to 80-ton installations, and not worse than within 15 per cent for installations smaller than 25 tons. *Cooling loads calculated so as to permit inaccuracies as described in this paragraph are not to be used for any purposes except as a basis for cost estimating work.*

Finally when equipment is selected, ducts designed, etc., for an *actual installation*, the cooling load should be calculated as exactly as possible. Remember, the success or failure of the installation depends upon the cooling-load calculations on which the design of the installation is based. Cooperation of the customer must be secured regarding maximum human occupancy, approximate frequency of door openings, use of lights, use of window shades and Venetian blinds, and use of any miscellaneous appliances distributing to cooling load. *U* factors must be determined *accurately*, even though it may be necessary to drill holes through walls, floors, and roofs of existing buildings to be sure of the construction materials.

In the problems at the end of this chapter, we have tried to present those that illustrate the calculation of cooling load in the different degrees of accuracy outlined in the preceding three paragraphs. As in the case of calculation of winter heating loads, the calculation of summer cooling loads is not particularly *difficult*, but it is lengthy and involves considerable work. We have therefore *not* presented 10 *complete* cooling-load calculations. Some of the shorter questions at the end of this chapter are chosen *not to test your knowledge* of the material in this chapter, but instead are selected *to instruct you further* by practical illustrations.

PROBLEMS

Problems 1 to 10 are based upon a building described as follows:

Refer to page 273, which shows the drawings of a building used in the review problems of Chap. XV. Use only the *drawing* of the building; the design data are to be as follows:

a. Location—New York City.

b. Direction of prevailing wind—southwest.

c. Indoor design conditions—80° dry bulb, 67° wet bulb, when the outdoor conditions are at maximums.

d. Outdoor design conditions (see Table 1, page 345, Chap. XIX).

e. Occupancy—150 persons, working as actively as average store clerks.

f. Machinery inside, driven by electric motors also located inside, 500 hp.

g. Walls are medium dark in color; the roof surface is of tar and gravel. Windows are equipped with inside Venetian blinds fully drawn.

h. There is a steam table having 14 sq ft of top surface (used for keeping manufactured articles warm). Six large artificial gas Bunsen burners are used in such a way that approximately 50 per cent of their heat output is sensible heat. Miscellaneous gas-heated equipment, 100 per cent sensible heat, consumes 40 cu ft per hr of artificial gas (Table 11).

i. Note that the thickness of the concrete in the roof of the building is 3 in. This may not be quite clear on your drawing.

1. From your work in Probs. 2 and 3 of Chap. XV, list the heat-transmission coefficients and the areas of the portions of the building subject to ordinary transmission-heat flow. Calculate the ordinary transmission-heat load in Btu per hour, *not including* windows at this time.

2. Determine the infiltration in cubic feet of air per hour. Can you use your infiltration figures from Prob. 4, Chap. XV? Why or why not?

3. Determine the infiltration heat load in Btu per hour of sensible heat and Btu per hour of latent heat.

4. Calculate the solar heat *transmitted into* the building *at* 8 A.M. Show separate figures for masonry of east wall, windows of east wall, masonry of west wall, windows of west wall and south wall. *Do not neglect the effect of time lag in all cases except window glass and doors.*

5. Repeat Prob. 4, for 12 o'clock noon.

6. Repeat Prob. 4 *for* 4 P.M., and tabulate the results of Probs. 4, 5, and 6 in a table. From your table determine at which time the total solar heat transmitted to the inside of the building is at a maximum value.

7. Calculate the sensible and latent heat due to human occupancy.

8. Calculate the sensible and latent heat due to machinery and appliances inside the conditioned space.

9. Assume that the conditioning equipment and ducts are located inside the conditioned space so that there is no transmission-heat flow into the ducts. Tabulate the total of the various components of cooling load showing "total sensible," "total latent," and "total of sensible plus latent."

10. Calculate the reduction in heat load if 2 in. rigid insulation is incorporated in the roof construction.

CHAPTER XX

PSYCHROMETRIC STUDY OF AIR-CONDITIONING EQUIPMENT AND PROCESSES

General. Before studying this chapter you should be thoroughly familiar with psychrometry in general and with psychrometric charts, especially the Trane psychrometric chart. The Trane chart is ideal for analyzing psychrometric changes. In this chapter we shall examine the equipment used to heat and cool air and to humidify and dehumidify air. In each case, changes in temperature and humidity occur simultaneously, and we wish to develop mental pictures of these changes by drawing the processes on psychrometric charts. We want to show the psychrometry of what happens to air when it passes through conditioning units composed of air washers, coils, or a combination of the two. In the next chapter, we shall examine in detail what happens to conditioned air after it arrives in air-conditioned spaces.

Many contractors engaged in air-conditioning work do not thoroughly understand the subject matter presented in this chapter. Instead, they depend upon the sales engineers of the equipment manufacturers to do their work when it is of the type outlined here. It will be possible for you to enter air-conditioning work successfully without a thorough knowledge of the subject matter of this chapter, but you will be in a great deal better position if you understand it thoroughly.

Types of Conditioning Equipment. Conditioning units are factory assembled in the case of small units, while large units are built in the field from parts shipped "knocked down." At this time, we are not interested in the mechanical construction of the conditioner. Whether the unit is made of copper or steel, whether its joints are welded or riveted, whether it does or does not contain air filters and fans—none of these concern us at the present. We are interested in the part of the conditioning unit *that produces the change* in the *psychrometric properties of the air* passing through the unit.

An air washer is *of itself* a complete conditioning unit. Such an air washer is illustrated by the schematic sketch in Fig. 1. Many conditioning units, especially those of the package type, are complete units including filters, fans, motors, and an *extended surface coil.* The extended surface coil, supplied with the proper amount of heating or cooling medium (the medium might be cold water, hot water, steam, or refrigerant if the coil is a direct-expansion evaporator), is the *only* part of some conditioning units that produces changes in the psychrometric properties of the air. Such coils are similar in general construction to automobile radiators.

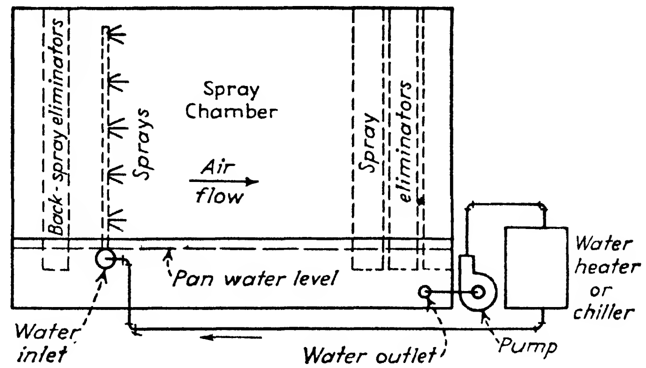


FIG. 1.—Air washer.

The heating or cooling medium flows within the tubes of the coils, and the air flows "through" or "over" the coil just as air is drawn "through" an automobile radiator. Typical extended surface coils are illustrated in Figs. 3 and 4. Figure 2 shows a cutaway section of a spray nozzle.

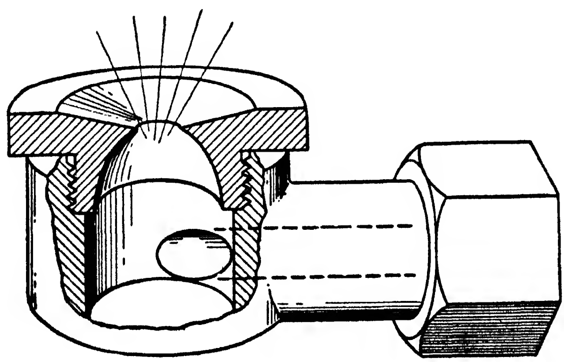


FIG. 2.—Cut-away view of spray nozzle.

The surfaces of an extended surface coil are dry except when wet by moisture that has been condensed out of the air. Hence it *is possible* to pass air over an extended surface coil without changing the dew point of the air. On the other hand, the interior of an air washer is very wet. In fact, the entire inside chamber of an air washer should be *filled* with a spray of small droplets, from the spray nozzles to the eliminator plates.

Therefore, it *is not possible* to pass air through an air washer without changing the dew point of the air.[1] Because of the difference in the characteristics of air washers and coil-type conditioning units, we shall study separately the psychrometry of air passing through these two types of conditioning units. Sometimes extended surface coils are located before or after an air washer, so we shall study this combined type of conditioning unit also.

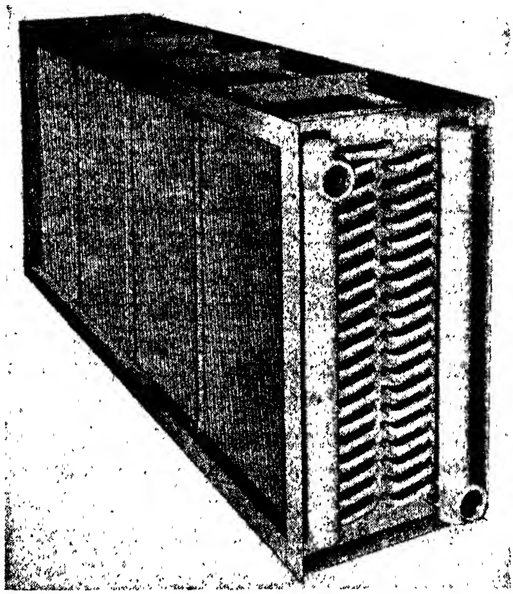


FIG. 3.—Extended surface coil-serpentine refrigerant type O-S. (*The Trane Co.*)

Throughout this chapter make it your object to understand what different types of conditioning units can do in the way of altering the psychrometric conditions of the air entering and passing through the unit. Remember that in either summer or winter air-conditioning work, we ordinarily want conditioned air at *certain definite psychrometric conditions*. In the following pages, we shall study what type of equipment is required to produce air at these different psychrometric conditions. Thus you will have a guide to the *type of conditioning equipment required.* Calculation of the *exact* psychrometric conditions of air leaving any particular conditioning unit should be made only from up-to-date capacity and performance ratings supplied by the manufacturer of the equipment.

[1] There is one exception. If air enters an air washer 100 per cent saturated and if water at just the same temperature as the air is supplied to the air washer, the air will pass through without any change in psychrometric properties. This is a theoretical consideration only and is of no practical significance.

Psychrometry of Air Washers. *With Hot Water Winter Operation.* Air washers are used for both winter and summer air conditioning. In winter air conditioning, the air washer (besides cleaning the air) is ordinarily used to add a great deal of water to the air. In other words, the air washer must produce a considerable increase in the dew point of the air. For this type of operation, there are two possible conditions:

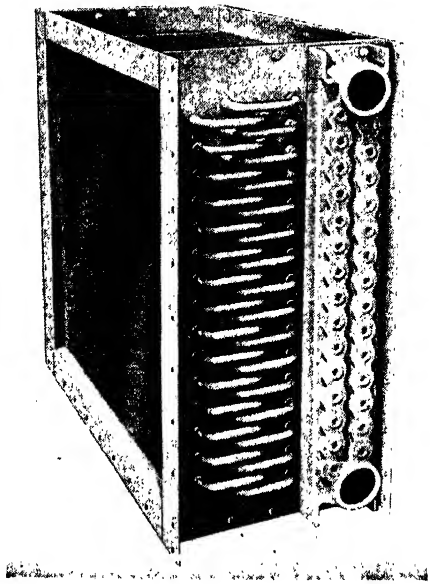


FIG. 4.—Extended surface coil—two row serpentine; type T-S. (*The Trane Co.*)

1. The temperature of the air-washer spray water may be *higher* than the dry-bulb temperature of the air entering the washer (see Fig. 5).
2. The temperature of the air-washer spray water may be slightly lower than the dry-bulb temperature of the air, but higher in temperature than the wet-bulb temperature of the air (see Fig. 6).

In Figs. 5 and 6, point P represents the psychrometric conditions of the air entering the air washer. Lines PXY represent theoretical conditions in Figs. 5 and 6 (and also in Figs. 7 and 8 which we shall discuss shortly). Point Y represents the temperature of the spray water in the air washer, *if* this spray water were maintained at a *constant temperature throughout the spray chamber* (entire inside) of the air washer. This condition is theoretical and cannot exist actually because for the conditions illustrated in Figs. 5 and 6 it is obvious that the spray will be cooled somewhat by the air. For this theoretical condition, therefore, the air would leave the air washer at some point such as X on the straight line PXY. Line PXY

is a *straight* line drawn from point *P*, the condition of the air entering the air washer, and point *Y*, which represents the temperature of the spray water. If the air washer were 100 or 200 ft long, point *X* would be at the same place as point *Y*, but since air washers are ordinarily 6 to 15 ft

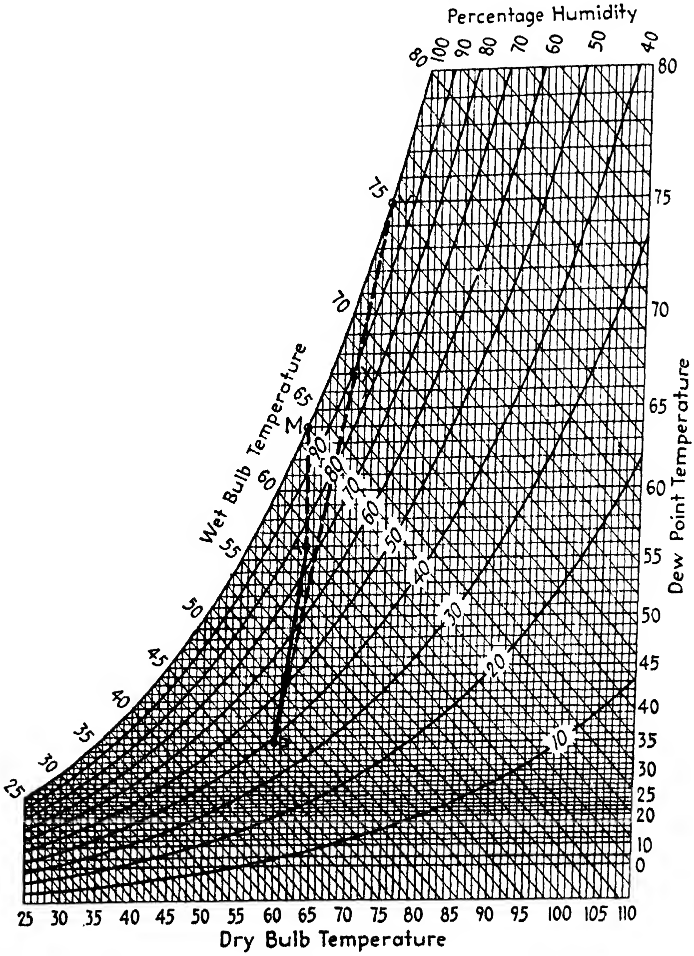


FIG. 5.—The temperature of the air-washer spray water may be higher than the dry-bulb temperature of the air entering the washer. (*The Trane Co.*)

in length, the air does not become fully saturated. Hence point *X* actually lies somewhere between point *P* and point *Y*; the longer the air washer, the nearer point *X* is to point *Y*.

We have just stated that line *PXY* represents theoretical conditions that are impossible to obtain in actual practice. Line *PAM* represents the *actual conditions*, point M being the temperature of the water *leaving* the air washer. In other words, point M, which represents a lower temperature than point Y in Figs. 5 and 6, represents the *average temperature* of the air-washer spray water *as it drops into the pan* after having been

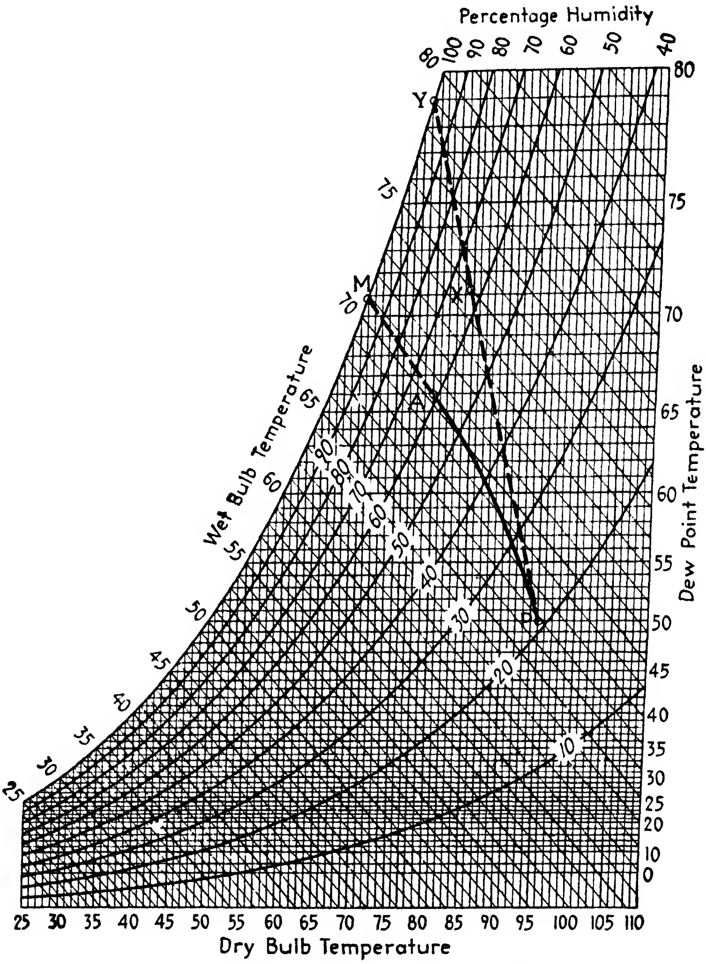


Fig. 6.—The temperature of the air-washer spray water may be slightly lower than the dry-bulb temperature of the air. (*The Trane Co.*)

cooled somewhat by the air. In addition to any sensible cooling effect of the air, the warm water supplied to the sprays must provide the latent heat required to evaporate the water that is evaporated into the air stream. This means that each droplet of spray water will cool in the same manner as the wet-bulb wick of a wet-bulb thermometer. The longer and more efficient the air washer, the nearer point A will be to point M. Curve PAM is *not* a straight line but *is slightly curved,* as indicated in Figs. 5 and 6.

Study carefully Figs. 5 and 6 and the two preceding paragraphs, as these indicate the types of humidifying performance that could be expected from

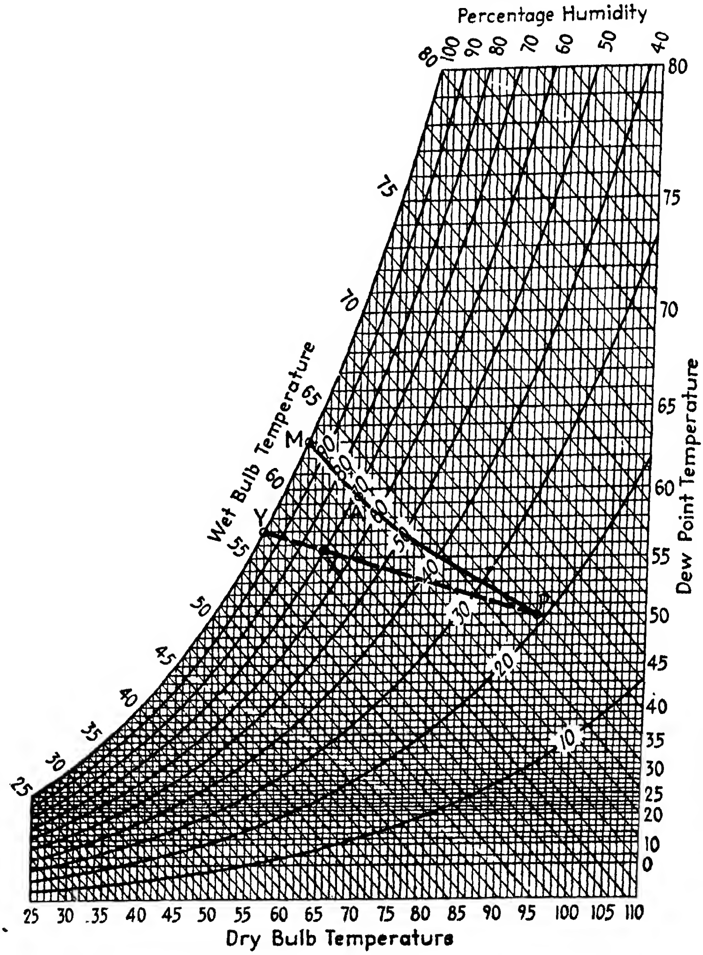


FIG. 7.—Line PXY represents the theoretical conditions of constant spray throughout the spray chamber of the washer. (*The Trane Co.*)

an air washer on a winter air-conditioning installation. Naturally, the water that is removed from the pan of the air washer must be heated in some manner before being pumped back to the sprays of the air washer.

With Cold Water. Summer Operation. Chilled water may be supplied to an air washer that is *cooler* than the *wet-bulb* temperature of the air, but *warmer* than the *dew-point* temperature of the air. In this case, the air will be *reduced in dry-bulb* temperature but will have its *dew point raised.* This operation is illustrated in Fig. 7, where again line *PXY* represents the *theoretical* conditions of constant temperature spray through-

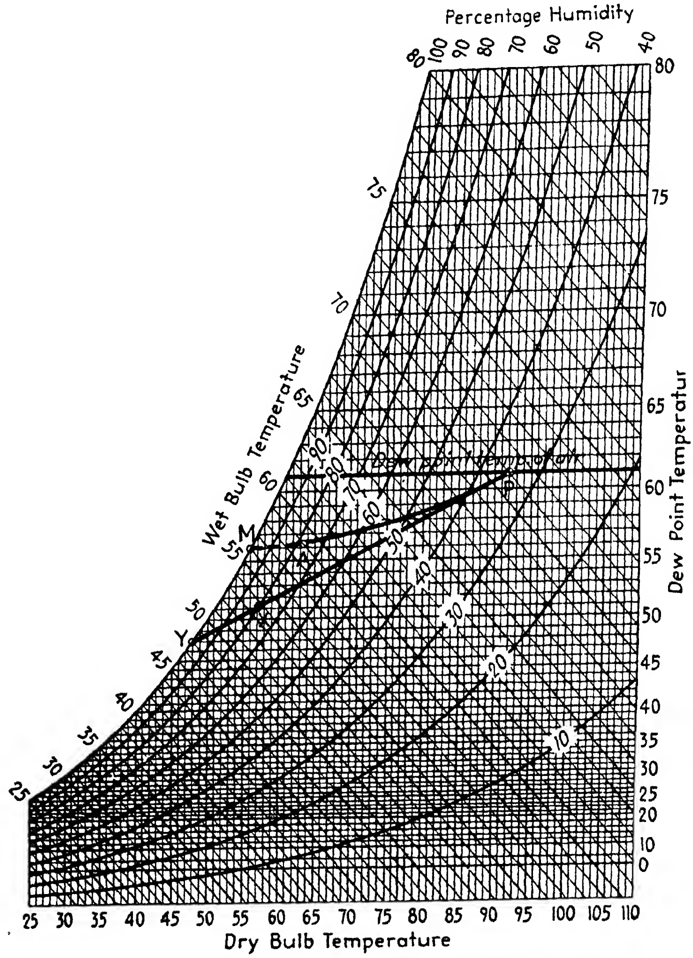


FIG. 8.—*PXY* here represents theoretical conditions for cooling and dehumidifying. (*The Trane Co.*)

out the spray chamber of the washer. Line *PAM* again represents the actual conditions. Point *M* represents the "mean" or average temperature of water *leaving* the air washer. Point *A* represents the psychrometric properties of the air leaving the air washer. Point *P* throughout these illustrations is used to represent the psychrometric conditions entering the air washer.

The most frequent use of an air washer in summer air conditioning is for cooling air at the same time that the dew point of the air is reduced. For this operation, it is necessary that the air washer be supplied with chilled water lower in temperature than the dew point of the air. Figure 8 illustrates this process, with curve *PXY* again representing *theoretical* conditions. If the chilled water enters the washer at the temperature indicated by point *Y*, we know that the water will be warmed somewhat by the air. Therefore, the average temperature of the spray droplets as they drop into the pan of the air washer would be at some warmer temperature, such as that represented by point *M*. The line *PAM* is a *curved line*, and point *A* represents the condition of the air leaving the air washer. *Note the following carefully*:

Point *M* may be brought closer to point *Y* by supplying a large quantity of water to the air washer and by having two or three banks of sprays instead of one. This merely means that the more chilled water with which we deluge the spray chamber of the washer, the less this spray water will warm up before dropping into the pan. If we move point *M* downward, we, of course, move the entire curve *PAM* downward.

As we have previously seen, point *A* may be moved closer to point *M* by using a longer air washer. As a practical limit, we can move point *A* approximately to a maximum 95 per cent relative humidity; even with a one-stage air washer we cannot have point *A* much below 70 per cent relative humidity.

Recirculated Spray Water. Summer or Winter. In Chap. XII we discussed the psychrometry of adiabatic (or constant total heat) cooling of air. Figure 9 illustrates this process on the psychrometric chart. For this type of operation, a pump recirculates the water from the pan of the air washer back to the sprays. A float valve (of the ordinary ball-float tank-valve type) admits city water to make up for the water that is evaporated into the air. Since the spray water is not piped away from the washer to be heated or cooled, no heat is removed from air or added to the air passing through the washer. Instead, the air *loses sensible* heat but *gains* this *same amount* of heat as *latent heat.* In other words, we may say that the *dry-bulb* temperature of the air is *reduced*, the *dew point increased*, but the *wet-bulb* temperature *remains constant.*

In Fig. 9, line *PAM* represents the air-washer process for winter operation. Outdoor air at 25°F dry bulb and 10° dew point is heated at constant dew point until the dry-bulb temperature becomes, say, 102°, as illustrated by point *P*. This air enters the air washer at the conditions shown as point *P* and is humidified to some point *A* on line *PAM*. Note that the line *PAM* is a constant wet-bulb line on the chart in Fig. 9. (The nearness of point A to point M will depend upon the length of the air washer, point A being nearest to point M in the case of very long air washers. Line $P'A'M'$ represents summer adiabatic cooling of air from 100 to 70° dry bulb. This is frequently done in the American Southwest

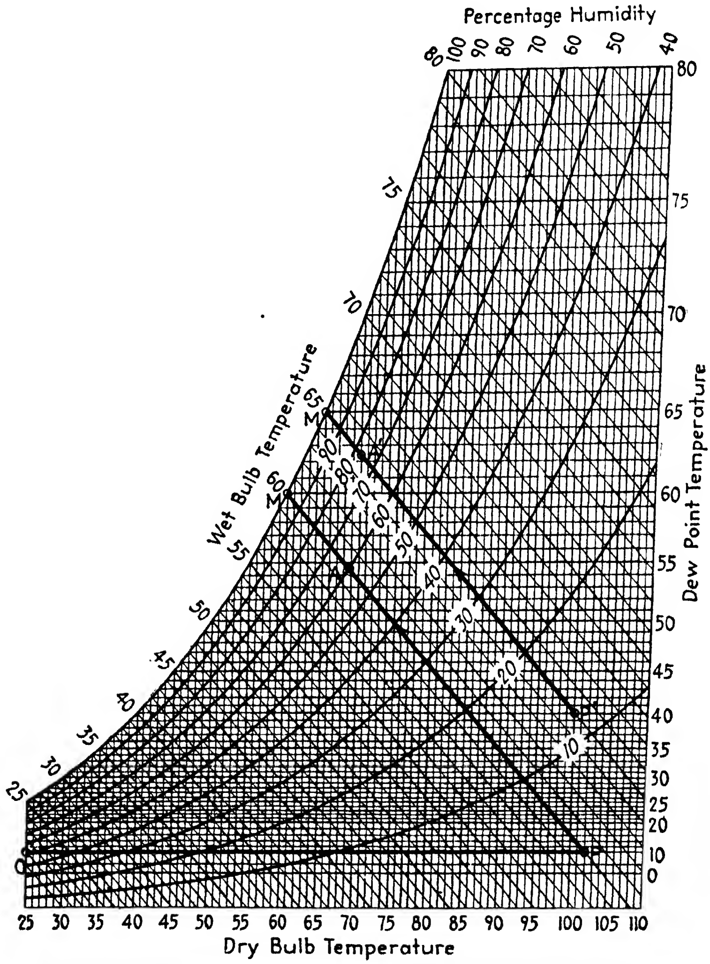


FIG. 9.—Process showing adiabatic cooling. (*The Trane Co.*)

where the dry-bulb temperatures are high and the dew points low. Point P' would represent the outdoor air that enters the air washer without any change. The process illustrated by line $P'A'M'$ is that which we discussed in Chap. XII as evaporative cooling or adiabatic cooling.

Extended Surface Coils. *Heating.* Figure 10a illustrates a side view of a heating coil with air flowing through the coil from left to right. We may assume that the coil is being heated by either hot water or steam. Since the surface of this coil is dry, the air passes through it without any change in dew point. The broken lines extending from left to right illustrate this constant dew point and the increase in dry bulb and wet bulb as the air passes over the hot surface of the coil. This process is illustrated in the psychrometric chart in Fig. 10*b*.

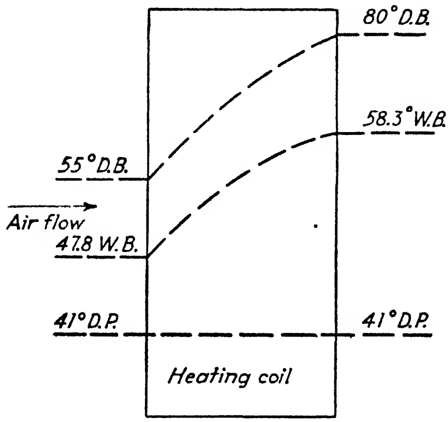


FIG. 10*a*.—Showing process of sensible heating alone.

The heating process illustrated in Fig. 10*b* consists of the addition of *sensible* heat only to the air. Heating coils are used in this manner whenever air is heated by steam or hot water in air-conditioning work. All small winter air-conditioning units contain coils that operate in this manner.[1] All the heating coils illustrated as well as the air washers described later in this chapter are used in the manner illustrated in Fig. 10*a*. This represents a simple but important psychrometric process.

Cooling. Provided the chilled water supplied to an extended surface coil is *not lower* in temperature than the dew point of the air passing over the coil, *sensible heat only* will be removed from the coil. The process is then the reverse of that illustrated in Fig. 10*b*. This process is illustrated in Fig. 11. In both cases, the line on the psychrometric chart is simply a horizontal line or constant dew-point line.

[1] Some small winter air-conditioning units contain humidifying spray nozzles that send fine sprays of water on to the hot surfaces, this water is then evaporated into the air for purposes of humidification. We do not illustrate this as a psychrometric process because such humidification varies greatly and is not a matter of precisely controlled humidification. Ordinarily such humidifying devices humidify at a rate greater than is required; they then shut off completely when the conditioned space is adequately humidified.

At this point in our discussion of extended surface coils, we shall take up the proper method of flow of the cooling or heating medium flowing within the tubes of such coils. The method of flow known as "counter-

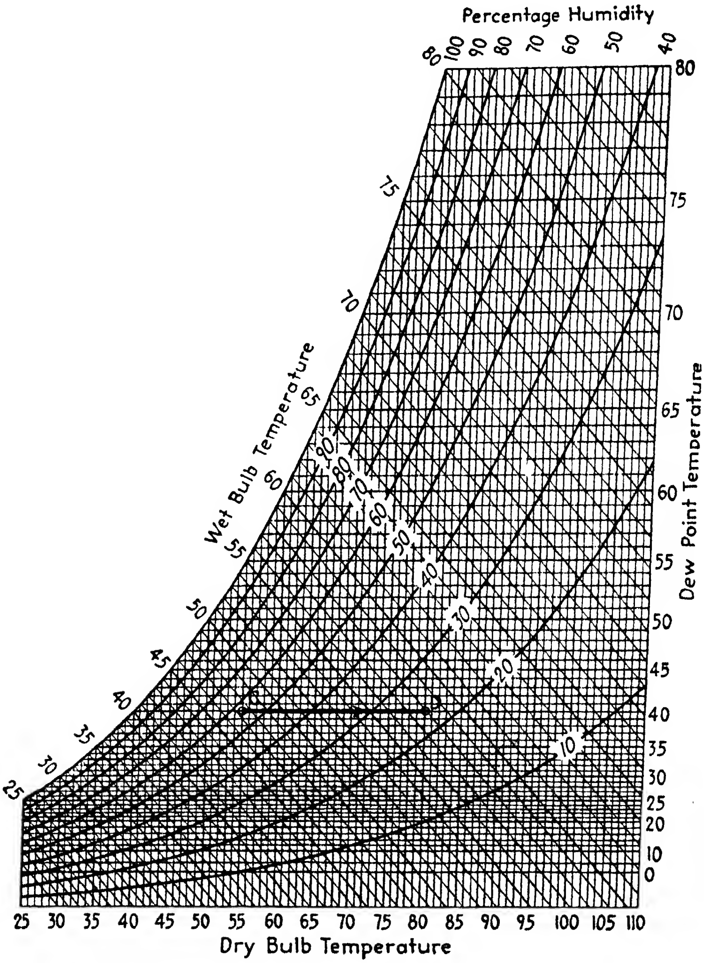


FIG. 10*b*.—Sensible heating alone is shown as a horizontal line. (*The Trane Co.*)

current" flow is always used because it is more efficient. In air conditioning, *countercurrent flow* is of great importance in the case of cooling coils, where the coils use chilled water or refrigerant within the tubes. Figure 12*a* illustrates countercurrent flow for a coil that is cooling according to the process illustrated in Fig. 11. Figure 12*b* illustrates a coil supplied with hot water for a heating process of the type illustrated in Fig. 10. In Fig. 12, the numbers 1, 2, 3, etc., indicate the *number of rows of tubes in* the coil. We always count the number of rows of "depth" of a coil in the *direction of the air flow.* Thus the coil in Fig. 12*a* has six rows of tubes,

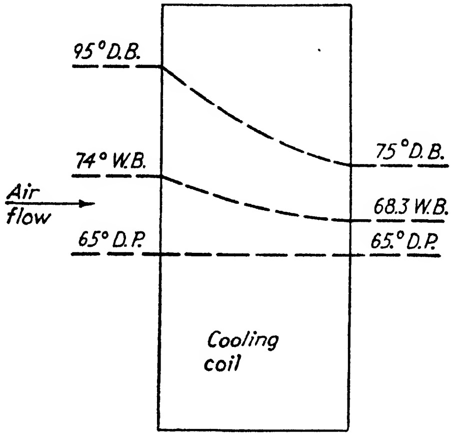


FIG. 11*a*.—Showing process of sensible cooling alone.

while the coil in Fig. 12*b* has four rows of tubes. Letters represent certain temperatures in Fig. 12, their meaning being as follows:

T_1 = dry-bulb temperature of air before passing over the coil
t_2 = dry-bulb temperature of air leaving coil, after being heated or cooled
t_i = temperature of water entering coil (inlet-water temperature)
t_o = temperature of water leaving coil (outlet-water temperature)

The *water enters* the coil at the face of the coil *from which the air leaves.* This means, in the case of Fig. 12*a*, that the coldest water is in the coil tubes at the point where the cool air is leaving the coil (tube 6). Similarly, in Fig. 12*a*, the *warmest water* (at a temperature t_o) is inside the tubes of *row* 1 where the warmest air is passing over the coil. Since this warmest air is at a temperature T_1 that may be 15 to 20° warmer than T_2, it can be successfully cooled by the warm water leaving the coil. On the other hand, the air passing tube row 6 *has already been cooled* a good deal. By means of the countercurrent-flow arrangement illustrated, the following temperature values in Fig. 12*a* would be possible:

$$T_1 = 95°, \quad T_2 = 70°, \quad t_i = 60°, \quad t_o = 72°$$

Make a sketch similar to Fig. 12*a*, but reverse the direction of the water flow. In your sketch you will have water *leaving tube row* 6 at a temperature of 72°F; hence you can see that with this arrangement with the water flow the reverse of that shown in Fig. 12*a*, it would be *impossible to cool the air to a final temperature* T_2 *of* 70°.

In ratings and capacities of extended surface coils, it is necessary to use

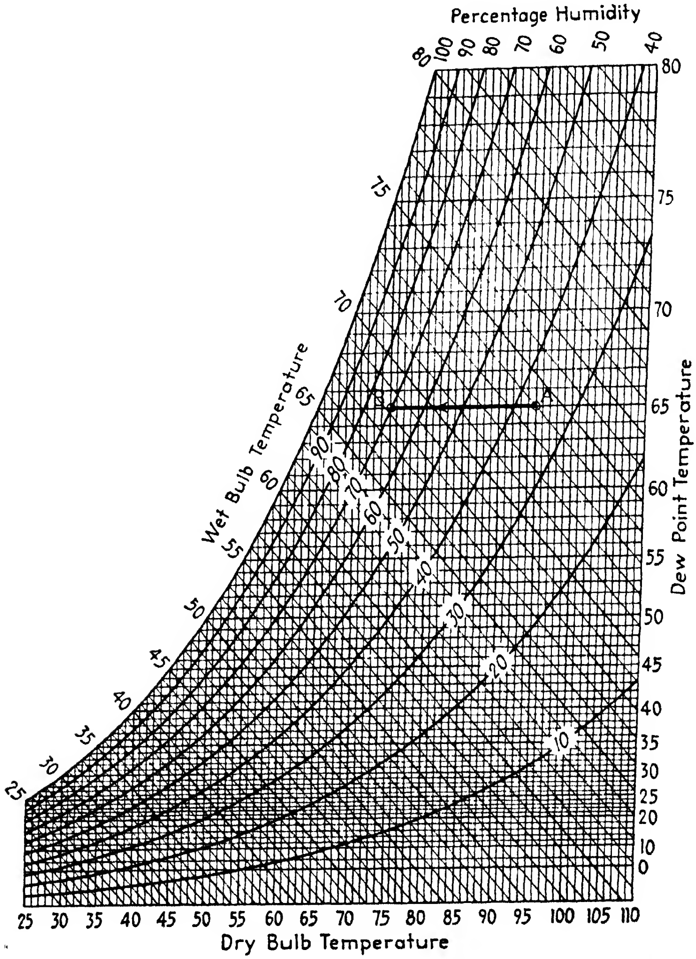


Fig. 11*b*.—Sensible cooling alone is shown as a horizontal line. (*The Trane Co.*)

the term "logarithmic mean temperature difference," abbreviated LMTD. Do not let this term disturb you as it may always be determined easily from tables in data given in manufacturers' catalogues. Each manufacturer has his own method to calculate capacities of extended surface coils.

Cooling and Dehumidifying. When coils are used for cooling *and dehumidifying,* the change in the properties of the air are as illustrated in the sketch in Fig. 13*a*. All that has already been said regarding logarithmic mean temperature difference (LMTD) applies in the case of coils used in this manner. In order that a cooling coil may dehumidify as well as cool, the *inlet water temperature* t_i must be *below* the *dew-point temperature* of the air entering the coil. Curve *PAM* in Fig. 13*b* compares

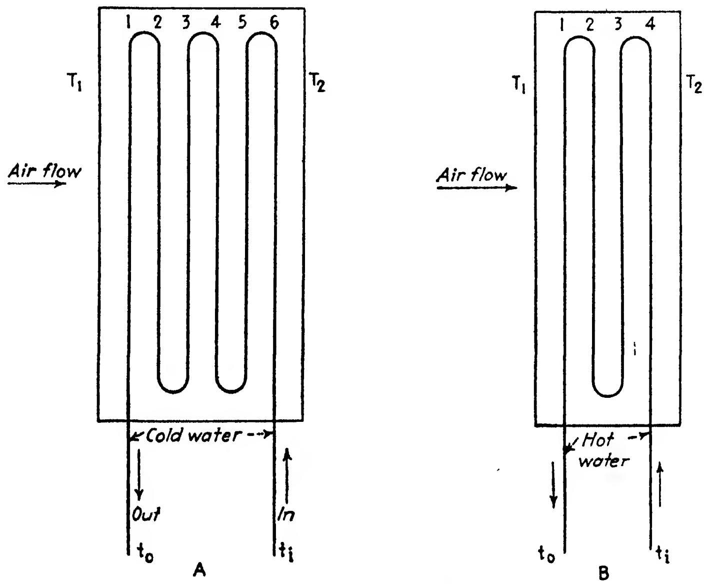


FIG. 12*a*.—Countercurrent flow during sensible cooling alone.
FIG. 12*b*.—Countercurrent flow during sensible heating alone.

with curve *PAM* in Fig. 8. Turn back and note this curve in Fig. 8 and you will see that its *direction of* curvature is *upward,* looking from point *P* toward point *M*. In Fig. 13*b*, on the other hand, the *direction of* curvature is decidedly *downward.* The reason is that Fig. 8 represents cooling and dehumidifying *by an air washer.* The spray droplets encountered by the air passing through the air washer become progressively *warmer* because they are being warmed by the air. With a coil using the *countercurrent* flow we have discussed, the metal surfaces of the coil from left to right in Fig. 13*a become colder and colder.* In other words, the air passing through the extended surface coils in Fig. 13*a* passes over surfaces that are *colder as the air moves further through the coil. This is true only because we are using countercurrent flow.* If we do not use countercurrent flow, curve *PAM* in Fig. 13*b* would be the same shape as in Fig. 8.

Point *M* in Fig. 13*b* represents the same *temperature* (t_i) of the *inlet water* entering the cooling coil. The greater the coil depth, *i.e.*, the more rows of coil used, the nearer point A will be to point M. Theoretically, if we use a coil 50 or 100 rows deep, point A would coincide with point M. In practice, coils deeper than 12 to 16 rows are rarely used, six-row coils being perhaps most common.

The determination of point A when using a coil to cool and dehumidify in a summer air-conditioning installation is of great importance. We cannot say that point A will be located at a certain point for a coil a certain number of rows deep. The location of point A depends upon the manufacturer of the cooling coil, and *different types* of cooling coils may vary greatly even though they are products of the same manufacturer. Most manufacturers make at least three *different types* of coils for cooling and dehumidifying service. Therefore, it is extremely important that you determine the amount of air you wish to pass over the coil-type cooling and dehumidifying unit. Then determine the temperature and the amount of the cooling medium available. Next select a coil from the manufacturers' rating data. Finally, and *before you place the order* for the coil or unit, obtain a statement in writing from the coil manufacturer, guaranteeing the "leaving air conditions" (or point A) for the *particular coil you are buying* and for *your particular operating conditions*.

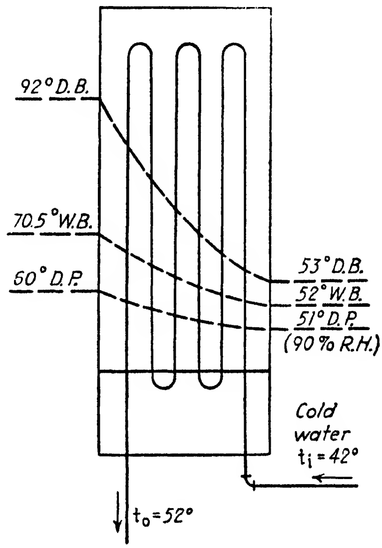


FIG. 13*a*.—Process showing cooling and dehumidifying through cold-water coil.

Central-heating Installations. Air Washer Combined with Heating Coil. When air-conditioning equipment is "built in" the building in the form of a large unit, whether for winter service only or for all-year-round service, the winter operating cycle frequently involves an air washer combined with several banks (or sets) of heating coils. Since the introduction of cold outdoor winter air, which is frequently below 32°F, would endanger the air washer because of freezing, it is necessary to heat this air before introducing it to the air washer. In localities such as New York City, where the outdoor winter conditions may vary from 60 to 0° dry bulb, we ordinarily do not try to achieve this heating by one single steam coil fed by a single steam pipe. With such a single steam coil there is danger that as the steam supply is reduced, under automatic temperature control, condensate may freeze in the bottom rows of the coil. A steam coil to be exposed to 0° air ordinarily must have the steam either *fully turned on* or *fully shut off* to avoid the risk of freezing the condensate and the consequent bursting of the coil.

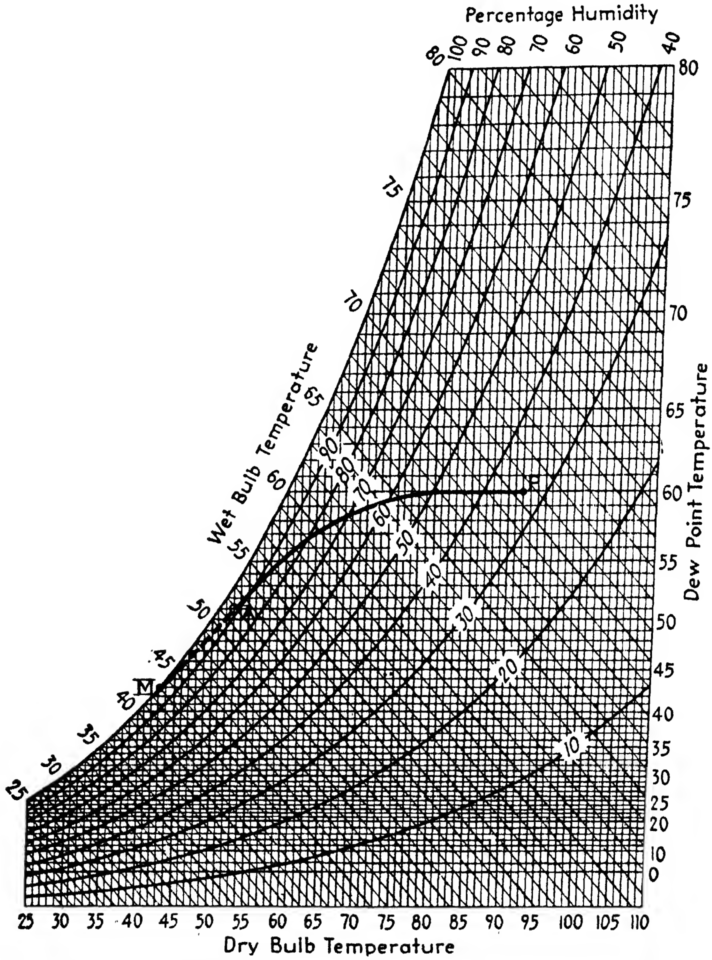


FIG. 13*b*.—Process on psychrometric chart illustrating temperature gradients in Fig. 13*a*. (*The Trane Co.*)

Customary practice therefore is to have the cold outdoor air enter over a steam-heating coil (usually a one-row coil) known as a "*preheating* coil." If the outdoor air is below freezing temperature, the steam valve to this preheating coil is open wide; if the outdoor temperature is above 35°F, the steam valve to the preheating coil is tightly closed. Preheating coils are usually chosen of a size such that they can heat air from 0° to 35 or 40°F. Therefore, a second steam coil is located between the preheating coil and the air washer. This second steam coil is known as a "tempering coil."

The tempering coil usually has a capacity to heat the air from 35°F to anywhere from 75 to 120°F. The steam valve supplying the tempering coil is capable of more delicate and gradual control than simply "full on" and "shutoff." The warm air, which is still at the same low dew point as outdoor air, enters the air washer and passes through the spray chamber thereof. Within the air washer, we may have two different sets of conditions:

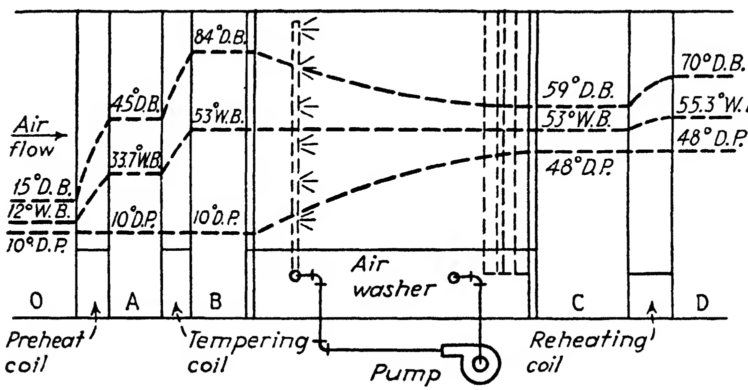


FIG. 14.—Adiabatic cooling preceded by heating and followed by reheating.

1. The air washer may be adiabatic, *i.e.*, a washer recirculating the water from the pan back to the sprays without heating the water. In this event, the process within the air washer itself is shown by the line *PA* in Fig. 9 (see Figs. 14 and 14*a*).

2. The air washer water may be withdrawn from the pan and pumped through a water-heating device and returned to the sprays at a temperature higher than the dry-bulb temperature of the air entering the air washer. In this event, the air washer is *not* adiabatic; the wet-bulb temperature of the air increases as the air passes through the air washer (see Figs. 15 and 15*a*).

The conditions described in 1 are illustrated in detail in Figs. 14 and 14*a*. The heavy broken line extending from left to right in Fig. 14 illustrates the change in dry bulb, wet bulb, and dew point of the air as it passes through the composite conditioning unit. Let us study Figs. 14 and 14*a*. The outdoor air enters the composite conditioning unit at the extreme left in Fig. 14. This point, representing the outdoor air, is marked with letter O at the bottom of Fig. 14. The conditions at point O cannot be shown on the psychrometric chart in Fig. 14a because the chart only goes down to 25° dry bulb.

The preheating coil heats the air at constant dew point, the new con-

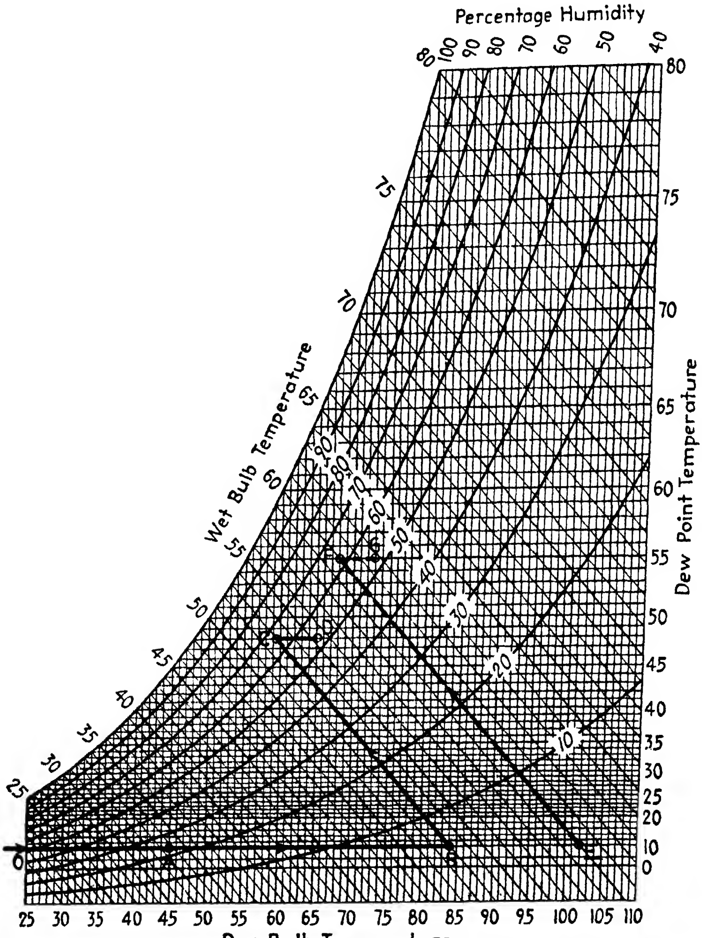


Fig. 14a.—Psychrometric chart shows process indicated on Fig. 14. (*The Trane Co.*)

ditions of the air leaving the preheating coil being shown by point A. Find point A for yourself, and note the conditions both in the diagram of Fig. 14 and on the psychrometric chart in Fig. 14a. The tempering coil adds still more sensible heat to the air, raising it to the conditions indicated by the letter B. The air at the conditions represented by B enters the air washer. Find *B* both in Figs. 14 and 14*a*. At *B* the air is at 84° dry bulb, 53° wet bulb, and 10° dew point.

As the air passes through the air washer, its dry-bulb temperature decreases, its dew point increases, and its wet-bulb temperature remains constant. This is illustrated clearly in Fig. 14. On the psychrometric chart in Fig. 14*a*, the air-washer process is illustrated by the line *BC*. The air leaves the air washer at the conditions represented by point *C*, which are 59° dry bulb, 53° wet bulb, 48° dew point. (In the event that we desire the air leaving the air washer to be at a higher dry-bulb temper-

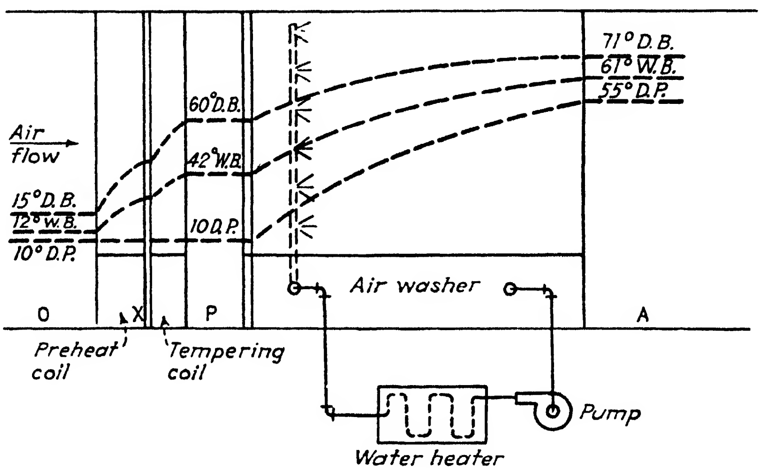


FIG. 15.—Nonadiabatic air washer.

ture and higher dew point, we would increase the heating effect of the tempering coil and line *BC* on the psychrometric chart would move to some new position as indicated by line *EF*.)

Since air at 59° wet bulb would be too low for introduction directly to an air-conditioned space in wintertime, a *reheating* coil would be located beyond this type of air washer, as indicated in Fig. 14. The reheating coil would add sensible heat only to the air, the process being that illustrated by line *CD* on the chart in Fig. 14*a*. The air finally delivered to the duct system to be carried to the conditioned space would be that shown by point *D*. The properties of this conditioned air at point *D* are 70° dry bulb, 55.3° wet bulb, 48° dew point.

Certain psychrometric conditions are cited in the explanation of Fig. 14 in the preceding paragraphs and in the explanation to follow regarding Fig. 15. *These figures simply represent typical possible conditions. Naturally these conditions vary depending upon the installation.*

In some cases it is desired to use the air-washer arrangement mentioned in alternative 2. In this event, it is sometimes possible to dispense with the reheating coil, although even in this case a reheating coil makes a more ideal installation. Figure 15 illustrates an air washer using water heated to an average temperature, for the spray droplets falling into the pan, of

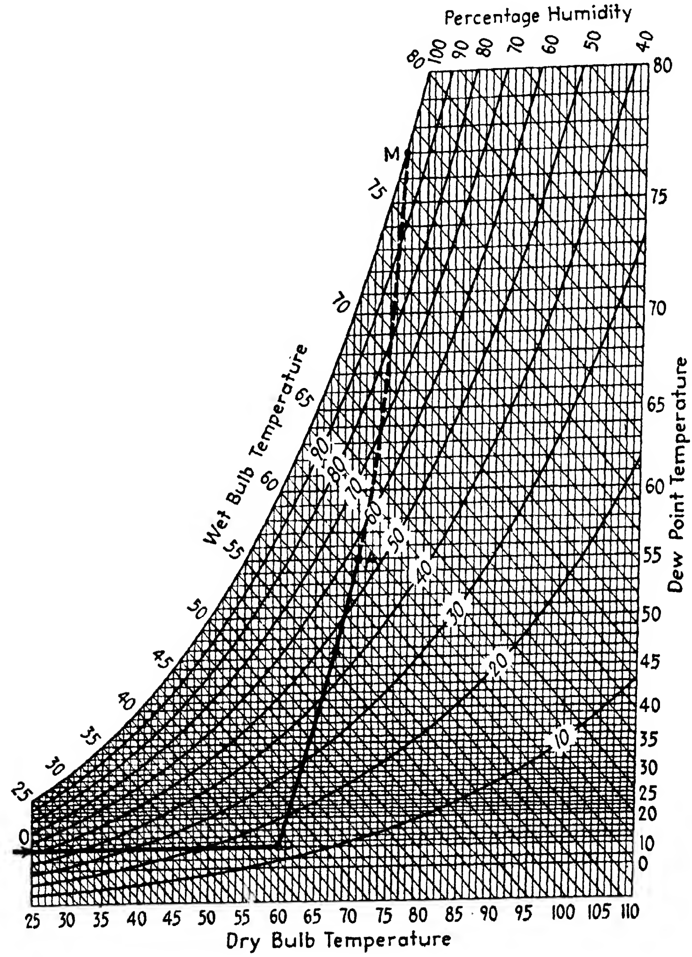


FIG. 15a.—Nonadiabatic air-washer process as shown on psychrometric chart. (*The Trane Co.*)

77°F, which is considerably higher than the dry-bulb temperature of the air entering the air washer. (The water leaving the sprays of the air washer is cooled to this temperature of 77°F because the water gives up heat to increase the latent and sensible heat of the air, therefore the water is supplied to the sprays at a temperature 8 to 20° warmer than 77°F.)

You will recall that in Fig. 5 we studied what would happen in an air washer of this type. Let us examine the composite conditioning unit of Fig. 15, along with the psychrometric chart illustrating it, Fig. 15*a*. Note that the preheating coil and the tempering coil in Fig. 15 perform in the same general manner as those which we have seen in Fig. 14. Outdoor air is heated to a point *P* by the preheating coil and the tempering coil. The condition of the air at point *P* is at 60° dry bulb, 42° wet bulb, 10° dew point. This air enters the air washer and is increased in all three properties illustrated. The heavy broken lines in Fig. 15 illustrate how the dew-point, wet-bulb, and dry-bulb temperature of the air increases as it passes through the air washer. This process is shown by line *PA* on the chart in Fig. 15*a*. If you compare line *PA* of Fig. 15*a* with line *PA* in Fig. 5, you will see that the process is identical. In Fig. 15*a*, point *M* represents the average temperature of the sprayed droplets in the air washer as they fall into the pan. From the installation illustrated in Fig. 15, we then have conditioned air as represented by point *A*, which is 7° dry bulb, 61° wet bulb, 55° dew point.

Chemical Dehumidification of Air. There are two general types of chemical dehumidification equipment. One general type uses a chemical in a *solid state* which has the ability to extract water vapor from air. With units of this type, a double installation of dehumidification equipment is required because the chemical rapidly becomes completely saturated with water vapor, at which time it must be "reactivated" or "regenerated." The air to be dehumidified passes through one of the pair of units while the other unit is being regenerated. The moisture is driven off from the solid-state chemical drying agent by means of heat from gas flames or from high-pressure high-temperature steam-heated coils. The dual unit operates in this manner for 7 to 12 min, whereupon automatic dampers send the air to the newly regenerated unit and the first unit commences its regenerative cycle.

There are several chemical dehumidifying units available using *liquids* as the air-drying agents. In this case, the dual unit is not necessary, as a portion of the liquid may be continuously pumped to a small unit where the water is driven out of the liquid by means of heat. Both the liquid and the solid types of chemical dehumidification units are in all cases (except one) of the adiabatic type. This means that the wet-bulb temperature of the air is constant while the dew point is being decreased. This process is represented on the psychrometric chart as a constant wet-bulb line; it is the same as that of an adiabatic air-washer process except that the dew point is *decreasing* instead of increasing. Line *PB* on the psychrometric chart in Fig. 16*a* illustrates adiabatic chemical dehumidification on the Trane psychrometric chart. Note that the air increases in dry-bulb temperature from 80 to 101°. This dry 101° air is of no value for summer air conditioning until it has been passed over a sensible cooling coil to reduce its dry-bulb temperature to about 60°F. Figure 16 illustrates chemical dehumidification with an "after cooling coil," the coil cooling the air at constant dew point. Line *BC* on the psychrometric chart in Fig. 16*a* illustrates the sensible cooling operation of this type of conditioning unit.

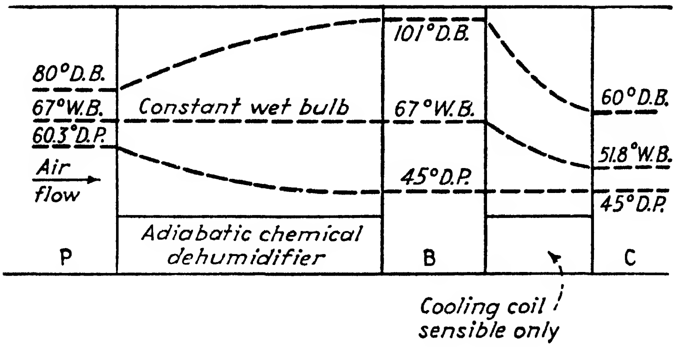


FIG. 16.—Chemical dehumidification and sensible cooling.

The one exception mentioned above, *i.e.*, a chemical dehumidification unit that is *not adiabatic*, is the "calorider" liquid type of chemical dehumidification manufactured by the Research Corporation, New York City. In a calorider-type dehumidifier, extended surface cooling coils are located *within* the dehumidifier itself, so that the air is cooled sensibly *at the same time* that its dew point is decreased. This results in certain advantages as to control and eliminates the need for an after cooling coil. The extended surface coil within a calorider may be cooled by any source of cooling water available. The air leaves a Research calorider at a dry-bulb temperature of 5 to 10° above the temperature of the cooling water supplied to the calorider, depending upon the unit selected. By a proper choice of the liquid dehumidifying agent, the air may leave the calorider at any relative humidity desired, down to as low as 15 per cent. In Figure 16*a*, line *PC* illustrates how a Research calorider would dehumidify air from point *B* to point *C*, using cooling water at a temperature of 55°F. If desired, this same calorider could use a different brine and bring the air down to point *X*, as indicated by the line *PX*.

Study the psychrometric chart in Fig. 16*a* carefully. Line *PAM* represents the cooling and dehumidifying of air by using *chilled water at a temperature of* 53°F *in an extended surface coil,* which we illustrated previously in Figs. 13 and 13*a*. The conditioned air leaving this coil would be at a point approximately like point *A*, in Fig. 16*a*, if chilled water at 53°F were used. Chemical dehumidification equipment (using an after cooling coil if the chemical dryer were adiabatic) could bring this air from point *P* down to point *O* using the same 53° cooling water. By means of a Research calorider, this could be brought down to point *X*. A larger adiabatic unit with deeper after cooling coils could also bring air from point

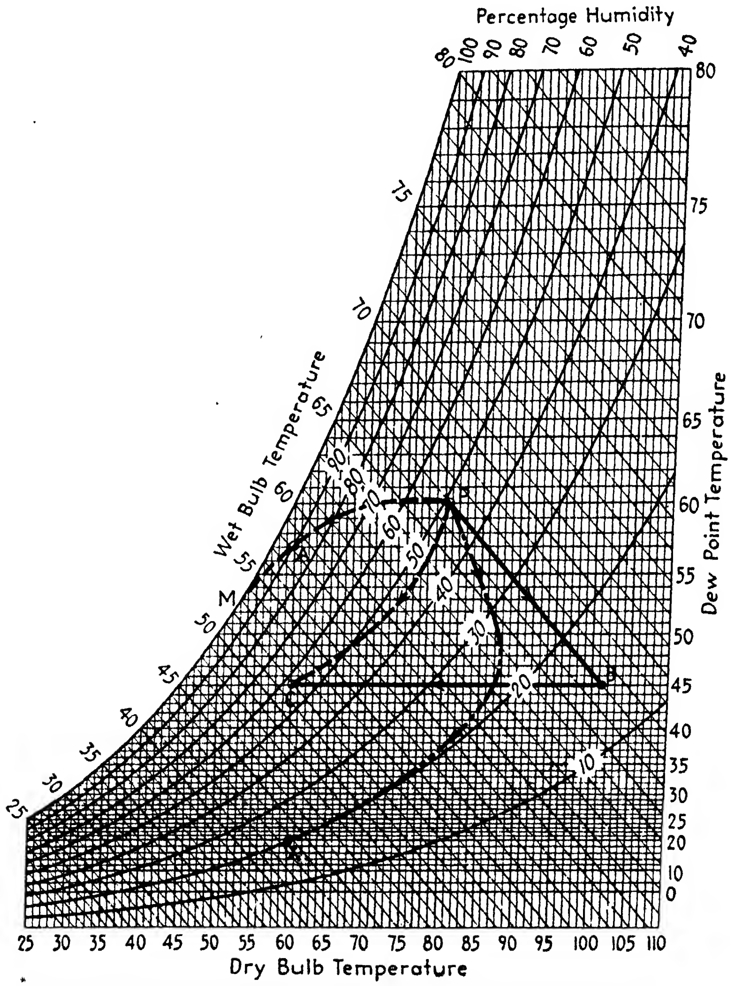


FIG. 16*a*.—Chemical dehumidification as shown on psychrometric chart. (*The Trane Co.*)

P to point *X*, *still using* 53° *cooling water*. The important fact is that, with 53° cooling water, process *PA* gives you air at a dew point of about 57°F and at a dry-bulb temperature of 60°F. Chemical dehumidifying equipment, although more complicated and somewhat more expensive to buy, could give conditioned air at this same dry-bulb temperature of 60°F *and at any dew point desired down to* 20°F, or even lower. Thus you see that chemical-dehumidification equipment could deliver conditioned air *at a dew point many degrees lower than the temperature of the chilled water* available.

Solving Air-mixture Problems. Trane Chart. Let us suppose that you have 1,000 cfm of air at 94° dry bulb, 74° wet bulb which is mixed with 1,000 cfm of air at 65° dry bulb, 54° wet bulb. If each of these two streams of air is measured in its respective duct at its actual condition, find the following:

1. The pounds per minute of air at the higher conditions and at the lower conditions.
2. The psychrometric conditions, including specific volume of the mixture produced by mixing the two ordinary streams of air.

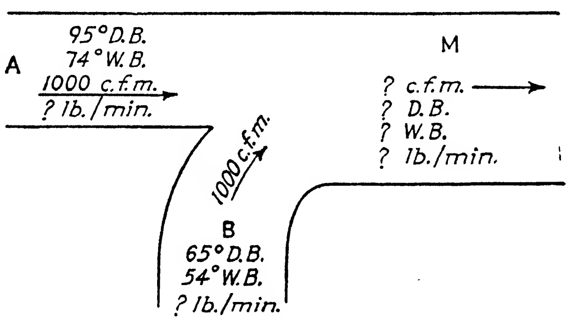


FIG. 17.—Mixture of two air streams.

3. The pounds per minute of air and the number of cubic feet per minute of air measured at the new conditions of the mixture of the two streams.

Problems such as this frequently occur in air conditioning. Almost every air-conditioning installation involves at least one problem in which two ducts meet carrying streams of air having different psychrometric properties. In all such cases, it is necessary for us to be able to determine the psychrometric conditions and the number of cubic feet per minute of the mixture which is found in the duct after the two smaller ducts have joined.

Figure 17 illustrates two ducts joining, giving us the conditions of the problem stated in the preceding paragraph. From Fig. 17, although the cubic feet per minute figures are deliberately chosen as simple ones you can see the importance of being able to calculate air-mixture problems. You already know how to calculate such problems, and you would proceed as outlined in the steps below. We admit that the method is lengthy and difficult, and we shall show you a much simpler and quicker method.

1. Find at points *A* and *B* (Fig. 17*a*) the specific volume, total heat absolute humidity, sensible heat, and pounds per minute.

2. Determine the sensible heat content of the air at *A* and at *B* by

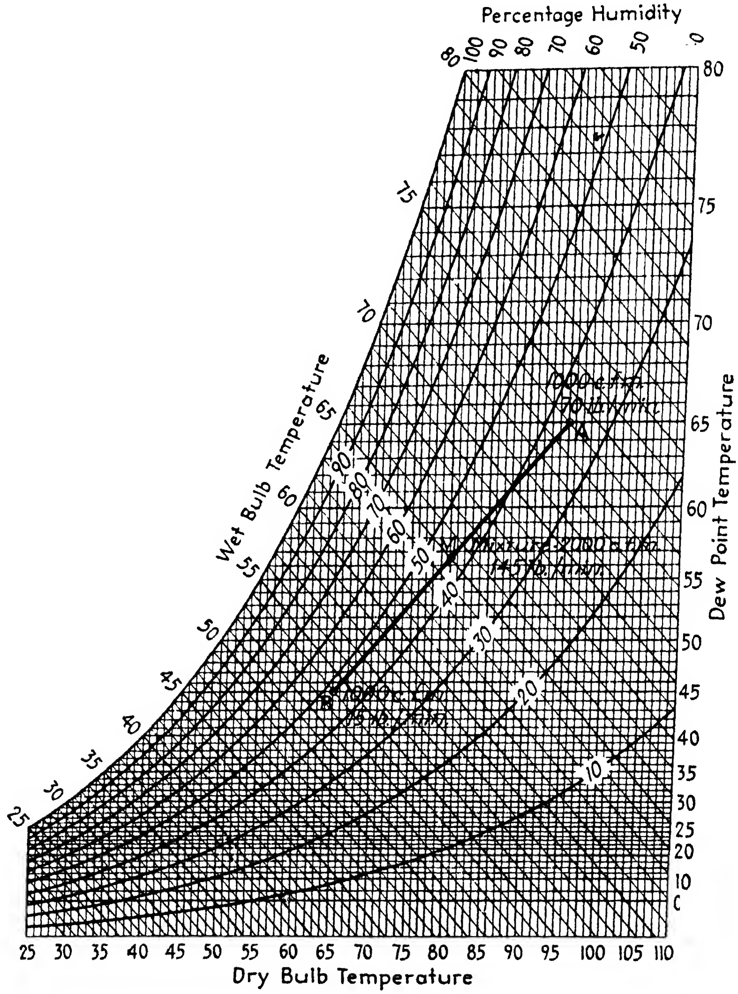


FIG. 17*a*.—Psychrometric chart showing mixing of two air streams. (*The Trane Co.*)

multiplying the sensible heat in Btu per pound by the number of pounds per minute.

3. Add the two quantities of sensible heat obtained in 2 and divide by the *total* number of pounds per minute *of mixture*, thus obtaining the sensible heat content in Btu per pound of this mixture.

4. From the sensible heat content of Btu per pound as determined in 3,

find from the Psychrometric Tables the dry-bulb temperature of the mixture.

5. Find the amount of absolute humidity present in the air at A and at B by multiplying the pounds per minute of air by the absolute humidity in grains per pound.

6. Divide the total absolute humidity computed in 5 by the total number of pounds per minute of air in the mixture, thus obtaining the absolute humidity *of the mixture* in grains per pound.

7. From the absolute humidity of the mixture as computed in 6, determine the dew point of the mixture.

8. In 4 you found the dry-bulb temperature of the mixture. In 7 you found the dew-point temperature of the mixture. This information would enable you to locate the mixture point M on the psychrometric chart. You can then determine the specific volume of the mixture and hence finally obtain the number of cubic feet per minute that you would measure in the duct systems of Fig. 17 if you use air-measuring instruments at point M.

You should be able to follow through the calculations described above from your knowledge of psychrometry in Chaps. VII and XVIII. Specific volume should now be found from the Trane volume chart, as described in Fig. 14*a*, Chap. XVIII, because this method is so much simpler than using the Psychrometric Tables. Below we present a solution of the problem in Fig. 17*a* and described above. You may skip this solution if you wish because we are giving it to you for two reasons: (1) to show you the exact and correct solution of the problem, and (2) to show you how difficult it is to obtain these exact answers by this method.

Solution to Problem of Fig. 17*a*. Computation Method. In the solution below, each step is marked with a number in parentheses corresponding to the numbered steps listed above. From now on, do not use the Psychrometric Tables in the Appendix, but use the new psychrometric table given on page 403.

(1) From Trane volume chart (see Fig. 18):

Specific volume at A = 14.27 cu ft per lb
Specific volume at B = 13.35 cu ft per lb

From the table, page 403:

Sensible heat at A (95°) = 22.95 Btu per lb
Sensible heat at B (65°) = 15.69 Btu per lb
Absolute humidity at A (65° D.P.) = 92.6 grains per lb
Absolute humidity at B (45° D.P.) = 44.2 grains per lb

(2) Pounds per minute at $A = \dfrac{1{,}000}{14.27} = 70$ lb per min

$$\text{Pounds per minute at } B = \frac{1{,}000}{13.35} = 75 \text{ lb per min}$$

$$\text{Total pounds per minute at } M = 75 + 70 = 145 \text{ lb per min}$$

(3) Total sensible heat at A = 70 (22.95) = 1608 Btu per min
Total sensible heat at B = 75 (15.69) = 1177 Btu per min
Total sensible heat at M = the sum = 2785 Btu per min

(4) $\dfrac{2785}{145} = 19.2$ Btu per lb

From page 403, 19.2 Btu per lb *sensible* heat corresponds to (by interpolation) a dry-bulb temperature of 79.5°F.

(5) and (6) See steps 1 and 2, then

Total absolute humidity at A = 70 (92.6) = 6,480 grains per min
Total absolute humidity at B = 75 (44.2) = 3,315 grains per min
Total grains per pound at M = the sum = 9,795 grains per min

$\dfrac{9{,}795}{145} = 67.5$ grains per lb absolute humidity at M.

(7) From the table, 67.5 grains per lb corresponds to a dew point of 56.3°F, by interpolation in column (3).

(8) Locate air at point M on a Trane chart at 79.5° *dry bulb* and 56.3° *dew point.* From the Trane volume chart the specific volume at M is 13.81 *cu ft per lb.* Hence

$$13.81\ (145) = 2{,}003\ \textit{cfm} \text{ at point } M$$

Even if you have merely looked at the solution presented above without studying it, you will agree that it is too lengthy to use for the solution of problems that occur as often as air-mixture problems do in practical work. We now want to present a simple and quick method of solving a problem like this, using the Trane psychrometric chart. Our short-cut method will not give *exactly* the correct answer. However, it will be correct within a fraction of 1°, and the error is so small that we can forget about it. The method is sufficiently accurate, provided you use a sharp pencil and work carefully.

Short Method for Solving Mixture Problems. Trane Chart. Figure 18 illustrates our problem on the Trane volume chart from which specific volumes have been read in the solution presented above. We also use the Trane volume chart to find the specific volume at point M in our easy solution. You cannot read this accurately on Fig. 18 because of its small size, but if you plot point M on your own Trane volume chart, after locating point M as described below, you will see that point M represents a specific volume of 13.81 cu ft per lb.

Using a Trane chart, turn to Fig. 19. Plot point *A* on your chart, representing 95° dry bulb, 74° wet bulb. Plot point *B* representing conditions of 65° dry bulb, 54° wet bulb. Draw a straight line carefully with a sharp pencil from point *A* to point *B*. *CAUTION: Use a sharp pencil;*

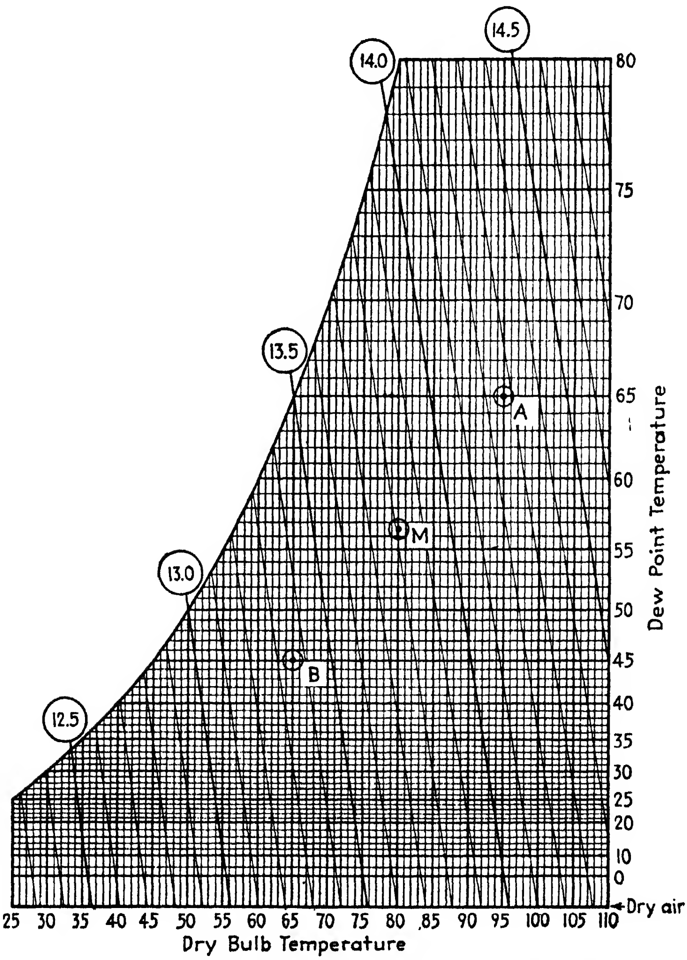


FIG. 18.—Specific volume from psychrometric chart. (*The Trane Co.*)

locate points A and B exactly; draw line AB correctly. You are going to measure distances from this line.

In this short-cut method we do not need to figure weights in pounds per minute for the two streams of air. Instead we may deal directly in cubic feet per minute. The steps listed below give the procedure for our short-cut method of solving mixture problems. Follow these steps on your own Trane chart on which you have drawn line AB, and see for yourself how easy this method is.

1. To obtain total cubic feet per minute of air at point M, add the cubic feet per minute at A to the cubic feet per minute at B. For our problem this is

$$1{,}000 + 1{,}000 = 2{,}000 \text{ cfm at } M$$

2. Express the cubic feet per minute at A and at B as a per cent of the total cubic feet per minute at point M. For our problem this is

Equation (1)

$$\text{At } A\text{: } \frac{\text{cfm at } A}{\text{cfm at } M} = \frac{1{,}000}{2{,}000} = 50\%$$

Equation (2)

$$\text{At } B\text{: } \frac{\text{cfm at } B}{\text{cfm at } M} = \frac{1{,}000}{2{,}000} = 50\%$$

3. Measure the line AB as exactly as you can. Use a ruler divided in tenths of an inch instead of eighths of an inch. For our problem this is almost exactly $4\frac{1}{8}$ in., or 4.12 in.

4. Take the per cent found in Eq. (1) in step 2, *expressed as a decimal*, and multiply it by the length of line AB as measured in step 3. For our problem this is

$$0.50\ (4.12) = 2.06 \text{ in., or } 2\tfrac{1}{16} \text{ in.}$$

5. Take the result of step 4 and measure this distance *from point B toward point A*. Note that although you obtained this distance by multiplying the length of line AB by the percentage of the air at A *you measure from point B toward point A*. (In our problem since we are measuring two equal air streams, each of which is 50 per cent of the total air at point M, the precaution just cited in step 5 is not important.) For our problem, measuring $2\frac{1}{16}$ in. from B toward A, we locate point M. In other words, point M in our problem is *halfway* between points A and B.

Your chart should now agree with Fig. 19. Point M is found from your chart or from Fig. 19 to be 80° dry bulb, 65° wet bulb, 56.6° dew point. The specific volume for point M is found to be 13.81 cu ft per lb from the Trane volume chart according to the method of Fig. 14*a*, Chap. XVIII.

We are sure that you agree that this graph-type method is much easier and quicker than the computation method of solving air-mixture problems. Let us now compare the accuracy of the results.

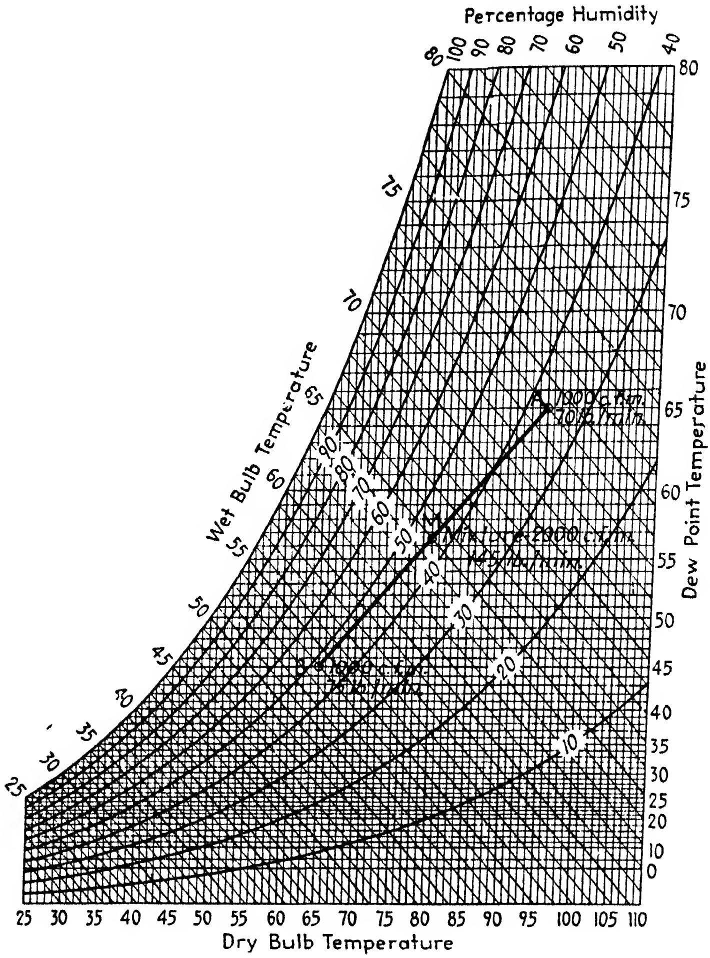


FIG. 19.—Showing result of problem on air mixtures. (*The Trane Co.*)

	Long method	Short graph method
Dry bulb	79.5°	80°
Wet bulb	64.8°	65°
Dew point	56.3°	56.6°
Relative humidity	44%	44%
Cubic feet per minute at *M*	2,003	2,000

From this comparison you can easily see that our short method, using the Trane chart, gives substantially accurate results. The slight error tends to make the dry-bulb, wet-bulb, and dew-point temperatures a little *too high,* which means that we shall never count on getting a *cooler* result from a mixture than will actually occur. In other words, the small error is in the direction of conservative and safe design, for summer air-conditioning work.

Let us now solve another mixture problem, in which the proportions are not exactly 50 = 50, using our short-cut method on the Trane chart. Let us assume that we are to mix 2,000 cfm at 95° dry bulb, 74° wet bulb (point A) with 4,000 cfm at 65° dry bulb, 54° wet bulb (point B). Following the method outlined in numbered steps on page 394, proceed as follows with your chart.

(1) 4,000 + 2,000 = 6,000 cfm at point M

(2) At A: $\dfrac{2{,}000}{6{,}000} = 0.333$

At B: $\dfrac{4{,}000}{6{,}000} = 0.667$

(3) Measure line AB, which is 4⅛ in. long
(4) 0.333 (4.12) = 1.37 in., or almost exactly 1⅜ in.
(5) Measure 1⅜ in. *from point B toward point A,* and you have located point M.

Your solution to the problem, which you have just solved by following the steps given above, should resemble Fig. 20. Point M should be located at 75° dry bulb, 61.5° wet bulb, 53° dew point, 46 per cent relative humidity.

Remember that point M will be *nearer to point B* than to point A if the amount of air at point B is greater. Conversely, if the amount of air is greater at point A, point M will be nearer point A. If you follow exactly the procedure given in the five steps described on this page you will not make any mistake in locating point M.

Conclusion. Before leaving this chapter, review the general characteristics of air washers and of extended surface cooling coils when used for *cooling and dehumidifying* air. Note that both air washers and coils *tend* to deliver air in a nearly saturated condition. The air will always be somewhat warmer than the chilled water or other cooling medium which is supplied. Keep in mind in your future work the following general characteristics:

1. Coils. Deliver air at a dry-bulb temperature 5 to 10° above the entering cooling medium, the air being at 70 to 90 per cent relative humidity. The relative humidity is usually near the higher figure.

2. Air Washers. Deliver air at 7 to 17° above the temperature of the chilled water supplied to the sprays, the air being at 70 to 99 per cent relative humidity. The relative humidity is usually between 85 and 95 per cent.

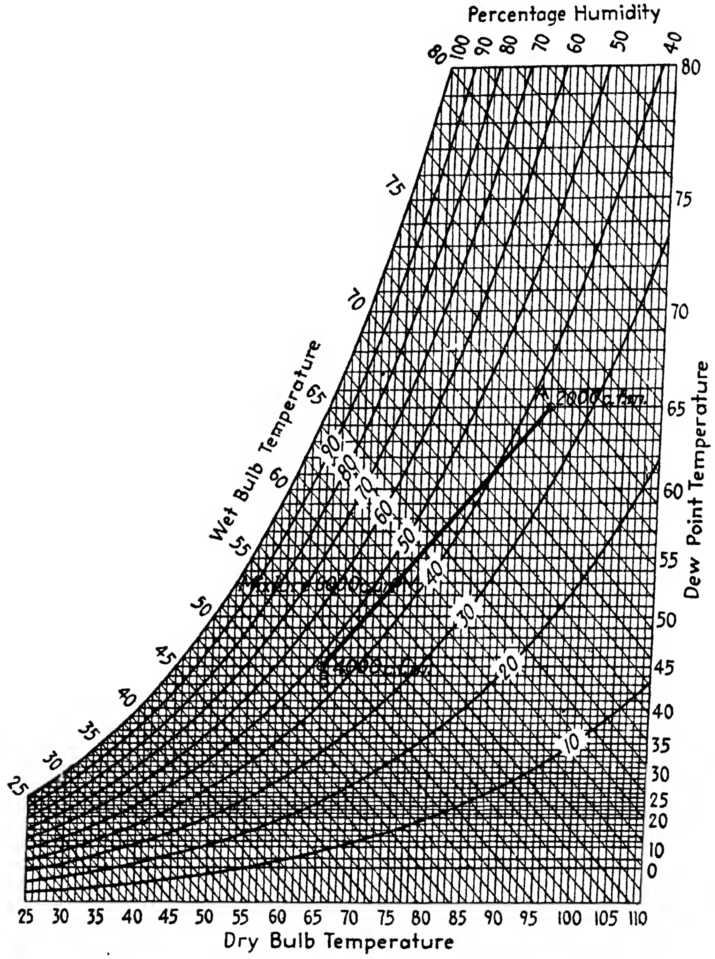


FIG. 20.—Solution of second air-mixture problem as shown on psychrometric chart. *(The Trane Co.)*

We realize that this chapter has been difficult because it has presented such a wide psychrometric study of air-conditioning processes of analysis on the psychrometric chart. We can assure you that this information will stand you in good stead in the future.

PROPERTIES OF MIXTURES OF AIR AND SATURATED WATER VAPOR *

Based on Barometric Pressure of 29.92 In.

Temp, °F	Weight of saturated vapor per pound of dry air: Pounds	Weight of saturated vapor per pound of dry air: Grains	Sensible heat of air, Btu per lb of dry air	Latent heat of vapor in mixture, Btu per lb of dry air	Total heat of mixture* Btu per lb of dry air
0	0.000781	5.47	0.000	0.852	0.852
2	0.000869	6.08	0.482	0.946	1.428
4	0.000963	6.74	0.964	1.047	2.011
6	0.001067	7.47	1.446	1.159	2.605
8	0.001183	8.28	1.928	1.285	3.213
10	0.001309	9.16	2.411	1.420	3.831
12	0.001447	10.13	2.893	1.568	4.461
14	0.001599	11.19	3.375	1.731	5.106
16	0.001764	12.35	3.858	1.908	5.766
18	0.001946	13.62	4.340	2.103	6.443
20	0.002144	15.01	4.823	2.314	7.137
22	0.002360	16.52	5.305	2.545	7.850
24	0.002596	18.17	5.787	2.796	8.583
26	0.002854	19.98	6.270	3.071	9.341
28	0.003134	21.94	6.752	3.370	10.122
30	0.003444	24.11	7.234	3.699	10.933
32	0.003782	26.47	7.716	4.058	11.783
33	0.003938	27.57	7.96	4.22	12.18
34	0.004100	28.70	8.20	4.40	12.60
35	0.004268	29.88	8.44	4.57	13.02
36	0.004442	31.09	8.68	4.76	13.44
37	0.004622	32.35	8.93	4.95	13.87
38	0.004809	33.66	9.17	5.14	14.31
39	0.005002	35.01	9.41	5.35	14.76
40	0.005202	36.41	9.65	5.56	15.21
41	0.005410	37.87	9.89	5.78	15.67
42	0.005625	39.38	10.14	6.01	16.14
43	0.005848	40.93	10.38	6.24	16.62
44	0.006078	42.55	10.62	6.48	17.10
45	0.00632	44.21	10.86	6.73	17.59
46	0.00656	45.94	11.10	6.99	18.09
47	0.00682	47.73	11.34	7.26	18.60
48	0.00708	49.58	11.58	7.54	19.12
49	0.00736	51.49	11.83	7.83	19.65
50	0.00764	53.47	12.07	8.12	20.19
51	0.00793	55.52	12.31	8.43	20.74
52	0.00823	57.64	12.55	8.75	21.30
53	0.00855	59.83	12.79	9.08	21.87
54	0.00887	62.09	13.03	9.41	22.45
55	0.00920	64.43	13.28	9.76	23.04
56	0.00955	66.85	13.52	10.13	23.64
57	0.00991	69.35	13.76	10.50	24.25
58	0.01028	71.93	14.00	10.89	24.88
59	0.01066	74.60	14.24	11.28	25.52
60	0.01105	77.3	14.48	11.69	26.18
61	0.01146	80.2	14.72	12.12	26.84
62	0.01188	83.2	14.97	12.56	27.52
63	0.01231	86.2	15.21	13.01	28.22
64	0.01276	89.3	15.45	13.48	28.93
65	0.01323	92.6	15.69	13.96	29.65
66	0.01370	95.9	15.93	14.46	30.39
67	0.01420	99.4	16.18	14.97	31.15
68	0.01471	103.0	16.42	15.50	31.92
69	0.01524	106.6	16.66	16.05	32.71
70	0.01578	110.5	16.90	16.61	33.51
71	0.01634	114.4	17.14	17.19	34.33
72	0.01692	118.4	17.38	17.79	35.17
73	0.01751	122.6	17.63	18.41	36.03
74	0.01813	126.9	17.87	19.05	36.91

Temp, °F	Weight of saturated vapor per pound of dry air: Pounds	Weight of saturated vapor per pound of dry air: Grains	Sensible heat of air, Btu per lb of dry air	Latent heat of vapor in mixture, Btu per lb of dry air	Total heat of mixture† Btu per lb of dry air
75	0.01877	131.4	18.11	19.71	37.81
76	0.01942	135.9	18.35	20.38	38.73
77	0.02010	140.7	18.59	21.08	39.67
78	0.02080	145.6	18.84	21.80	40.64
79	0.02152	150.6	19.08	22.55	41.63
80	0.02226	155.8	19.32	23.31	42.64
81	0.02303	161.2	19.56	24.11	43.67
82	0.02381	166.7	19.80	24.92	44.72
83	0.02463	172.4	20.04	25.76	45.80
84	0.02547	178.3	20.29	26.62	46.91
85	0.02634	184.4	20.53	27.51	48.04
86	0.02723	190.6	20.77	28.43	49.20
87	0.02815	197.0	21.01	29.38	50.39
88	0.02910	203.7	21.25	30.35	51.61
89	0.03008	210.6	21.50	31.26	52.86
90	0.03109	217.6	21.74	32.39	54.13
91	0.03213	224.9	21.98	33.46	55.44
92	0.03320	232.4	22.22	34.59	56.78
93	0.03430	240.1	22.46	35.69	58.15
94	0.03544	247.1	22.71	36.86	59.56
95	0.03662	256.3	22.95	38.06	61.01
96	0.03783	264.8	23.19	39.30	62.48
97	0.03908	273.6	23.43	40.57	64.00
98	0.04036	282.5	23.67	41.88	65.55
99	0.04169	291.8	23.91	43.24	67.15
100	0.04305	301.3	24.16	44.63	68.79
101	0.04446	311.2	24.40	46.07	70.47
102	0.04591	321.4	24.64	47.54	72.18
103	0.04741	331.9	24.88	49.07	73.95
104	0.04895	342.7	25.13	50.64	75.77
105	0.0505	354	25.37	52.26	77.63
106	0.0522	365	25.61	53.92	79.53
107	0.0539	377	25.85	55.64	81.49
108	0.0556	389	26.09	57.41	83.50
109	0.0574	402	26.33	59.23	85.57
110	0.0593	415	26.58	61.11	87.69
111	0.0612	428	26.82	63.04	89.86
112	0.0631	442	27.06	65.04	92.10
113	0.0652	456	27.30	67.10	94.40
114	0.0673	471	27.55	69.22	96.77
115	0.0694	486	27.79	71.40	99.10
116	0.0717	502	28.03	73.65	101.68
117	0.0739	518	28.27	75.97	104.24
118	0.0763	534	28.51	78.36	106.87
119	0.0788	551	28.76	80.80	109.56
120	0.0813	569	29.00	83.37	112.37
125	0.0953	667	30.21	97.33	127.54
130	0.1114	780	31.42	113.64	145.06
135	0.1305	913	32.63	132.71	165.34
140	0.1532	1072	33.85	155.37	189.22
145	0.1800	1260	35.06	182.05	217.1
150	0.2122	1485	36.27	214.03	250.3
155	0.2511	1758	37.48	252.61	290.1
160	0.2987	2091	38.69	299.55	338.2
165	0.3577	2504	39.91	357.75	397.7
170	0.4324	.....	41.12	431.2	472.3
175	0.5290	.....	42.33	526.0	568.3
180	0.6577	.....	43.55	651.9	695.5
185	0.8359	.....	44.76	826.1	870.9
190	1.0985	.....	45.97	1082.3	1128.3
200	2.2953	.....	48.40	2247.5	2296.0

* Reprinted by permission from G. A. Goodenough, "Properties of Steam and Ammonia," John Wiley & Sons, Inc.

† Values in this column do not include the heat of the liquid.

SOURCE: Table arrangement used by permission of The Trane Company, La Crosse, Wis.

Total Heat Content of Air at Various Wet-bulb Temperatures *

Interpolated to Tenths of a Degree from Goodenough's Table of "Properties of Air"

Condensed Table

Wet-bulb temp	Btu per lb	Wet-bulb temp	Btu per lb	Wet-bulb temp	Btu per lb	Wet-bulb temp	Btu per lb	Wet-bulb temp	Btu per lb	Wet-bulb temp	Btu per lb	Wet-bulb temp	Btu per lb	Wet-bulb temp	Btu per lb	Wet-bulb temp	Btu per lb	Wet-bulb temp	Btu per lb
40	15.21	40.0	15.21	45.0	17.59	50.0	20.19	55.0	23.04	60.0	26.18	65.0	29.65	70.0	33 51	75 0	37.81
41	15.67	0.1	15.26	0.1	17.64	0.1	20.25	0.1	23.10	0.1	26 25	0.1	29 72	0.1	33 59	0 1	37.90
42	16.14	0.2	15.30	0.2	17.69	0.2	20.30	0.2	23.16	0.2	26.31	0.2	29.80	0.2	33 67	0 2	37.99
43	16 62	0.3	15.35	0.3	17.74	0.3	20.36	0.3	23.22	0.3	26.38	0.3	29 87	0.3	33 76	0 3	38 09
44	17.10	0.4	15.39	0.4	17.79	0.4	20.41	0.4	23.28	0.4	26.44	0.4	29 95	0.4	33 84	0.4	38 18
45	17.59	40.5	15.44	45.5	17 84	50.5	20.47	55.5	23.34	60.5	26.51	65.5	30.02	70 5	33 92	75.5	38 27
46	18.09	0.6	15 49	0.6	17.89	0.6	20 52	0.6	23.40	0.6	26.58	0.6	30 09	0.6	34 00	0.6	38 36
47	18.60	0.7	15.53	0.7	17.94	0.7	20.58	0.7	23.46	0.7	26.64	0.7	30.17	0.7	34 08	0.7	38 45
48	19.12	0.8	15 58	0.8	17 99	0.8	20.63	0.8	23.52	0.8	26.71	0 8	30 24	0.8	34 17	0 8	38.55
49	19.65	0.9	15.62	0.9	18.04	0.9	20.69	0.9	23.58	0.9	26.77	0.9	30.32	0.9	34.25	0.9	38.64
50	20.19	41.0	15.67	46.0	18.09	51.0	20.74	56.0	23.64	61.0	26.84	66 0	30 39	71.0	34 33	76.0	38.73
51	20.74	0.1	15.72	0.1	18.14	0.1	20.80	0.1	23.70	0.1	26.91	0.1	30.47	0.1	34.41	0.1	38.82
52	21.30	0.2	15.76	0.2	18.19	0.2	20.85	0.2	23.76	0 2	26 98	0.2	30.54	0.2	34.50	0.2	38.92
53	21.87	0.3	15 81	0.3	18 24	0.3	20.91	0.3	23.82	0.3	27.04	0 3	30.62	0.3	34 58	0.3	39.01
54	22.45	0.4	15.86	0.4	18.29	0.4	20.96	0.4	23.88	0.4	27.11	0.4	30 69	0.4	34.67	0.4	39.11
55	23.04	41.5	15.91	46 5	18.35	51.5	21.02	56.5	23.95	61.5	27.18	66.5	30 77	71.5	34.75	76.5	39.20
56	23.64	0.6	15.95	0.6	18.40	0.6	21.08	0.6	24 01	0.6	27.25	0.6	30 85	0.6	34.83	0.6	39.29
57	24.25	0.7	16.00	0.7	18.45	0.7	21.13	0.7	24 07	0.7	27.32	0.7	30.92	0.7	34.92	0.7	39 39
58	24.88	0.8	16.05	0.8	18.50	0.8	21.19	0.8	24.13	0.8	27.38	0.8	31.00	0.8	35.00	0.8	39.48
59	25.52	0.9	16.09	0.9	18.55	0.9	21.24	0.9	24.19	0.9	27.45	0.9	31.07	0.9	35.09	0.9	39.58
60	26.18	42.0	16.14	47 0	18.60	52.0	21.30	57.0	24.25	62.0	27.52	67.0	31.15	72.0	35.17	77.0	39.67
61	26 84	0.1	16.19	0.1	18.65	0.1	21.36	0.1	24.31	0.1	27.59	0.1	31.23	0.1	35 26	0.1	39.77
62	27.52	0.2	16.24	0.2	18.70	0.2	21.41	0.2	24.38	0.2	27.66	0.2	31.30	0.2	35.24	0.2	39.86
63	28.22	0.3	16.28	0.3	18.76	0.3	21.47	0.3	24.44	0.3	27.73	0.3	31.38	0.3	35.43	0.3	39.96
64	28.93	0.4	16.33	0.4	18.81	0.4	21.53	0.44	24.50	0.4	27.80	0.4	31.46	0.4	35.51	0.4	40.06
65	29.65	42.5	16 38	47.5	18.86	52.5	21.59	57.5	24.57	62.5	27.87	67.5	31.54	72.5	35.60	77.5	40.16
66	30.39	0 6	16.43	0.6	18.91	0.6	21.64	0.6	24.63	0.6	27.94	0.6	31.61	0.6	35.69	0.6	40.25
67	31.15	0.7	16.48	0.7	18.96	0.7	21.70	0.7	24.69	0.7	28.01	0.7	31.69	0.7	35 77	0.7	40.35
68	31.92	0.8	16.52	0.8	19.02	0.8	21.76	0.8	24.75	0.8	28.08	0.8	31.77	0.8	35.86	0.8	40.45
69	32.71	0.9	16.57	0.9	19.07	9.9	21.81	0.9	24.82	0.9	28.15	0.9	31.84	0.9	35.94	0.9	40.54
70	33.51	43.0	16.62	48.0	19.12	53.0	21.87	58.0	24.88	63.0	28.22	68.0	31.92	73.0	36.03	78.0	40.64
71	34.33	0.1	16.67	0.1	19.17	0.1	21.93	0.1	24.94	0.1	28.29	0.1	32.00	0.1	36.12	0.1	40.74
72	35.17	0.2	16.72	0.2	19.23	0.2	21.99	0.2	25.01	0.2	28.36	0.2	32.06	0.2	36.21	0.2	40.84
73	36.03	0.3	16.76	10 3	19.28	0.3	22.04	0.3	25.07	0.3	28.43	0.3	32.16	0.3	36.29	0.3	40.94
74	36.91	0.4	16.81	0.4	19.33	0.4	22.10	0.4	25.14	0.4	28.50	0.4	32.24	0.4	36.38	0.4	41.04
75	37.81	43.5	16.86	48.5	19.39	53.5	22.16	58.5	25.20	63.5	28.58	68.5	32.32	73.5	36.47	78.5	41.14
76	38.73	0.6	16.91	0.6	19.44	0.6	22.22	0.6	25.26	0.6	28.65	0.6	32.39	0.6	35.56	0.6	41.23
77	39.67	0.7	16.96	0.7	19.49	0.7	22.28	0.7	25.33	0.7	28.72	0.7	32.47	0.7	36.65	0.7	41.33
78	40.64	0.8	17.00	0.8	19.54	0.8	22.33	0.8	25.39	0.8	28.79	0.8	32.55	0.8	36.73	0.8	41.43
79	41.63	0.9	17.05	0.9	19.60	0.9	22.39	0.9	25.46	0.9	28.86	0.9	32.63	0.9	36.82	0.9	41.53
80	42.64	44.0	17.10	49.0	19.65	54.0	22.45	59.0	25.52	64.0	28.93	69.0	32.71	74.0	36.91	79.0	41.63
81	43.67	0 1	17.15	0.1	19.70	0.1	22.51	0.1	25.59	0.1	29.00	0.1	32.79	0.1	37.00	0.1	41.73
82	44.72	0.2	17.20	0.2	19.76	0.2	22.57	0.2	25.65	0.2	29.07	0.2	32.87	0.2	37.09	0.2	41.83
83	45.80	0.3	17.25	0.3	19.81	0.3	22.63	0.3	25.72	0.3	29.15	0.3	32.95	0.3	37.18	0.3	41.93
84	46.91	0.4	17.30	0.4	19.87	0.4	22.69	0.4	25.78	0.4	29.22	0.4	33.03	0.4	32.27	0.4	42.03
85	48.04	44.5	17.35	49.5	19.92	54.5	22.75	59.5	25.85	64.5	29.29	69.5	33.11	74.5	37.36	79.5	42.14
86	49.20	0.6	17.39	0.6	19.97	0.6	22.80	0.6	25.92	0.6	29.36	0.6	33.19	0.6	37.45	0.6	42.24
87	50.39	0.7	17.44	0.7	20.03	0.7	22.86	0.7	25.98	0.7	29.43	0.7	33.27	0.7	37.54	0.7	42.34
88	51.61	0.8	17.49	0.8	20.08	0.8	22.92	0.8	26.05	0.8	29.51	0.8	33.35	0.8	37.63	0.8	42.44
89	52.86	0.9	17.54	0.9	20.14	0.9	22.98	0.9	26.11	0.9	29.58	0.9	33.43	0.9	37.72	0.9	42.54
90	54.13																

* Table arrangement used by permission of The Trane Company, La Crosse, Wis.

CHAPTER XXI

CALCULATIONS FOR SUMMER AIR CONDITIONING

General. Let us review briefly our knowledge to date regarding psychrometry and summer air conditioning. In Chap. VII we learned the principles of psychrometry from the psychrometric tables. In Chap. XIX we learned how to calculate summer cooling load, based upon data in Chap. XIV. In Chap. XVIII, we studied different types of psychrometric charts. In Chap. XX, we studied the psychrometry of various air-conditioning processes and we presented a new, more compact, and more accurate psychrometric table.

We now wish to consider typical problems in which a summer air-conditioning cooling load has been calculated, and where our problem consists of determining the psychrometric conditions of the cold conditioned air to be supplied to the conditioned space. Because we also must determine the *quantity* of air to be supplied to the conditioned space, we must learn to determine the effect of fresh air, since few jobs operate on a basis of 100 per cent recirculation. In some of our calculations, however, we shall *assume* 100 per cent recirculation for simplicity, after which we shall consider the use of some fresh outdoor air. We also wish to consider the same air-conditioning installation at a time when its cooling load is at a *maximum* and on a cool cloudy summer day when its cooling load is *greatly reduced.*

We shall find installations in which if we cool the air to a low enough dew point to handle the latent heat load the conditioned space would be too cold. We must find the remedy for this condition and discuss the equipment used to provide this remedy. We must also set up standards regarding how cold the supply of conditioned air may be without causing objectionable cold spots and drafts in the conditioned space.

This chapter will be treated with a minimum of pure text material; instead practical problems will be presented and solved. Study these problems carefully and learn from them. The first problem will be solved by using arithmetic computations based upon the psychrometric table given at the end of Chap. XX.

Diffusion Temperature Differential. We are now ready to solve Prob. 1, but before we do this we wish to decide upon standards as to the *dry-bulb* temperature of the cool conditioned air to be supplied for summer air conditioning. The conditioned air leaving the supply grilles is obviously cooler than the room temperature. This temperature difference is known by several names, perhaps the most popular of which is "diffusion temperature differential." This name arises because we consider the supply air as being "diffused into" or "mixed with" the room air. An ideal diffusion temperature differential would be 10 or 12°, because it would be easier to make sure that no extremely cold air would strike occupants of the conditioned space. However, a diffusion temperature differential of 20° requires just *half as much air* as a diffusion temperature differential of 10°. [Remember the equation $Q = cW(T_2 - T_1)$. If we *double* the temperature difference $T_2 - T_1$, we could cut W *in half* and still obtain the same value for Q.]

This means that a 20° diffusion temperature differential will involve duct work and fans half the size that would be required with a 10° diffusion temperature differential. Also the conditioner itself may have a smaller air-handling capacity, although its heat-removal capacity must be the same. Selection of diffusion temperature differential is thus a question of compromise in order to keep down the size of the air-distribution system and assure reasonable installation cost. We may set up the following standards regarding diffusion temperature differential:

1. Less than 15°. Only used on work of the highest class where installation cost and space for ducts and fans is not considered. Rarely encountered.

2. 15° Diffusion Temperature Differential. Use in good-class work, especially where air-supply outlets are near occupants of the conditioned space. Size and cost of fans and ductwork is greater than desired.

3. 20° Diffusion Temperature Differential. A good practical standard resulting in reasonable size and cost of the air-distribution system. Care will ensure mixing of the cold air with the room air before it strikes occupants.

4. 25° Diffusion Temperature Differential. Used where duct work and fan sizes must be reduced to fit cramped space conditions. Frequently no cost saving results because great care must be taken to prevent the cold air from striking occupants of the room.

We shall solve Prob. 1 for two alternate diffusion temperature differentials.

Alternate A will be based on 15° diffusion temperature differential (which means conditioned air at 65° dry bulb, in Prob. 1).

Alternate B will be based upon a 20° diffusion temperature differential (which means conditioned air at 60° dry bulb, in Prob. 1).

***Problem* 1.**

Inside design conditions

80° dry bulb, 67° wet bulb, 60.3° dew point

Cooling load computed as follows:

Sensible heat	**48,000 Btu per hr**
Latent heat	**12,000 Btu per hr**
Total heat	**60,000 Btu per hr**

Conditioners are available that will deliver air at 75 to 90 per cent relative humidity, from 55 to 65° dry bulb. For a 100 per cent recirculation job (no fresh air), find:

1 Psychrometric conditions of air leaving conditioner (supply air).

2. Amount of conditioned air *in cubic feet per minute*, measured at the psychrometric conditions of the cool conditioned air.

Solution. We shall first solve alternate A based upon conditioned air at 65° dry bulb. We shall determine the sensible heat that each pound of air supplied can pick up in raising from 65 to 80°. This is done in steps 1, 2, and 3. We shall then divide the sensible cooling load by this sensible-heat pick-up value in Btu per pound, which will give us a definite value for the pounds per minute of conditioned air to be supplied. This is done in steps 4 and 5 of the solution. After determining the pounds per minute of conditioned air that is to be supplied, we must determine how much the dew point of the conditioned air will be increased by the latent-heat load. To do this we must, of course, determine the latent load expressed in *grains of moisture per minute.* This is done in steps 6 and 7 of the solution. Steps 8 and 9 of the solution are self-explanatory. The solution of alternate A of the Prob. 1 follows directly below:

Alternate A. Conditioned air at 65° dry bulb

(1) Sensible heat at 80° dry bulb (conditioned space) = 19.32 Btu per lb
(2) Sensible heat at 65° dry bulb (supply air) = 15.69 Btu per lb

(3) Sensible heat *each pound* of conditioned air can pick up = *difference* = 3.63 Btu per lb

(4) From problem data, sensible heat load $= \dfrac{48{,}000}{60} = 800$ Btu per min

(5) $\dfrac{800}{3.63} = 220$ lb per min of air required for sensible cooling

(6)[1] Latent-heat load $= \dfrac{12{,}000}{60}\left(\dfrac{1}{1{,}050}\right) = 0.19$ lb moisture per minute

or $0.19\ (7{,}000) = 1330$ grains moisture per minute

(7) $\dfrac{1{,}330}{220} = 6.03$ grains moisture per pound of supply air

(8) Moisture content desired in conditioned space 60.3° D.P. = 78.17 grains per lb
Less moisture added to each pound of air supplied = 6.03 grains per lb
Difference = moisture content required in supply air = 72.14 grains per lb

From the table on page 403, Chap. XX, this is a supply dew point of 58.1°F. From the Trane chart, air at 65° dry bulb, 58.1° dew point, is 60.7° wet bulb

This is our supply air. Ans.

(9) From the Trane volume chart, our supply air has a specific volume of 13.44 cu ft per lb. Hence,

220 lb per min (13.44) = 2,950 cfm of supply air. *Ans.*

We solve alternate B of Prob. 1 by the exact same method as that used for the solu-

[1] 12,000 divided by 60 = Btu per *min* latent heat load. 1,050 is taken as average latent heat of vaporization (h_{fg}) of the moisture evaporated into the air in Btu per pound of water.

tion of alternate A above. The only difference is that our diffusion temperature differential is 20° in alternate B.

Alternate B. Conditioned air at 60° dry bulb. (This solution follows the same steps as alternate A, the explanation being omitted.)

(1) 19.32 Btu per lb

(2) 14.48 Btu per lb

(3) Differential, step 1 minus step 2 = 19.32 − 14.48 = 4.84 Btu per lb

(4) $\frac{48{,}000}{60} = 800$ Btu per min

(5) $\frac{800}{4.84} = 165$ lb per min

(6) Same as alternate A

(7) $\frac{1{,}330}{165.5} = 8.06$ grains per lb

(8) 78.17 − 8.06 = 70.11 grains per lb
70.13 grains per lb is a dew point of 57.3°, which with a 60° dry bulb gives 58.4° wet bulb, *supply air.* *Ans.*

(9) Specific volume supply air = 13.3 cu ft per lb
165(13.3) = 2190 cfm of supply air. *Ans.*

Since the total cooling load for alternates A and B of Prob. 1 *is the same* (60,000 Btu per hr), we can obtain the tonnage based upon 100 per cent recirculation as

$$\frac{60{,}000}{12{,}000} = 5.0 \text{ tons}$$

Problem 2. A dehumidifier is handling 500 ib of recirculated air per minute with entering conditions of 80° dry bulb and 50 per cent relative humidity and the leaving-air temperature at 50°. Find the following:

1. Refrigerating duty in Btu per minute.
2. Change in moisture content in grains per pound.

Solution

Condition	Entering	Leaving
Dry bulb, °F	80	50
Wet bulb, °F	67	50
Dew point, °F	60	50
Relative humidity, %	50	100
Specific volume, cu ft per lb	13.84	13.0
Grains per pound	77.3	53.50
Total heat, Btu per lb	31.15	20.19

A dehumidifier or air washer may be considered as completely saturating the air that passes through it.

1. Difference in total heats, entering and leaving conditions:

$$31.15 - 20.19 = 10.96 \text{ Btu per lb}$$
$$500 \times 10.96 = 5480 \text{ Btu per min}$$

This is the refrigerating duty.

2. Difference in humidity at the two dew points:

$$77.3 - 53.5 = 23.8 \text{ grains per lb, a decrease}$$

Problem 3. An air washer is handling 400 lb of recirculated air and 100 lb of outside air. The recirculated air is at 80° dry bulb and 50 per cent relative humidity. The outside air is at 95° dry bulb and 75° wet bulb. The leaving-air temperature out of the air washer is 53°. Find the following:

1. Refrigerating duty in Btu per minute.
2. Total moisture extracted in pounds of moisture per minute.
3. Refrigerating duty that the air can accomplish in rising to 80° dry bulb and 50 per cent relative humidity in Btu per minute.
4. Divide the refrigerating duty in 3 into sensible and moisture Btu per minute, and determine the grains of moisture that will be absorbed.

Solution

Condition	Entering	Leaving
Dry bulb, °F	83	53
Wet bulb, °F	68.9	53
Dew point, °F	62.0	53
Relative humidity, %	48	100
Specific volume, cu ft per lb		
Grains per pound	83	59.9
Total heat, Btu per lb	32.8	21.9

1. $(32.8 - 21.9) \times 500 = 5450$ Btu per min.

2. $(83.0 - 59.9) \times 500 - \dfrac{1}{7000} = 1.64$ lb water per minute.

3. Total heat at 80° dry bulb and 67° wet bulb is found to be 31.18 Btu per lb.

$$(31.18 - 21.9) \times 500 = 4640 \text{ Btu per min}$$

This is the refrigeration duty accomplished.

4. Grains per pound at 60° dew point............77.0
Grains per pound at 53° dew point............59.9
17.1 grains per lb added

$$17.1 \times 500 \times \frac{1}{7{,}000} \times 1{,}050 = 1290 \text{ Btu per min latent}$$

$$500 \times 24 \times (80 - 53) = 3240 \text{ Btu per min dry}$$

$$1290 + 3240 = 4530 \text{ Btu per min total load}$$

(Close enough check.)

Problem 4. A space to be conditioned has a sensible-heat load of 180,000 Btu per hr and a moisture load of 47.5 lb per hr; it is required that the air conditions be maintained at 80° dry bulb and 50 per cent relative humidity. Determine the quantity of air per minute in pounds and the leaving-air temperature required out of the dehumidifier (air washer). The air introduced into the conditioned space must be able to absorb both the sensible and moisture tons of refrigeration in rising to the room conditions required.

Solution

Sensible heat load is 180,000 Btu per hr, or 3000 Btu per min
Moisture load is 47.5 lb water per hour

$$\frac{47.5 \times 7000}{60} = 5{,}540 \text{ grains per min}$$

Desired conditions are 80° dry bulb and 50 per cent relative humidity.
By trial-and-error method, assume a 50°F air-washer leaving-air condition, then

$$80 - 50 \text{ equals a } 30^\circ \text{ range}$$

Moisture content at 80° and 50 per cent R.H. is 77 grains per lb.
Moisture content at 50° and saturated is 53.5 grains.

$$77 - 53.5 = 23.8 \text{ grains per lb}$$

$$\text{Sensible load } \frac{3000}{0.24 \times 30} = 418 \text{ lb per min}$$

$$\text{Moisture load } 5540 \times \frac{1}{23.8} = 233 \text{ lb per min}$$

Since these pounds per minute values do not agree, our first assumption is incorrect. Now, assume a 56°F air-washer leaving-air condition, then

$$80 - 56 = \text{a } 24^\circ \text{ range}$$

By the same treatment as before, then

$$\text{Sensible load } \frac{3000}{0.24 \times 24} = 521 \text{ lb air per min}$$

$$\text{Moisture load } \frac{5540}{10.5} = 527 \text{ lb air per min}$$

Since these values agree closely, the temperature of the air off the air washer may be taken as 56°F. The quantity of air is 527 lb per min.

CHAPTER XXII

FLUID FLOW

General. You may have an idea that "fluid" means "liquid." This is not true because fluid actually means "matter not in the solid state" or "matter in *either* the *liquid* or *gaseous* state." In other words, steam, air, refrigerant gases, water, mercury, oil, carbon dioxide are all fluids.

The basic theories of fluid flow apply to the flow of air through ducts, whether square, rectangular, or round. The same basic theories apply to the flow of water through piping systems. Therefore, it is obvious that fluid flow must be understood in a general way in order to be able to design duct work and water-piping systems and to select properly the fans and pumps to be used in such systems. It is possible to design water-piping systems on a rule-of-thumb basis because only certain standard sizes of pipe are manufactured. Similarly as to fittings, only certain valves, elbows, tees, etc., are available for use. Finally, pump manufacturers make hundreds of different rotors and sizes of pumps. In some cases, two pumps differ only by ⅛ in. in the diameter of the rotor (or impeller). All this serves to simplify the design of water piping compared with air ducts.

Air ducts are almost invariably custom-made to fit the individual conditions of a particular installation. The ducts are usually rectangular instead of round, which means that there are thousands of different duct sizes. Furthermore, elbows and other duct fittings may be made up in many different styles, shapes, and sizes. Finally, while fan manufacturers make many different fans, a particular design of fan may be available in 20 or 30 different sizes instead of several hundred. From this comparison you can see that the design of duct work is considerably more complicated than the design of water-piping systems.

You are already thoroughly familiar with the idea of "pressure." Now that we are ready to discuss flow of fluids, we shall have occasion to use another word meaning pressure. This new word is "head." From now on we shall use the words "pressure" and "head" interchangeably. You will soon see the logic of using this word.

Static Pressure. The word *static* means "standing still," "at rest," or "not in motion." All the pressures we have discussed thus far have been those of liquids or gases that were substantially standing still.[1] We have

[1] It is true that we discussed the pressure of refrigerants inside pipe lines, and these refrigerants were moving rather than standing still. However, we discussed mainly the pressure of the refrigerant within the condenser and the evaporator where the refrigerants were most nearly "standing still" or static. In any event, refrigerants were moving slowly and were under high absolute pressures.

considered the pressure within various tanks, condensers, evaporators, etc., as measured by ordinary pressure gauges. These pressures, expressed on the absolute or gauge pressure scale or in inches of mercury, as given in Chap. III, have been *static* pressures.

We have not called your attention to the term "static pressure" thus far because we have not dealt with any other type of pressure. Since we are soon to encounter another type of pressure different and distinct from static pressure, it is important that you learn the definition of static pressure at this time. We may define static pressure as follows: "Static pressure is the compressive (or squeezing) pressure existing in a fluid and is a measure of its pressure energy; static pressure is exerted in all directions, is not affected by motion of the fluid, and may exist in a fluid either at rest or in motion."

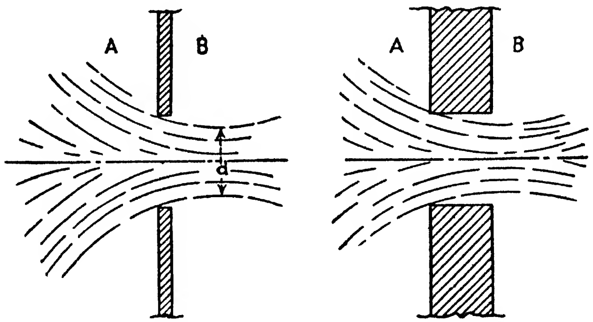


FIG. 1.—Flow through orifices. Direction from left to right.

At the ordinary pressures that will be encountered in handling air by means of fans and ducts, the static pressure of the air will be slightly different from atmospheric pressure. This difference is so slight that we neglect the volume change that it causes according to the pressure-volume law (see Chap. III).

Static-pressure Difference As a Cause of Flow. Figure 1 illustrates a section of a tank or duct wall, the pressure to the left of the duct wall being slightly higher than the pressure to the right. Under these conditions, a fluid will flow from left to right, *i.e.*, from region *A* to region *B*. In other words, air or any other fluid will flow *from* a region of *higher pressure* to a region of *lower pressure.*

For the conditions illustrated in Fig. 1, it makes no material difference whether the absolute pressure in regions *A* and *B* is slightly above or slightly below atmospheric pressure. The *difference* between the two pressures causes the flow to take place in the direction from the higher pressure to the lower pressure. The greater this pressure *difference*, the higher the velocity with which the fluid flows through the opening. (The dimensions marked d in Fig. 1 are actually less than three-fourths of the area of the hole in the wall, but this phase of fluid flow need not concern us now.)

Holes through which flow of fluids takes place, as illustrated in Fig. 1, are called "nozzles," regardless of whether they are round, square, triangular, or any other shape. We also call them nozzles regardless of whether the wall is a thin piece of sheet metal or a much thicker material. The thickness of the material does not matter greatly, as is illustrated in Fig. 1.

We have stated that it is the difference in *absolute* static pressure which causes the flow of a fluid through a nozzle, as illustrated in Fig. 1. These pressures need not be *absolute* pressures, since we are taking the *difference* between them. As a matter of fact, we need not even know the absolute pressure or gauge pressure on the two sides of a nozzle provided *we do have a means of knowing the pressure difference across the nozzles.*

The formulas concerning the flow of fluids are based simply upon this *pressure difference.* In order to have a general equation applying to the flow of any fluid (any liquid or any gas), the equation of flow does not state a pressure difference in pounds per square inch, ounces per square inch, inches of mercury, or any of the expressions of pressure which are well known to you now. Instead, the equation for flow through a nozzle is based upon the head (or pressure) in *feet of the liquid* that is flowing through the nozzle. In the next paragraph, we commence a careful study of this new method of expressing pressure or head. First, however, we wish to give you the formula for fluid flow through an orifice, in two forms:[1]

$$V_s = \sqrt{2gh} \tag{1}$$

$$V = 60\sqrt{2gh} \tag{2}$$

where V_s = velocity, ft per sec
V = velocity, fpm
g = "acceleration due to gravity" which is a *constant* for our purpose, its value being 32.2 ft per sec per sec
h = heat (or pressure) in *feet of the fluid* causing the flow

Meaning of "Head in Feet of Fluid." You have probably experienced a pressure of water on your ear drums when diving deeper than 5 or 6 ft. When we discussed atmospheric pressure in Chap. III we took account of

[1] The formulas as given on this page and as discussed in this chapter are based upon "perfect" or frictionless orifices. Actual orifices do involve some friction losses. To take account of this friction loss, we write the flow equation as

$$V_s = k\sqrt{2gh} \tag{1}$$

The value k is a constant for a particular orifice; it might be 0.8 for a good orifice and 0.5 for a poor orifice. ("Orifice" here is the same as "nozzle.")

the actual condition of the earth's atmosphere, which was such that the air near the earth's surface is at a higher pressure and at a higher density than the air far above the earth's surface. The column of air shown in Fig. 1, Chap. III, illustrates this. In our present discussion, we shall consider columns of fluid in which the fluid at the bottom of the column is *not* compressed. In other words, we now are going to consider columns of various fluids under conditions where the fluid has the same weight per cubic foot at all points of altitude in the column.

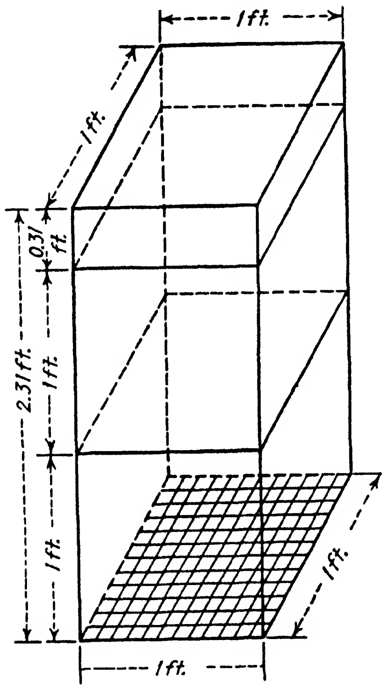


FIG. 2.—Method of determining unit pressure.

Figure 2 illustrates a column of water. If we had an *empty* tank the shape of Fig. 2, a pressure *gauge* connected to the bottom of the tank would show a pressure of 0 psig. This would be correct because the pressure would be simply atmospheric pressure. If we now fill the tank with water to a height of 2.31 ft, as indicated in Fig. 2, there will be a definite pressure at the bottom of the tank, caused by the weight of the water column 2.31 ft high. Let us calculate the pressure at the bottom of the tank in Fig. 2. One cubic foot of water weighs 62.4 lb. It is obvious from a study of Fig. 2 that the water column represented there has an area of 1 sq ft and is 2.31 ft high. Therefore, its volume is 2.31 cu ft. Hence the weight of the water shown in Fig. 2 may be found thus:

$$2.31\ (62.4) = 144 \text{ lb}$$

The water column shown in Fig. 2 is resting on the bottom of the tank, which has an *area* of 1 sq ft, or 144 sq in. Therefore, the weight of the water just calculated is resting uniformly on a total area of 144 sq in. (If you have trouble seeing this, count the small squares.) We may obtain the weight of water that is resting on any one square inch:

$$\frac{144 \text{ lb}}{144 \text{ sq in.}} = 1.00 \text{ psi}$$

The area of the water column does not affect the pressure *per square inch* at the bottom of the column. Draw a picture similar to Fig. 2, but make the area of the tank 2 sq ft, ½ sq ft, or some value other than 1 sq ft. The total weight of the water will be different, but so will the *number of square inches* of tank bottom supporting this water. Even if you take the extreme case of a column 2.31 ft high measuring 1 by 1 in., the pressure at the bottom of the column will be 1.00 psi. An accurate pressure gauge connected at the *bottom* of the tank in Fig. 2 would read 1.00 psi.

Figure 3 illustrates a tank containing water to a level 23.1 ft above bottom of the tank. The pressure gauge at the bottom of the tank, such as the one to the *left* in Fig. 3, would therefore read 10 lb, because the water column here is ten times as high as the column we have studied in detail in Fig. 2.

Suppose that we have a pipe outlet at the bottom of the tank, as shown to the right in Fig. 3. Let us assume that this pipe outlet is tightly capped and a pressure gauge is inserted in the pipe, as shown in Fig. 3. This second pressure gauge will, of course, also read 10 psi. (The pressure as indicated by the two gauges in Fig. 3, and all pressures that we discussed in connection with Fig. 2, are *static* pressures.)

In this section we started to explain the meaning of "head in feet of liquid." By now it should be clear that the pressure caused by a column of fluid depends upon the *weight* of the fluid being considered. In the case of water, a height of 23.1 ft of water causes a pressure in pounds per square inch of 10.0. In our work with fluid flow, we shall have frequent occasions in the case of water-piping systems to speak of "head in feet of water." Try to become as familiar with this expression of pressure as you now are with "pounds per square inch."

We shall also have occasion in connection with duct work to speak of pressure measured in *inches of water.* In this case, we are using the same basic pressure-measuring means as when we speak of head in feet of water. One in. of water = 1/12 ft, or 0.0833 ft of water. We speak of 1 in. of water simply because this is a more familiar measure than 0.833 ft of water. You already have all the following pressure equivalents. The tabulation below is presented for convenient reference:

1 psi = 2.31 ft of water = 0.493 in. Hg
1 oz per sq in. = 1/16 psi = 0.1443 ft of water = 1.73 in. of water
1.00 in. of water = 0.578 oz per sq in.
1.00 ft of water = 6.94 oz per sq in. = 0.434 psi

(See Tables 1 and 2, for detailed equivalents between ounces per square inch and inches of water.)

Analysis of Flow Due to a Static Head. Note that Fig. 4 is very similar to Fig. 3, except that the cap on the outlet pipe has been removed. A faucet or water tap has been added above the tank. Water is flowing out of the tank, and the faucet is adjusted so that water comes in at exactly the same rate at which water flows out at point B. Thus the water level in Fig. 4 is maintained at a height of 23.1 ft above the bottom of the tank. Let us now rewrite Eqs. (1) and (2).

$$V_s = \sqrt{2gh} \tag{1}$$

$$V = 60\sqrt{2gh} \tag{2}$$

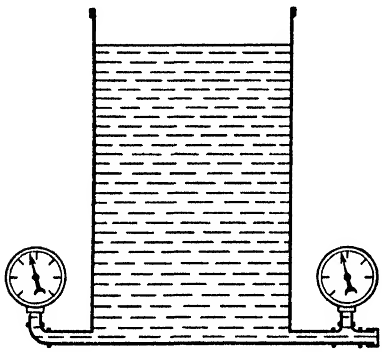


Fig. 3.—Gauges read static pressure.

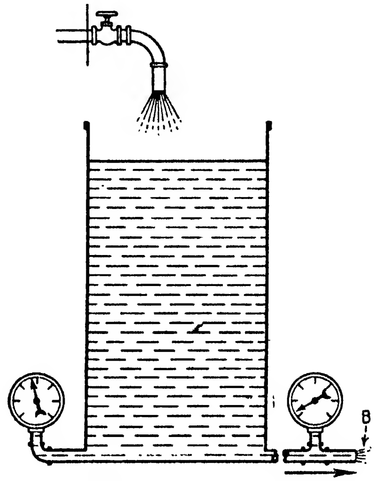


Fig. 4.—Left-hand gauge reads static pressure; difference in gauge readings is pressure drop due to flow.

If we assume that our outlet pipe is a nozzle having no friction, we may figure the velocity as follows: from Eq. (1),

$$V_s = \sqrt{2(32.2)(23.1)}$$

$$V_s = 38.56 \text{ ft per sec}$$

from Eq. (2),

$$V = 60\sqrt{2(32.2)(23.1)}$$

$$V = 2{,}314 \text{ fpm}$$

From the above you can see that we may calculate the velocity of the water leaving the outlet pipe *without* actually knowing or using the value of pressure in pounds per square inch. All we need is the *head in feet of water.* Similar calculations may be made with any other fluid than water, including air, oil, or mercury.

Compare the pressure gauges at the right in Figs. 3 and 4. In Fig. 3 the gauge shows a pressure equivalent to the full static pressure head of water. In Fig. 4 the pressure gauge shows no pressure at all. Why is the pressure indicated by the gauge different in Fig. 4 from that in Fig. 3? The reason is that in Fig. 4 we show conditions in which the full pressure head is being *used to cause flow, i.e.,* to give *energy of motion* to the water. In this case, the static pressure energy formerly existing is no longer present *in this form.* Instead in the outlet pipe the water contains all this energy *converted into* the form of *energy of motion* (or kinetic energy).

If you have trouble understanding the preceding paragraph, note point B just beyond (outside) the end of the outlet pipe in Fig. 4. Certainly at point B the water is not subject to any static or compressing pressure, for if it were it would expand or explode to an increased diameter upon leaving the pipe. Since the water is subject to no static pressure at point B (except atmospheric pressure), the same condition is true where the pressure gauge is connected, just inside the pipe a *short distance* from the outlet. The phenomenon illustrated in Fig. 4 has important applications to air flow in ducts as we shall begin to see on page 420. Before we reach these practical applications, we must understand two more types of pressure.

Velocity Pressure and Total Pressure. In the preceding section we explained that *static pressure* is converted (or changed) into *velocity pressure* of the flow occurring through an orifice. We have stated that the pressure energy appeared as kinetic energy, or energy of motion. Instead of thinking of this energy as kinetic energy, we may think of it as a new and different type of pressure. Suppose that you were to hold up an iron shield against which the stream of water from a fire hose were directed. As to *static* pressure, it is obvious that the stream of water is simply at atmospheric pressure; nevertheless you would probably be knocked down by the stream of water. Why not think of the force that knocked you down as being caused by the *velocity pressure* of this water. Water at the same static pressure, atmospheric, could be spilled slowly on a shield held by you without tending to knock you over. Only when the water strikes the shield at a *high speed* would you be knocked down. In other words, the *higher the speed* of moving flow; the *greater* its *velocity pressure.*

The preceding paragraph should give you a general understanding of velocity pressure. We may define velocity pressure as follows: Velocity pressure is a pressure corresponding to the velocity of the flow of a fluid, and is a measure of kinetic energy in the fluid. For the frictionless flow which we are assuming, velocity pressure is equivalent to the static pressure which would produce flow through a nozzle at the velocity which we have.

Before going further, make sure that you clearly understand the meaning of velocity pressure and static pressure. As soon as you understand these two types of pressure we may define our last type of pressure very simply by the following equation:

$$\text{Static pressure} + \text{velocity pressure} = \text{total pressure} \qquad (3)$$

$$\text{S.P.} + \text{V.P.} = \text{T.P.}$$

Equation (3) defines total pressure as an equation and gives you the customary abbreviations for the three types of pressure with which we have to deal. Bear in mind that static pressure *acts in all directions*, upward, downward, sideways, etc. Bear in mind also that static pressure can exist in fluids at rest and also in fluids in motion. If a fluid is in motion, under a definite static pressure, and if the static pressure acts in *all* directions, then the static pressure must *act in the same directions as the velocity pressure* as well as in all other directions.

We emphasized the above statement because we do not have a means of *measuring* velocity pressure directly. As you will see later in this chapter, we measure static pressure by a gauge inserted perpendicular to the direction of flow. Velocity pressure does not affect such a gauge; therefore it records only static pressure. We insert another gauge *in the direction of flow*, such that it measures the velocity pressure *plus* the static pressure, which is total pressure. Having measured total pressure and static pressure, we can easily obtain velocity pressure by the following transposed form of Eq. (3):

$$\text{V.P.} = \text{T.P.} - \text{S.P.} \tag{4}$$

Application to Flow of Air. Thus far in this chapter we have spoken of flow of fluids in general, illustrating our discussion with figures regarding the fluid *water*, which happens to be a liquid. All that we have stated is equally true of other fluids, including the fluid *air*, which happens to be a gas. We have not used illustrations regarding a column of air producing a pressure head of a certain number of feet of air, because it is hard to visualize a column of air. Now that you have become familiar with the principles of flow, based upon illustrations of a column of water which you can visualize, let us consider the flow of air.

Draw a sketch similar to Fig. 2, making your column have an area of 1 sq ft and a height of 1,920 ft. Let us now calculate the pressure due to the weight of this column of air. We shall assume that atmospheric pressure exists on the top and also at the bottom of the column of air. We shall also assume that 1 cu ft of air weighs the same at the top and at the bottom of the column. Finally, we must assume a specific volume for the air, let us take 13.33 cu ft per lb. From this obtain the weight per cubic foot of our air, which is

$$\frac{1}{13.33} = 0.075 \text{ lb per cu ft}$$

Then following the same method we used regarding Fig. 2, we have

$$1{,}920\ (0.075) = 144.0 \text{ lb}$$

Since this 144 lb of air is resting on a surface having an area of 144 sq in., we have

$$\frac{144}{144} = 1.00 \text{ psi}$$

From the above you can clearly see that while a column of water 2.31 ft high produces a pressure head of 1 psi at the bottom of the column, a head of 1,920 ft of air is equivalent to this same pressure of 1.00 psi.

In working with air flow through ducts, we never encounter pressure differences as great as 1 psi; pressures *much small than* 1 oz per sq in. are instead the rule. Let us determine the head in feet of air equivalent to a pressure of 1.0 oz per sq in. Since there are 16 oz in 1 lb, we have

$$\frac{1{,}920}{16} = 120 \text{ ft of air column}$$

Let us now determine from Eq. (2) the velocity that would be caused by a static pressure difference (or static pressure head) of 120 ft of air.

$$V = 60\sqrt{2gh} \tag{2}$$

$$V = 60\sqrt{2(32.2)(120)}$$

$$V = 60\sqrt{7{,}728}$$

$$V = 60(87.9)$$

$$V = 5{,}270 \text{ fpm}$$

From the preceding paragraph we see that if we have an orifice such as the simple arrangement shown in Fig. 1, the pressure in region A being 120 ft of air *greater than* the pressure in region B (a pressure difference of one ounce per square inch), flow will occur through the orifice at the surprisingly high velocity of 5,270 fpm. To take a more practical case than the simple orifice shown in Fig. 1, we can have fans blowing into a chamber called a "plenum chamber," as illustrated in Fig. 5. If the fans were able to produce a static pressure of 1 oz per sq in. inside the plenum chamber, air would leave the discharge nozzle at a velocity of 5,270 ft, as indicated in Fig. 5. This velocity is high when you realize that 5,280 fpm is 60 mph, and the highest duct air speed permitted in industrial applications is 3,000 to 4,000 fpm.

For the standard air that we assumed on page 418, having a specific volume of 13.33 cu ft per lb, we have seen that a head of 120 ft of air is equivalent to 1.0 oz per sq in. of pressure. As a matter of fact, we ordinarily measure static pressures of air, *not* in feet of air or in ounces per square inch, but in inches of water. We do this because the static pressures with which we deal in air-flow work are very small. We have information enabling us to convert pressure from any one of these units into the other

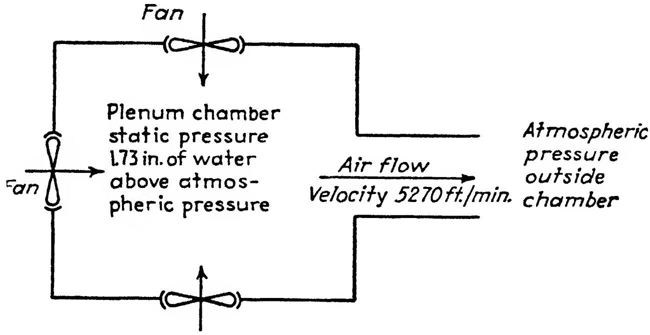


FIG. 5.—Effect of plenum chamber is to even out flow from fans.

two (see page 422). For your convenience we present Tables 1 and 2 which give this information, tabulated to the nearest tenth.

Measuring Static Pressure and Total Pressure in Ducts. Refer to Figs. 6 and 7, and note the glass water gauge at the extreme left in each figure. Figure 6 represents a supply-air duct, *i.e.*, a duct through which air

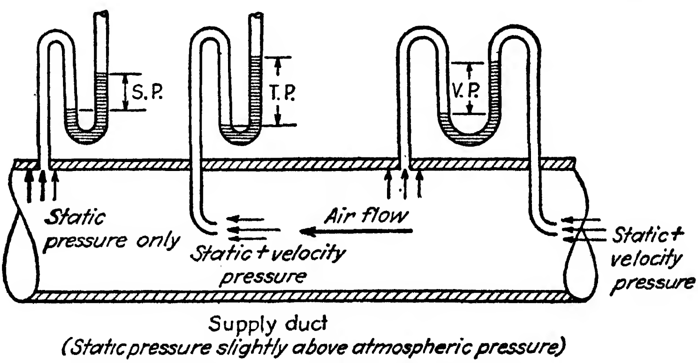


FIG. 6.—Static pressure at the end of an open duct is zero.

is *blown to* a conditioned space. The static pressure within such a duct is slightly *above* atmospheric pressure. Figure 7 represents an exhaust or recirculation duct through which air is *drawn from* a conditioned space. Such a duct is under a slight suction, *i.e.*, the static pressure inside the duct is slightly below atmospheric pressure.

Look at the U gauges to the extreme left in Figs. 6 and 7, note that atmospheric pressure is exerted downward, through the open end of the tube, upon the surface of the water. (Because clean water would be hard to see, we often put a drop of red dye in the water in such gauges.) The other leg of the U gauge is connected to the side of the duct, the connection being at *right angles* to the side of the duct. By means of this arrangement, *static pressure only* is exerted through this connection above the surface of the closed leg of the U gauge. The water column would be displaced as indicated in Figs. 6 and 7, giving a direct reading of the *difference* between atmospheric pressure and the *static* pressure inside the duct.

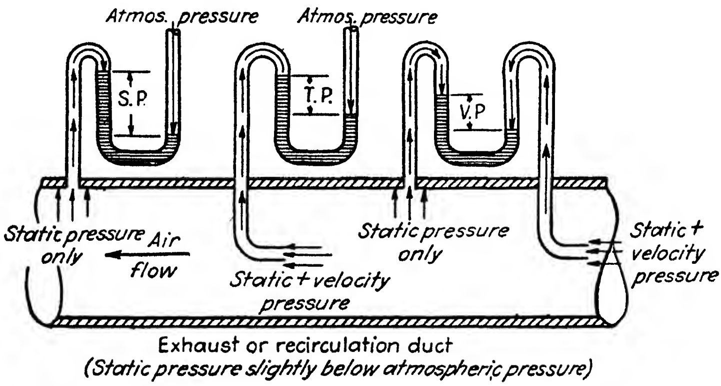


FIG. 7.—Static pressure readings increase from zero at the inlet to a maximum at the fan suction box.

Now study the middle U gauge in Figs. 6 and 7 (marked T.P.). Here again, atmospheric pressure is exerted upon the surface of the water through the open end of the U gauge. The closed end of the gauge does not merely connect to the side of the duct but extends inside the duct and turns so that the *opening faces directly into the air stream.* By this arrangement, the full impact of the velocity of the air exerts the *velocity pressure* of the air through this side of the U gauge. Since static pressure is exerted in all directions, it is also exerted through this side of the U gauge. Thus this gauge measures the difference between atmospheric pressure and *total* pressure inside the duct.

Measurement of Velocity Pressure. Refer now to the combination U gauges shown at the extreme right in Figs. 6 and 7. Note that these U gauges consist of an arrangement whereby we have an ordinary static pressure gauge, with the open end now *connected to the measuring jet that records total pressure.* Hence in the combined gauge, static pressure alone is exerted on the surface of the water in one leg. In the other leg, velocity pressure plus static pressure act upon the surface of the water. What is the unbalanced force or pressure in this case which will cause a displacement (or change in level) within the gauge? Since static pressure acts on the water surface in *both legs* of the gauge, and velocity pressure acts upon the surface in only one leg, the difference in levels is caused by *velocity pressure.* Thus this combined gauge reads velocity pressure directly. In other words, it performs for us the arithmetic of this operation.

$$V.P. = T.P. - S.P. \tag{4}$$

TABLE 1.—PRESSURE IN OUNCES PER SQUARE INCH CORRESPONDING TO INCHES OF WATER

Head, in.	Decimal parts of an inch									
	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0	....	0.06	0.12	0.17	0.23	0.29	0.35	0.40	0.46	0.52
1	0 58	0.63	0.69	0.75	0.81	0.87	0.93	0.98	1.04	1.09
2	1.16	1.21	1.27	1.33	1.39	1.44	1.50	1.56	1.62	1.67
3	1.73	1 79	1.85	1.91	1 96	2.02	2.08	2.14	2.19	2.25
4	2.31	2 37	2.42	2 48	2 54	2.60	2.66	2.72	2.77	2.83
5	2.89	2.94	3.00	3.06	3 12	3.18	3.24	3.29	3.35	3.41
6	3.47	3.52	3.58	3.64	3.70	3.75	3.81	3.87	3.92	3.98
7	4.04	4.10	4.16	4.22	4.28	4.34	4.39	4.45	4.50	4.56
8	4.62	4.67	4.73	4.79	4.85	4.91	4.97	5.03	5.08	5.14
9	5.20	5.26	5.31	5 37	5.42	5.48	5.54	5.60	5.66	5.72

TABLE 2.—PRESSURE IN INCHES OF WATER CORRESPONDING TO OUNCES PER SQUARE INCH

Pressure, oz per sq in.	Decimal parts of an ounce									
	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0	....	0.17	0.35	0.52	0.69	0.87	1.04	1.21	1.38	1.56
1	1.73	1.90	2.03	2.25	2.42	2.60	2.77	2.94	3.11	3.29
2	3.46	3.63	3.81	3.98	4.15	4.33	4.50	4.67	4.84	5.01
3	5.19	5.36	5.54	5.71	5.88	6.06	6.23	6.40	6.57	6.75
4	6.92	7.09	7.27	7.44	7 61	7.79	7.96	8.13	8.30	8.48
5	8.65	8.82	9.00	9.17	9.34	9.52	9.69	9.86	10.03	10.21
6	10.38	10.55	10.73	10.90	11.07	11.26	11.43	11.60	11.77	11.95
7	12.11	12.28	12.46	12.63	12.80	12.97	13.15	13.32	13.49	13.67
8	13.84	14.01	14.19	14.36	14.53	14.71	14.88	15.05	15.22	15.40
9	15.57	15.74	15.92	16.09	16.26	16.45	16.62	16.79	16.96	17.14

A gauge for measuring velocity pressure, of the type shown in Figs. 6 and 7, is practical only when it is to be *permanently* installed on a particular duct. For temporary use to take test readings, such a gauge would be impractical because it involves two holes in the duct and because it would be difficult to make a tight 90° connection as required for the static-pressure connection. A special instrument is built which may be inserted into the duct and which will record the difference between total pressure and static pressure. Figure 8 below illustrates this instrument, which is known as "Pitot tube" (pronounced Pee-toe). The Pitot tube consists of one tube sealed within another. The inner tube faces directly into the air stream

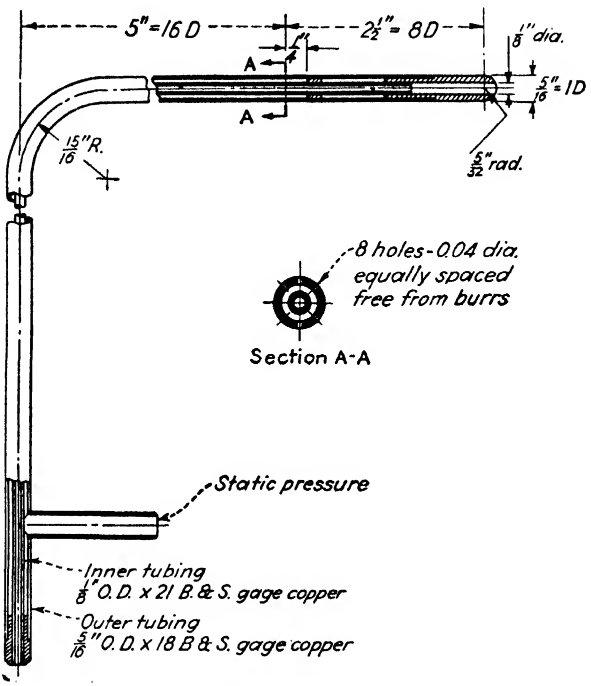


FIG. 8.—Connecting both velocity and static tubes reads velocity head on manometer.

and records total pressure. The outer tube has eight equally spaced holes drilled through it at a fixed distance back from the tip. These holes are perpendicular to the air flow and hence record static pressures. Connections are made by means of rubber hoses, from the end of the Pitot tube outside the duct, to the U gauge.

In extremely accurate test work, Pitot tubes are used to measure velocity pressure, although in most heating and ventilating work the velometer has the advantage of being much quicker to use for the same purpose. The velometer reads velocity directly in feet per minute. When the Pitot tube is used, readings are taken at various points inside the duct, at the same points as would be used with the long duct jet of the velometer.

While the Pitot tube is somewhat more accurate, its velocity-pressure readings must be converted by means of Eq. (10) (which appears later in this chapter) in order to get velocity in feet per minute.[1]

Flow Equation Based on Head in Inches of Water. In the form giving velocity in feet *per minute,* Eq. (2) was our flow equation based upon h = head in feet of the fluid concerned (see page 416).

$$V = 60\sqrt{2gh} \tag{2}$$

Applying this equation to the flow of air, h = head in feet of air. We now wish to obtain a new form of Eq. (2). In this new form we want to have inside our square root sign one letter, representing head *in inches of water.* We desire this new form as a practical measure because we usually measure static pressure of air in inches of water. We never measure it in the theoretical form "head in feet of air." Those not interested in the derivation of this new form of Eq. (2) may skip now to Eq. (10). We show how we arrive at Eq. (10) merely for those who are interested in the origin of this new equation.

Let H_w = head, inches of water
x = head, feet of water

then

$$\frac{H_w}{12} = x$$

The heads in feet of two different fluids, to produce the same pressure, are inversely proportionate to the density of the fluids, *i.e.*, the heavier the fluid, the less feet of head required to produce the same pressure. Hence we may write an inverse proportion thus:

$$\frac{h}{x} = \frac{62.4}{d} \tag{5}$$

where h = h of our Eq. (2), in this case *head in feet of air*
x = head equivalent to h, but *in feet of water*
62.4 = density of water, lb per cu ft
d = density of air, lb per cu ft

But, since $x = H_w/12$, we can substitute in Eq. (5) and get

$$\frac{h}{\frac{H_w}{12}} = \frac{62.4}{d} \tag{6}$$

[1] **Some Pitot tubes have characteristics such that a constant factor k must be multiplied by the right-hand side of Eq. (10). The correct equation is given with the instructions for any Pitot tube.**

From Eq. (6), we solve for h.

$$h = \frac{62.4}{12}\left(\frac{H_w}{d}\right) = 5.2\left(\frac{H_w}{d}\right) \tag{7}$$

We now take the value h from Eq. (7) and substitute it in Eq. (2)

$$V = 60\sqrt{2g(5.2)\left(\frac{H_w}{d}\right)} = 60\sqrt{2(32.2)(5.2)\left(\frac{H_w}{d}\right)} = 60\sqrt{335\left(\frac{H_w}{d}\right)}$$

or

$$V = 1{,}096.5\sqrt{\frac{H_w}{d}} \tag{8}$$

We know from Chap. VII, and from psychrometric work, that for air

$$\text{Density, lb per cu ft} = \frac{1}{\text{specific volume, cu ft per lb}}$$

or

$$d = 1/v$$

(letting v = specific volume. (Note that *this* is a *small* v.) Substituting $1/v$ for d in Eq. (8), we get

$$V = 1{,}096\sqrt{H_w(v)} \tag{9}$$

Standard air in fan and duct problems is taken as 70°F, 29.92 in. Hg abs pressure, for which $d = 0.75$ lb per cu ft and $v = 13.3$ cu ft per lb. For standard air, we can get a simple expression for V from either Eqs. (8) or (9).

$$V = 1{,}096.5\sqrt{\frac{H_w}{0.75}} \quad \text{or} \quad V = 1{,}096.5\sqrt{13.3(H_w)}$$

Either of the above results in the important equation (10) given below, which is simply another form of Eq. (2).

$$V = 4{,}005\sqrt{H_w} \tag{10}$$

where V = velocity of air, fpm
H_w = pressure, inches of water

Friction—A Cause of Pressure Loss in Fluid Flow. In the discussion of fluid flow we have so far neglected the effect of friction. We have spoken only of the static pressure necessary to impart to a fluid the necessary kinetic energy to bring the fluid up to a certain velocity from an initial condition of zero velocity. If there were no such thing as friction, heating, ventilating, and air-conditioning fans would consume very little power indeed, because only a little static pressure is required to impart a considerable velocity to air. Friction plays the same part in water flow through pipes and in air flow through ducts.

Probably you have had the experience of connecting about 100 ft of garden hose to a water faucet and discovering that there is much less pressure at the end of the hose than that originally existing at the faucet. As a matter of fact, if 500 ft of garden hose were connected to an ordinary water faucet, the water would merely trickle out of the end of the hose, indicating very slight static pressure near the end. This static pressure has been "lost" in overcoming friction. Actually, the static pressure is *converted into*

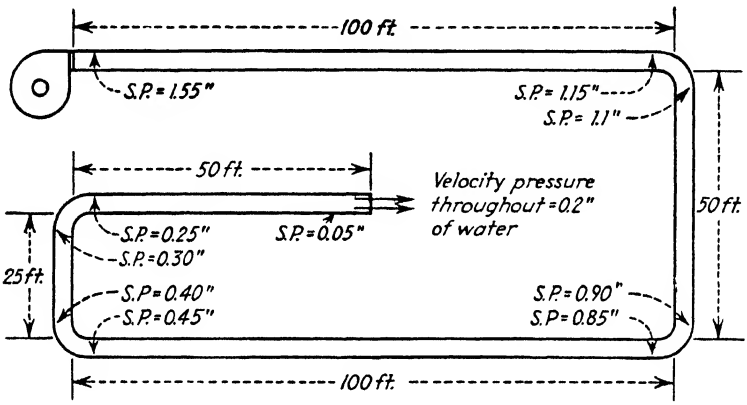


FIG. 9.—Velocity pressure throughout is 0.2 in. water gauge because duct is of constant cross-section.

velocity energy continuously, to make up for the velocity energy that is destroyed by the friction of the flowing fluid against the walls of the pipe or duct.

Consider the duct shown in Fig. 9. This winding duct, 325 ft long, is impractical and is presented for illustration only. Note that the duct in Fig. 9 is a round duct 14 in. in diameter. The air velocity is the same throughout the duct and is 1,800 fpm. From Eq. (10) the result is a velocity pressure throughout the duct in Fig. 9 of 0.20 in. of water.

You do not yet have the design information to determine friction loss in this duct. (Chap. XXIV deals with duct design.) However, for each 100 ft of length of *straight* duct in Fig. 9, there is a static pressure loss of 0.4 in. of water. The static pressure inside the duct in Fig. 9 is shown at the end of each straight run of duct. Figure 9 assumes a static-pressure drop of 0.50 in. required to force air through the grille at the end of the duct. The static-pressure loss caused by each elbow or turn is also shown in Fig. 9.

Impact—Another Cause of Loss of Static Pressure. In fluid flow any change of direction results in an impact as the air strikes against the duct and has its direction changed. Also any piece of apparatus, such as a bank of filters, extended surface coils, or air-washer spray eliminators, introduces an obstacle in the flow of fluid, and an impact pressure loss results. Finally, changes in duct sizes resulting in changes in velocity of the air cause a change in static pressure. We shall now consider the three types of pressure loss.

Elbows and Turns. In general, the "sharper" the turn—in other words the shorter the radius of the turn—the greater the pressure loss caused by elbows. In Chap. XXIV you will find accurate information for calculating the pressure loss caused by elbows and turns. The pressure loss caused by elbows is not given in terms of static pressure because it depends upon the velocity at which the fluid is flowing. For elbows in water piping the resistance of an elbow (or valve, or other fitting) is given by stating that such an elbow has the same resistance *as a certain number of feet of straight pipe.* In the case of duct elbows, charts are available which give the pressure loss caused by the elbow in *per cent of the velocity head* of the air flowing in the duct (see Fig. 9). The elbows in Fig. 9 are selected such that they cause a static-pressure loss equal to 25 per cent of the velocity head of the air flowing in the duct. (Do not try to understand why this figure is "25 per cent" until Chap. XXIII.)

In Fig. 9, the velocity head is 0.20 in.; 25 per cent of this velocity head is then 0.05 in. Study the static pressures shown on each side of the elbows in Fig. 9, and you will see that a *static pressure loss* of 0.05 in. is shown for each elbow. From this it would seem as though we would be required to calculate the velocity head in inches of water, based upon the actual velocity of our air, every time we wish to figure the static pressure loss caused by an elbow. Instead, however, we find it convenient to calculate these velocity heads once and for all and to present the values in tabular form. Table 3 gives this information in even steps of pressure. Table 4 gives the same information in *even steps of air velocity.* Table 4 does not show such high velocities as Table 3, but it does show considerably lower velocities. Table 4 covers the useful range of air velocities for heating, ventilating, and air-conditioning work. *Tables 3 and 4 are based upon dry air at 70°F and a barometric pressure of 29.92 in. Hg.*

Apparatus and Equipment. Coils, filters, louvers, air washers, etc., all cause static-pressure losses because of impact and friction. These losses can be given in terms of per cent of velocity-head loss as is done with elbows. The static-pressure drop could then be determined from Table 4 as is done with elbows. However, manufacturers usually present this information directly in *static drop in inches of water,* stating the information for various air velocities. As an example, see the graph on page 33.

TABLE 3.—CORRESPONDING PRESSURES AND VELOCITIES OF DRY AIR AT 70°F AND 29.92 IN. HG BAROMETRIC PRESSURE

Inches of water	Ounces per sq in.	Velocity, fpm	Inches of water	Ounces per sq in.	Velocity, fpm
0.05	0.0289	896	1.73	1.0000	5,273
0.10	0.0577	1,266	1.75	1 0092	5,298
0.20	0.1154	1,791	2.00	1.1535	5,664
0.25	0.1443	2,003	2.17	1.2500	5,895
0.30	0.1730	2,193	2.25	1.2975	6,007
0.40	0.2308	2,533	2.50	1.4418	6,332
0.43	0.2500	2,637	2.60	1.5000	6,457
0.50	0.2884	2,832	2.75	1.5860	6,641
0.60	0.3460	3,102	3.00	1.7300	6,937
0.70	0.4037	3,351	3.03	1.7500	6,976
0.75	0.4326	3,468	3.25	1.8740	7,220
0.80	0.4614	3,582	3.47	2.0000	7,457
0.87	0.5000	3,729	3.50	2.0185	7,492
0.90	0.5190	3,800	3.75	2.1630	7,756
1.00	0.5768	4,005	3.90	2.2500	7,910
1.25	0.7209	4,478	4.00	2.3070	8,010
1.30	0.7500	4,566			
0.50	0.8650	4,905			

TABLE 4.—CORRESPONDING PRESSURES AND VELOCITIES OF DRY AIR AT 70°F AND 29.92 IN. HG BAROMETRIC PRESSURE

Velocity, fpm	Corresponding velocity head, in. of water	Velocity, fpm	Corresponding velocity head, in. of water
300	0.0056	1,200	0.090
400	0.0100	1,300	0.105
500	0.0155	1,400	0.122
600	0.0225	1,500	0.140
700	0.0302	1,600	0.160
800	0.040	1,700	0.180
900	0.050	1,800	0.202
1,000	0.061	1,900	0.225
1,100	0.073	2,000	0.250

Reduction in Duct Area. Increase in Velocity. If the size of a duct or pipe is reduced, the velocity of the fluid must increase because of the smaller area of pipe or duct. This increase in the velocity of the flow means an increase in the velocity heat, the amount of which can be calculated by means of Eq. (10) or taken directly from Table 4, if we know the velocities in the large and small ducts.

Refer to the bottom sketch in Fig. 10. Duct construction when a reduction in area is required should be made in accordance with the note on the lower sketch in Fig. 10. Such a construction is not often encoun-

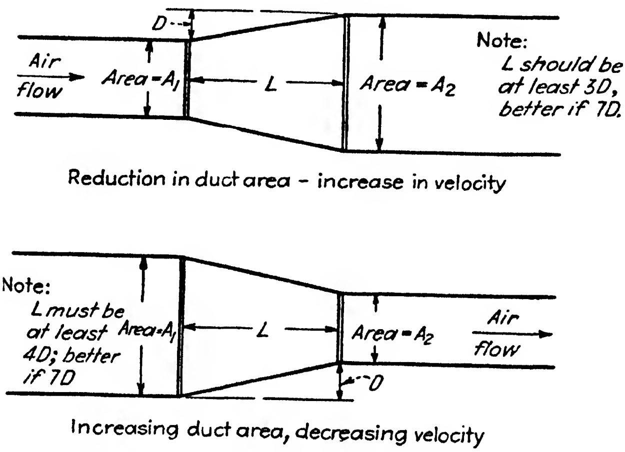


FIG. 10.—Reduction and increase must be gradual to reduce friction losses due to turbulence.

tered, but when it is, it is important that the slope of the transformation piece be as gradual as possible. If the change in duct area is made too abrupt, there will be an additional pressure loss over and above the static pressure loss necessary to increase the velocity of the fluid.

In the case of water piping, reduction fittings having gradual slopes are available in a few cases, but ordinarily standard fittings cause a rather abrupt change in the piping area. The pressure loss caused by such fittings is obtained, in terms of the length of straight pipe which would cause the same resistance, from tables that will be found in Chap. XXIV.

More frequently in air-conditioning duct work we have occasion to *increase the area* of the duct in order to *decrease the velocity* of the air. Such a construction is illustrated in the upper sketch in Fig. 10. The most frequent use of this arrangement is at the outlet of a fan, in which case the fan outlet would be the area marked A_1. With this type of construction, theoretically the static pressure would really *increase* in the larger duct, the increase being equivalent to a certain percentage of the velocity-head decrease. The velocity head will decrease somewhat because of the slower velocity in the larger duct. Theoretically, 30 to 70 per cent of this decrease is converted into static head, causing a slight *increase* in the static head. In duct-design work, a figure of 50 per cent "static regain" may be taken for construction of this type, provided the relation between dimensions D and L are held to the specifications given on the upper sketch in Fig. 10.

Such static regains are very small at the velocities used in ordinary duct work. We introduce a slight safety factor if we neglect them entirely. We recommend for installations where the highest velocity does not exceed 1,500 fpm that static regain due to decrease in velocity be neglected. This will ordinarily result in the calculated static pressure being a few per cent too high, which is a good way of introducing a small safety factor.

Flow of Refrigerants, Liquid and Gaseous. Throughout this chapter we have mentioned the flow of water in pipes and the flow of air in ducts, both being practical examples to which we must apply our knowledge of fluid flow. How about a refrigerant gas in the suction line in the discharge line from the compressor to the condenser? Actually, this is the flow of a gas through a pipe or "duct," and the theories of fluid flow apply similarly. The liquid line from the condenser or liquid receiver to the expansion valve presents a problem similar to that of a water flowing through a pipe.

Because we have occasion to encounter different refrigerants, each of which has properties quite different from air or water, and because most of these refrigerants are under pressures much different from atmospheric pressure, we do not design refrigerant piping systems according to the procedures that we shall use in Chap. XXIV for the design of duct systems and water-piping systems. Instead, we shall at the end of Chap. XXIV present accepted standards for the sizing of suction and liquid lines for Freon refrigeration as applied to air conditioning.

Conclusion. We have seen that a certain energy input is required in order to give air or any other fluid the kinetic energy or velocity head required to set it in motion or start flow. Except for the now obsolete gravity warm-air furnace and gravity circulation hot-water heating systems, this velocity head is imparted to the fluid *by a fan or pump.*

In the latter portion of this chapter we have seen that certain friction and impact losses occur throughout the duct or piping systems. These losses may be expressed as static pressure drops, *in inches of water for the flow of air.* For *water flow* these static pressure drops are usually expressed directly as head loss *in feet of water.* The fan or pump also serves to produce the necessary static pressure to overcome these friction and impact losses.

TABLE 5.—FOR PITOT-TUBE TEST, VELOCITY PRESSURES AND CORRESPONDING VELOCITIES

Formula: $V = 4005\sqrt{p}$
where V = velocity, fpm
p = velocity pressure, inches of water

Velocity pressure, in. H_2O	Velocity, fpm	Velocity pressure, in. H_2O	Velocity, fpm	Velocity pressure, in. H_2O	Velocity, fpm	Velocity pressure, in H_2O	Velocity, fpm
0.005	282	0.205	1,810	0.405	2,550	0.605	3,110
0.01	400	0.21	1,830	0.41	2,560	0.61	3,120
0.015	490	0.215	1,850	0.415	2,580	0.615	3,135
0.02	565	0.22	1,870	0.42	2,595	0.62	3,150
0.025	632	0.225	1,985	0.425	2,615	0.625	3,160
0.03	690	0.23	1,920	0.43	2,625	0.63	3,175
0.035	750	0.235	1,935	0.435	2,640	0.635	3,185
0.04	800	0.24	1,960	0.44	2,660	0.64	3,200
0.045	850	0.245	1,980	0.445	2,675	0.645	3,215
0.05	895	0.25	2,000	0.45	2,685	0.65	3,220
0.55	935	0.255	2,020	0.455	2,700	0.655	3,235
0.06	980	0.26	2,040	0.46	2,720	0.66	3,250
0.065	1,020	0.265	2,060	0.465	2,735	0.665	3,260
0.07	1,060	0.27	2,080	0.47	2,745	0.67	3,280
0.075	1,100	0.275	2,095	0.475	2,760	0.675	3,290
0.08	1,135	0.28	2,115	0.48	2,775	0.68	3,300
0.085	1,170	0.285	2,135	0.485	2,790	0.685	3,310
0.09	1,200	0.29	2,150	0.49	2,800	0.69	3,325
0.095	1,230	0.295	2,170	0.495	2,820	0.695	3,340
0.100	1,270	0.300	2,190	0.500	2,835	0.700	3,350
0.105	1,300	0.305	2,205	0.505	2,840	0.705	3,360
0.11	1,330	0.31	2,220	0.51	2,850	0.71	3,370
0.115	1,360	0.315	2,245	0.515	2,870	0.715	3,380
0.12	1,385	0.32	2,260	0.52	2,880	0.72	3,395
0.125	1,410	0.325	2,280	0.525	2,900	0.725	3,410
0.13	1,440	0.33	2,295	0.53	2,910	0.73	3,420
0.135	1,470	0.335	2,310	0.535	2,925	0.735	3,430
0.14	1,500	0.34	2,330	0.54	2,940	0.74	3,440
0.145	1,520	0.345	2,360	0.545	2,950	0.745	3,450
0.15	1,550	0.35	2,370	0.55	2,960	0.75	3,465
0.155	1,570	0.355	2,385	0.555	2,980	0.755	3,475
0.16	1,600	0.36	2,400	0.56	2,990	0.76	3,490
0.165	1,625	0.365	2,420	0.565	3,000	0.765	3,500
0.17	1,650	0.37	2,440	0.57	3,020	0.77	3,510
0.175	1,675	0.375	2,455	0.575	3,035	0.775	3,520
0.18	1,700	0.38	2,465	0.58	3,045	0.78	3,535
0.185	1,720	0.385	2,480	0.585	3,060	0.785	3,545
0.19	1,740	0.39	2,500	0.59	3,070	0.79	3,555
0.195	1,770	0.395	2,520	0.595	3,080	0.795	3,570
0.200	1,790	0.400	2,535	0.600	3,100	0.800	3,580

CHAPTER XXIII

FANS AND PUMPS

General. The problem of pump selection is less pressing than that of fan selection for two reasons. (1) In air-conditioning work we have occasion to use only a few of the many types of pumps that are manufactured, while in the case of fans, we have a choice varying from those with a few hundred to many thousands of cubic feet per minute capacity. Furthermore, in the case of fans we can choose units of different shapes and characteristics, depending upon the installation, while in pumps we are ordinarily

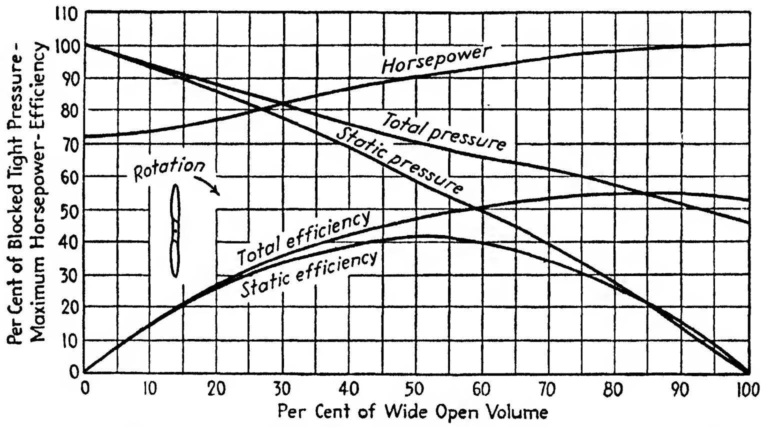


FIG. 1.—Operating characteristics of an airplane propeller fan. (*Courtesy of A.S.H.V.E. "Guide."*)

limited to choice among only two or three different pumps for a given job. (2) In air-conditioning work we use fans on *every installation*, while we use pumps only occasionally. This means that we must try to become sufficiently familiar with fan selection to be able to choose fans without the help of the fan manufacturers' engineers. In pump selection, on the other hand, many air-conditioning men simply give their requirements to the pump manufacturer and request him to select the pump.

Axial-flow or Propeller-type Fans. Except in cooling towers and evaporative condensers, axial-flow fans are rarely used in air-conditioning work. Figure 1 illustrates an axial-flow fan. The name "axial flow" is used because the air flow is along the axis or "axle" of the fan. The use of the name "propeller type" for this type of fan is self-explanatory.

When axial-flow fans have airplane-propeller type blades (two, three, or four blades per fan), the operating characteristics are somewhat different from axial-flow fans having three to eight or more *wide blades* not shaped like airplane-propeller blades. In either case, however, the efficiency is not good when the fan is operating at a capacity (or air volume) of less than 50 per cent of the "wide-open" volume. "Wide-open" volume means the volume of air that the fan would handle against *no static* pressure at all, *i.e.*, not connected to any duct work or apparatus.

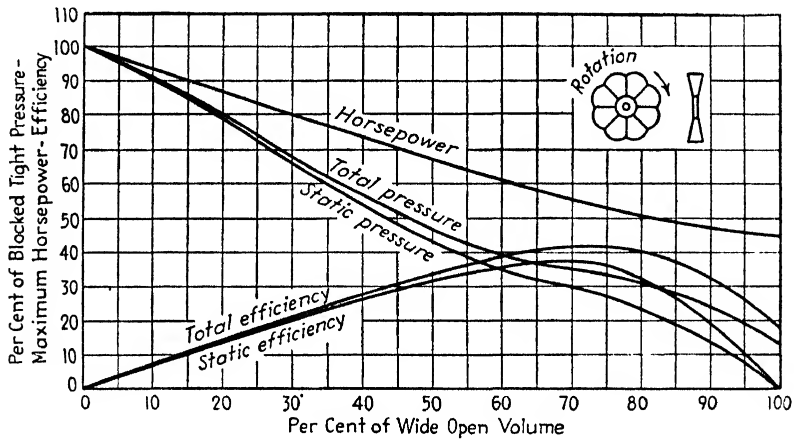


FIG. 2.—Operating characteristics of an axial flow fan. (*Courtesy of A.S.H.V.E. "Guide."*)

Figure 1 illustrates a typical performance curve of an axial-flow fan having the airplane-propeller type blades. Notice in Fig. 1 that the horsepower does not change greatly, but that the efficiency is less than 50 per cent for air volumes below 60 per cent of wide-open air volume. Figure 2 gives similar curves for an axial-flow fan of the wide-blade type. Note the difference in the horsepower characteristics as shown in Fig. 2, and note that the total efficiency is even poorer for this type of fan.

The B. F. Sturtevant Company has developed a special axial-flow fan. These fans have efficiencies above 70 per cent at capacities between 70 and 95 per cent of wide-open volume. Compare the characteristic curves shown on Figs. 2 and 3.

Centrifugal Fans. Almost all air-conditioning fans are of the centrifugal type. Centrifugal fans operate upon the same principle that permits you to take a small pail full of water in your hand and whirl the pail up over your head and down again without spilling any water. If you were to do such a stunt with a pail having a hole in the bottom and if you were to whirl the pail rapidly, water would squirt out of the bottom of the pail *even when the pail was upside down* (above your head). A centrifugal fan entraps air between blades that extend a part of the way from the outside edge of a fan wheel toward the center. This air is thrown outward by centrifugal force at *all points* around the fan wheel. Since we do not wish to deliver air in all directions around a fan, a scroll casing or "housing" is provided around the fan wheel, so that the air thrown out by the wheel may be "caught" and directed toward a discharge opening, which we call the "fan outlet." The air enters a centrifugal fan from the side, *parallel* to the fan axis, and is discharged perpendicularly to the fan axis.

The scroll casing or fan housing of a centrifugal fan also serves another purpose. The air thrown off from between the blades is at a higher static pressure than the air that entered the fan, but its primary energy content is *velocity energy*. This means that its velocity pressure is high because it is traveling very fast. The scroll case of the fan has an increasing cross-sectional area, so that the air *slows up* by the time it reaches the fan outlet. Thus a portion of the velocity pressure of the air, at the time it leaves the fan blades, is converted into static pressure by the time the air reaches the fan outlet.

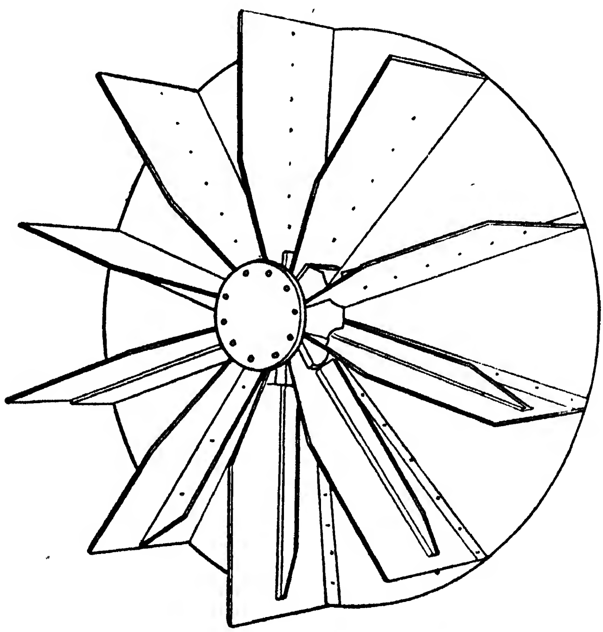


FIG. 3.—Construction of straight-blade fan.

Straight-blade or Radial-blade Fans. Radial-blade centrifugal fans are not used at all in air-conditioning or ventilating work at the present time. You may encounter radial-blade fans in a building having a ventilating system many years old. Radial-blade fans are used for certain types of furnace blowers and some types of factory exhaust fans where sawdust or other light materials pass through the fan. In radial-blade fans, the blades are *straight* blades lying along the *radii of a circle*, as you look at the fan wheel from the side view. Figure 3 illustrates one type of radial-blade fan showing blades with the housing removed.

Forward-curved-blade Centrifugal Fans. Forward-curved-blade fans of the multiblade or multivane types have been and are still used in some air-conditioning and ventilating installations. These fans are, however, being replaced in modern applications by fans of the backward-curved-

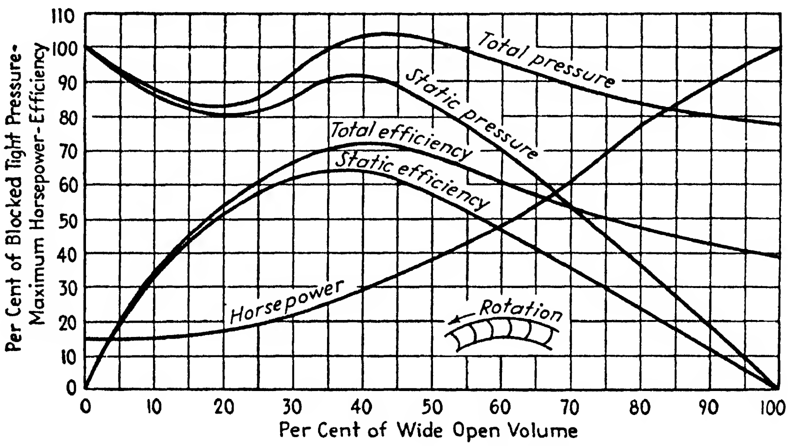


FIG. 4.—Operating characteristics of a fan with blades curved forward. (*Courtesy of A.S.H.V.E. "Guide."*)

blade type which we shall discuss in the next section of this chapter. The forward-curved-blade fans have two general disadvantages, compared with a properly chosen backward-curved-blade fan operating under the same operating conditions, in almost all ventilating and air-conditioning applications. The first is that these fans are usually somewhat *more noisy*, in spite of the fact that the *tip speed* of this type may be lower than that of the backward-curved-blade type. The other disadvantage is that the horsepower of a forward-curved-blade fan rises steeply as duct and apparatus resistance is reduced. Under these conditions, wide-open volume operation is approached with a great increase in cubic feet per minute and in horsepower. Characteristic curves of a typical forward-curved-blade fan are given in Fig. 4.

There is some doubt among fan engineers and fan manufacturers as to the relative operating noise level of forward-curved and backward-curved-blade fans. However, many members of The National Association of Fan Manufacturers consistently recommend the backward-curved-blade fan from the standpoint of noise. Aside from sound-level characteristics, the horsepower and pressure characteristics of forward-curved-blade fans are often a serious disadvantage in air-conditioning work. Frequently the static resistance of a system may be 1¼ to 1½ in. under certain conditions, but when by-passing the conditioning unit or air washer for some reason, the static resistance may drop as low as ½ to ¾ in. of water. A forward-curved-blade fan will handle a greatly increased amount of air *with a great increase in the horsepower required.* This might mean a serious overload of the motor driving the fan, and one of two steps would have to be taken in order to permit operation. Either the fan speed would have to be decreased, or a damper would have to be inserted to increase artificially the static resistance of the duct system.

Sometimes a conditioning unit of the factory-assembled type will contain forward-curved-blade fans. Operation will be satisfactory provided this unit is installed with the V-belt-drive arrangement to give the proper fan speed and provided the duct resistance is accurately known. Let us suppose that such a unit is installed, with another fan flowing into or drawing through the unit. You might think that the auxiliary fan would "help" the fan in the unit, decreasing its load. On the contrary, the effect of the auxiliary fan is to reduce the static pressure against which the unit fan must operate. Under these conditions, if the fan in the unit is operating at a speed intended to overcome the higher static resistance, you can readily see that the fan in the unit will be operating under a lower static pressure than that for which it has been selected. From the static-pressure and horsepower curves in Fig. 4, you can see that the reduction in static pressure will cause the fan to operate nearer wide-open volume conditions *and horsepower will increase.* This would require either a larger motor or a decrease in speed of the forward-curved-blade fan.

Backward-curved-blade Fans. As we have said, centrifugal fans having backward-curved blades are the most common in air-conditioning work, except for small multiblade fans built into some factory-assembled units. Clarage type W fans, Sturtevant Silentvane fans, and Buffalo Forge Company type CL Conodial fans are centrifugal fans of the backward-curved-blade type. Figure 5 illustrates curves giving the operating characteristics of a typical backward-curved-blade fan. Note that the efficiency is above 70 per cent at any capacity from 45 to 75 per cent of wide-open air volume. Hence a backward-curved-blade fan can always be sure of operation at high efficiencies, because a fan may be selected from the wide range of sizes available such that it will always operate in this efficient range. Note also in Fig. 5 that the horsepower curve rises to a peak between 60 and 70 per cent of wide-open volume and *decreases beyond* 70 *per cent of wide-open volume.*

This characteristic of backward-curved-blade fans has caused them to be called "limit load" or "nonoverloading" fans. For these fans at any given speed in rpm, there is a certain maximum horsepower which the fan can consume regardless of the static pressure against which it is operating. This maximum horsepower corresponds to the peak of the horsepower curve in Fig. 5. When selecting fans, it is always possible to determine just what the maximum horsepower will be. This means that if desired a motor may be provided that could not be overloaded, regardless of the static-pressure conditions at which the fan is to operate. The B. F. Sturtevant Company and the Clarage Fan Company give the limit load

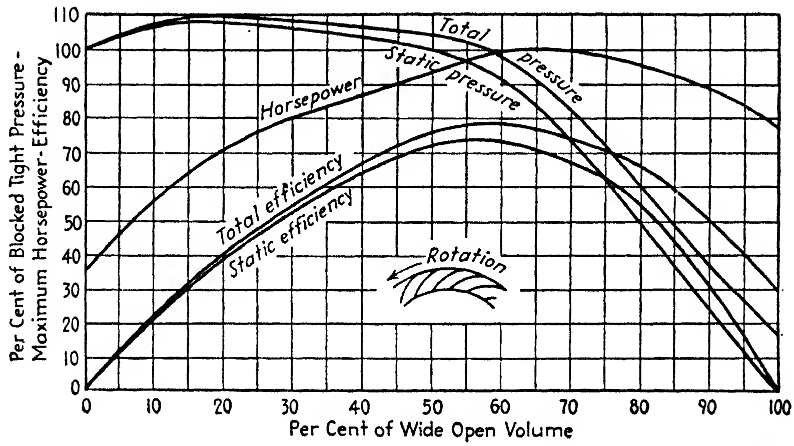


FIG. 5.—Operating characteristics of a fan with blades curved backward. (*Courtesy of A.S.H.V.E. "Guide."*)

or maximum horsepower in their catalogues for each size of fan by a formula of the following type:

$$\text{Maximum brake horsepower} = k\left(\frac{\text{rpm}}{c}\right)^3$$

where c = a constant such as 500 or 1,000 which will be given in the manufacturers literature

k = a constant factor depending upon the size of the fan and which may vary from less than 1.0 upward *depending upon the size.* This number is also given in the manufacturer's literature

In the rating tables of the limit-load Conodial (type CL) fans of the Buffalo Forge Company, the limit-load horsepower values for various fan speeds is calculated.

Backward-curved-blade fans have the additional advantage of operating at higher rpm than forward-curved-blade fans, which means that a higher speed, less expensive electric motor may be used. This operation at higher speed would be productive of noise, which would be objectionable, except that the operating characteristics of the backward-curved blade is such that operation is very quiet for properly selected fans in spite of a high tip speed.

Fan Laws. Those interested in a complete detailed discussion of *all* the variations of the 15 operation laws that apply to centrifugal fans in general are referred to "Fan Engineering,"[1] a valuable handbook that embraces heating, ventilating, air conditioning, and general information, in addition to fan information. A few simple fan laws should be understood by everyone connected with the handling of air in ventilation and air conditioning. The following laws apply to a fan of a *fixed* size, connected to a duct system having static resistance as described in each case:

1. If by means of dampers the static pressure of a duct system is adjusted so that it remains constant,

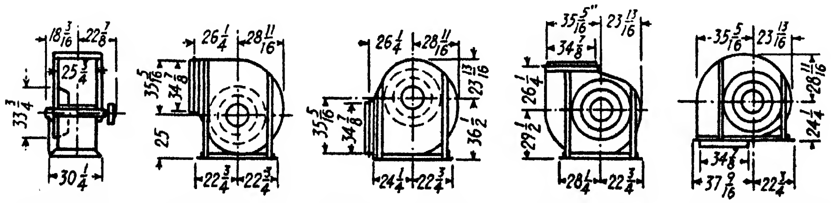
Capacity varies directly with rpm
Horsepower varies in proportion to rpm^3

2. If a fan of fixed size is installed in a certain duct system and you contemplate changing the duct system so that the static pressure will change, still desiring the same amount of air to be handled, the following is true:

Rpm varies as $\sqrt{\text{pressure}}$.
Horsepower varies as $\sqrt{(\text{pressure})^3}$

3. Given a fan of fixed size installed in a certain duct system, the fan to operate at constant speed, we sometimes find the resistance of the duct changing because of changes in filter resistance, damper setting, use of by-passes, etc., under this condition the following is true:

Capacity decreases as pressure increases, the amount depending on the curves (as shown in Figs. 4 and 5) for the particular fan.



Single Width | Top Horizontal | Bottom Horizontal | Up Discharge | Down Discharge

[1] Section 4, pp. 189–234, Buffalo Forge Company, Buffalo, N. Y.

TABLE 1.—BUFFALO LIMIT-LOAD CONOIDAL FAN (TYPE CL), SIZE 5½

Capacities and Static Pressures of Single Width at 70°F and 29.92 in. Barometer

Outlet velocity fpm	Capacity, cu ft of air per min	Static pressure, in.																Limit load	
		¼		⅜		½		⅝		¾		⅞		1		1¼			
		Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp
800	4,890	304	0.292	346	0.408	384	0.534	419	0.669	453	0.807	481	0.950	511	1.10	563	1.40	300	0.285
900	5,501	323	0.350	360	0.474	397	0.609	430	0.753	462	0.902	491	1.06	519	1.21	571	1.54	330	0.380
1,000	6,113	341	0.419	377	0.551	410	0.693	442	0.846	472	1.00	502	1.17	529	1.34	580	1.69	360	0.493
1,100	6,724	362	0.500	395	0.642	426	0.790	456	0.950	485	1.12	513	1.29	540	1.47	590	1.84	390	0.628
1,200	7,335	383	0.586	414	0.744	443	0.901	470	1.07	499	1.24	526	1.43	551	1.61	600	2.00	420	0.782
1,300	7,947	404	0.693	433	0.855	460	1.03	488	1.20	513	1.38	539	1.57	565	1.77	611	2.18	450	0.965
1,400	8,558	425	0.812	454	0.980	480	1.16	506	1.35	529	1.54	554	1.74	579	1.94	624	2.36	480	1.17
1,500	9,169	447	0.942	475	1.12	500	1.32	525	1.52	547	1.71	571	1.92	594	2.13	637	2.57	510	1.40
1,600	9,781	469	1.09	496	1.29	521	1.48	544	1.69	566	1.90	588	2.12	610	2.33	651	2.78	540	1.66
1,700	10,392	492	1.25	518	1.47	542	1.67	564	1.89	585	2.11	606	2.34	627	2.56	667	3.03	570	1.96
1,800	11,003	515	1.44	540	1.66	563	1.87	584	2.10	605	2.33	625	2.57	645	2.81	684	3.29	600	2.28
1,900	11,615	538	1.63	562	1.87	584	2.10	605	2.34	625	2.58	644	2.82	664	3.07	702	3.58	630	2.64
2,000	12,226	562	1.83	584	2.10	605	2.34	626	2.59	645	2.83	664	3.09	684	3.35	720	3.88	660	3.05
2,100	12,837	...	...	606	2.35	627	2.61	648	2.87	666	3.12	685	3.39	704	3.66	738	4.22	690	3.48
2,200	13,449	...	...	628	2.64	649	2.89	670	3.16	687	3.44	706	3.71	724	3.98	757	4.56	710	3.78
2,300	14,060	...	...	652	2.94	671	3.19	692	3.48	708	2.78	727	4.05	745	4.33	776	4.94	740	4.28
2,400	14,671	...	...	676	3.24	694	3.53	714	3.83	730	4.13	748	4.42	766	4.70	796	5.33	770	4.83
2,500	15,283	...	...	...	...	718	3.91	736	4.19	753	4.51	770	4.81	787	5.11	817	5.74	800	5.41

TABLE 1.—BUFFALO LIMIT-LOAD CONOIDAL FAN (TYPE CL), SIZE 5½—(*Continued*)

Outlet velocity, fpm	Capacity, cu ft of air per min	Static pressure, in.																Limit load	
		1½		2		2½		3		3½		4		4½		5			
		Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp
1,200	7,335	647	2.41	727	3.26	802	4.15	870	5.09	932	6.07	*988*	*7.09*	*1,043*	*8.16*	*1,092*	*9.24*	*860*	*6.05*
1,300	7,947	657	2.60	736	3.50	811	4.44	878	5.41	940	6.42	*995*	*7.48*	*1,051*	*8.60*	*1,100*	*9.73*	*860*	*6.73*
1,400	8,588	668	2.81	746	3.75	820	4.73	887	5.75	949	6.80	*1,003*	*7.88*	*1,059*	*9.04*	*1,108*	*10.2*	*890*	*7.46*
1,500	9,169	680	3.03	756	4.01	829	5.04	896	6.10	958	7.19	*1,011*	*8.31*	*1,068*	*9.50*	*1,117*	*10.7*	*920*	*8.25*
1,600	9,781	692	3.27	767	4.27	839	5.34	905	6.45	967	7.60	*1,019*	*8.76*	*1,077*	*9.97*	*1,126*	*11.2*	*950*	*9.07*
1,700	10,392	705	3.52	779	4.57	850	5.67	915	6.83	975	8.02	*1,028*	*9.23*	*1,086*	*10.5*	*1,135*	*11.7*	*980*	*9.97*
1,800	11,003	720	3.80	791	4.88	862	6.02	925	7.21	986	8.44	*1,038*	*9.71*	*1,095*	*11.0*	*1,144*	*12.3*	*1,010*	*10.9*
1,900	11,615	735	4.09	806	5.21	874	6.39	935	7.60	996	8.86	*1,048*	*10.2*	*1,104*	*11.5*	*1,153*	*12.9*	*1,040*	*11.9*
2,000	12,226	752	4.42	821	5.55	886	6.77	947	8.03	1,007	9.34	*1,058*	*10.7*	*1,114*	*12.0*	*1,162*	*13.5*	*1,070*	*12.9*
2,100	12,837	770	4.77	837	5.93	899	7.17	960	8.47	*1,018*	*9.81*	*1,069*	*11.2*	*1,124*	*12.6*	*1,172*	*14.1*	*1,100*	*14.1*
2,200	13,449	789	5.14	852	6.33	912	7.60	973	8.94	*1,029*	*10.3*	*1,080*	*11.7*	*1,134*	*13.2*	*1,182*	*14.7*	*1,130*	*15.3*
2,300	14,060	807	5.54	868	6.76	927	8.05	986	9.41	*1,040*	*10.8*	*1,091*	*12.3*	*1,144*	*13.8*	*1,192*	*15.3*	*1,160*	*16.5*
2,500	15,283	847	6.39	904	7.70	959	9.04	1,014	10.5	*1,064*	*12.0*	*1,115*	*13.5*	*1,166*	*15.1*	*1,212*	*16.7*	*1,190*	*17.8*
2,700	16,505	888	7.35	940	8.76	993	10.2	1,042	11.6	*1,092*	*13.2*	*1,141*	*14.8*	*1,189*	*16.5*	*1,235*	*18.2*	*1,220*	*19.2*
2,900	17,728	929	8.43	980	9.92	1,031	11.4	1,078	13.0	*1,125*	*14.6*	*1,170*	*16.2*	*1,217*	*17.9*	*1,261*	*19.7*	*1,250*	*20.7*
3,100	18,951	971	9.62	1,020	11.2	1,069	12.8	1,114	14.4	*1,159*	*16.1*	*1,200*	*17.8*	*1,245*	*19.6*	*1,287*	*21.4*	*1,280*	*22.2*
3,300	20,173	1,014	11.0	1,060	12.6	1,108	14.3	1,152	16.0	*1,195*	*17.8*	*1,235*	*19.6*	*1,277*	*21.4*	*1,318*	*23.2*	*1,310*	*23.8*
3,500	21,396	1,058	12.5	1,103	14.1	1,148	15.9	*1,190*	*17.8*	*1,233*	*19.6*	*1,274*	*21.4*	*1,311*	*23.3*	*1,352*	*25.2*	*1,340*	*25.4*

Note: Wheel diameter 33 in., wheel width 15¼ in. Approximate weight of single fan 810 lb. Actual fan horsepowers are shown in table, add for loss through drive when selecting motor.
Ratings in italics require Class II construction.

Standard Fan Arrangements and Designations. When we speak of "fan arrangement," we mean the manner in which the fan is arranged to drive. All members of the National Association of Fan Manufacturers

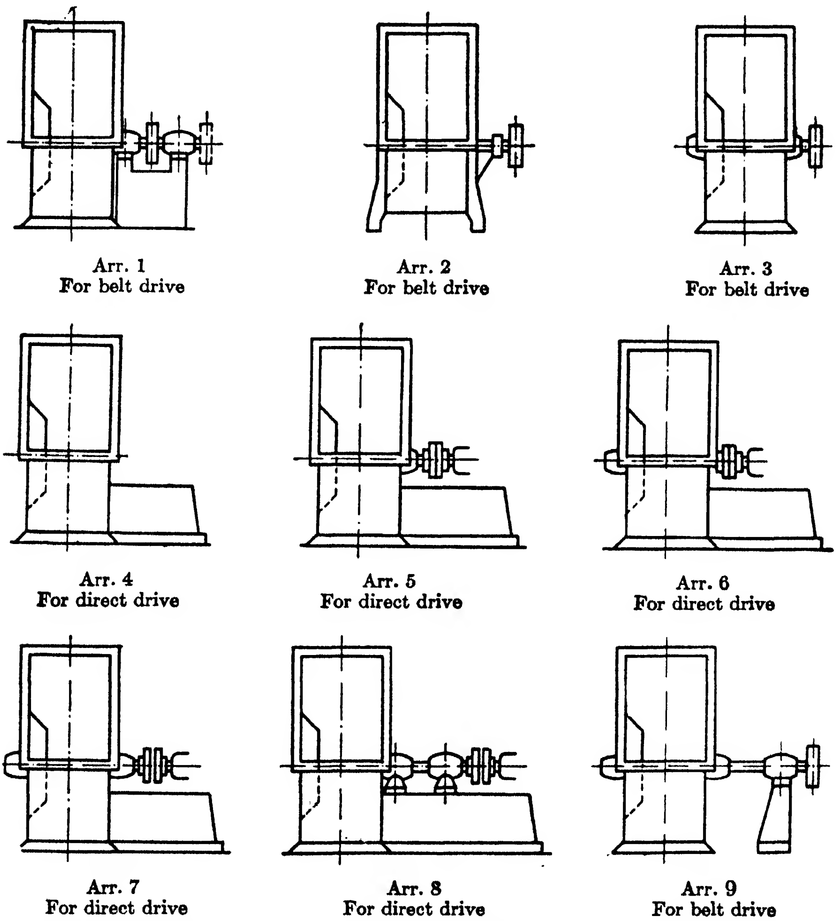


Fig. 6.—Standard fan arrangements. Arrangements are either for belt or for direct drive. Bearing locations can be easily noted.

build fans having different arrangements of bearings and different arrangements regarding the type of V-belt drive or direct drive. For convenience, these arrangements have been standardized so that "arrangement 3" would mean the same style of fan, regarding drive, for all manufacturers who are members of the National Association of Fan Manufacturers. Figure 6 illustrates the eight standard arrangements for centrifugal fans.

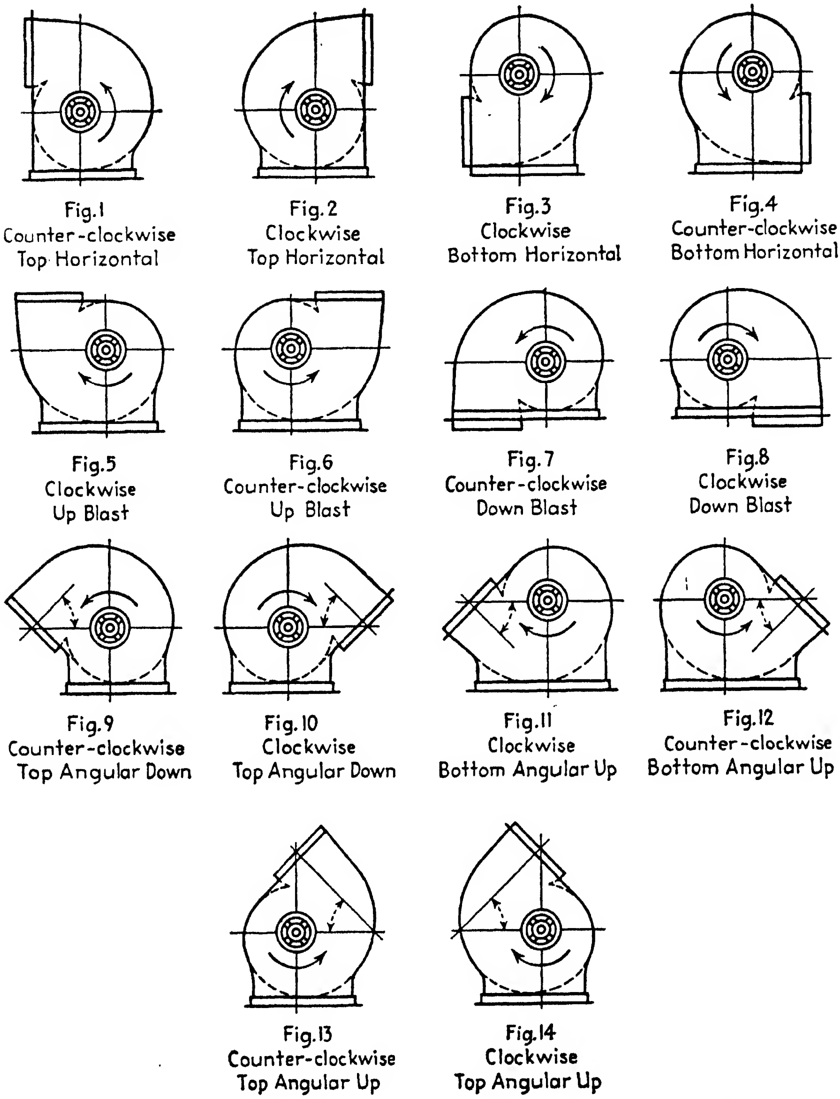


FIG. 7.—Standard designation of fans. Notations are those made when facing the driver side of the fan.

Note that arrangement 9 *is not a standard arrangement but is a special Sturtevant design.*

When we speak of the *designation* of a centrifugal fan we mean two things: its direction of rotation and the direction in which the fan outlet faces; the latter is also called "direction of discharge." Figure 7 illustrates the standard designation of fans which is used by all members of the National Association of Fan Manufacturers. Note that regarding direction of rotation we specify *clockwise* or *counterclockwise,* as we see the rotor if we stand *facing the drive side of the fan.*

Double-width and Single-width Fans. Single-width fans are fans of the ordinary type, having a single inlet on one side of the fan. The complete name of such fans is "single width, single inlet." This is abbreviated "S.W.-S.I." A double-width fan simply consists of a rotor twice the width of a single-width rotor, the rotor being installed in a wider casing having *air inlets on both sides.* The full name of this fan is "double width, double inlet," abbreviated "D.W.-D.I."

Sometimes fan manufacturers publish separate tables giving the ratings of their D.W.-D.I. fans. The capacities and performances of D.W.-D.I. fans may be obtained from rating tables on single-width single-inlet fans by using the following multiplier factors:

	Multiply by
Outlet velocity	1.11
Rpm	1.01
Horsepower	2.04
Outlet area	1.8

Manufacturers' Rating Tables of Fans. Table 1 gives manufacturers' rating tables for the size 5½ Buffalo limit-load Conoidal. Study this carefully so that you become familiar with the methods by which the manufacturers publish this information.

In using manufacturers' data to select a fan, determine the volume of air that you wish to handle, after which you must design a duct system and calculate its resistance according to the method presented in Chap. XXIV. Your information will then consist of a statement such as the following:

Required—A fan to handle 8,000 cfm against an over-all static resistance of 1.2 in. of water

From the manufacturers' data you will find that several fans can meet this condition. Select the most efficient fan possible which will fit into the space available and which can be purchased with the money available.

Water Pumps. The water pumps used in air-conditioning work are ordinarily of the centrifugal type which operate on the same basic principle as a centrifugal fan. Figure 8 shows a view looking at the inlet side of a centrifugal pump, the casing removed, and the impeller or rotor cut open to show the vanes or blades. The water enters the center or "eye" of the impeller through the housing, which is removed in Fig. 8. The water is then thrown outward between the fins or blades of the impeller and directed out of the discharge by the scroll casing or impeller housing. In Fig. 8, the discharge of the pump is the pipe pointing upward.

Some years ago almost all water pumps were of the so-called "cradle-mounted" type, as illustrated in Fig. 9. Such pumps are still manufactured and are widely used for many pumping purposes in factory and city buildings. The main advantage of such a pump is that the electric motor (or other driving power unit) can be removed without disturbing the pump. Also V-belt drive, flat-belt drive, gasoline engine, steam turbine, or other types of drive may be utilized to operate the cradle-mounted pump.

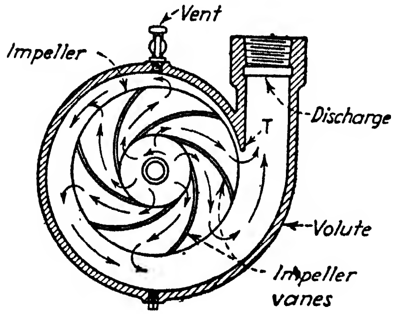


Fig. 8.—View looking at the inlet side of a centrifugal pump. (*Courtesy of Power.*)

For most air-conditioning applications, the water pumps required are small, as water pumps go. Except for very large cooling-tower installations, the pumps used in air-conditioning work rarely exceed 500 to 800 gpm capacity. (Large cooling-tower pumps may be 2,000 gpm and greater.) For these smaller pumping requirements, pumps of unit construction, *i.e.*,

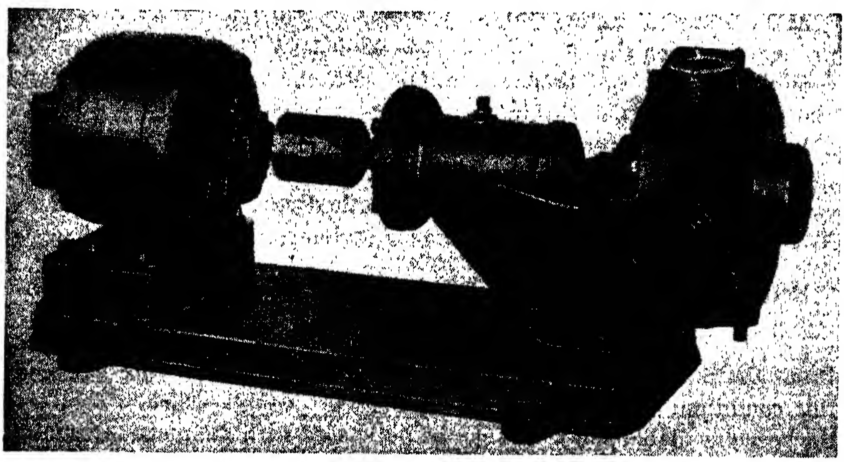


Fig. 9.—View of single-stage Ingersoll-Rand class CRVN pump with motor drive. (*Ingersoll-Rand Co.*)

having the pump and motor assembled into a single foundation, requiring no base plate or cradle, have become widely accepted in recent years. The only disadvantage of such pumps is that if the electric motor must be removed for repairs it *cannot* be removed and replaced by another motor *in a matter of minutes.* This is not a serious disadvantage because standard nationally known electric motors are used, which should not have to be removed for *emergency* repairs if properly cared for. Even if a new motor should be required, it could easily be obtained from the pump manufacturer or the electric-motor manufacturer.

The Ingersoll-Rand product known as the Cameron Motorpump is an outstanding example of this pump-and-motor *unit construction.*

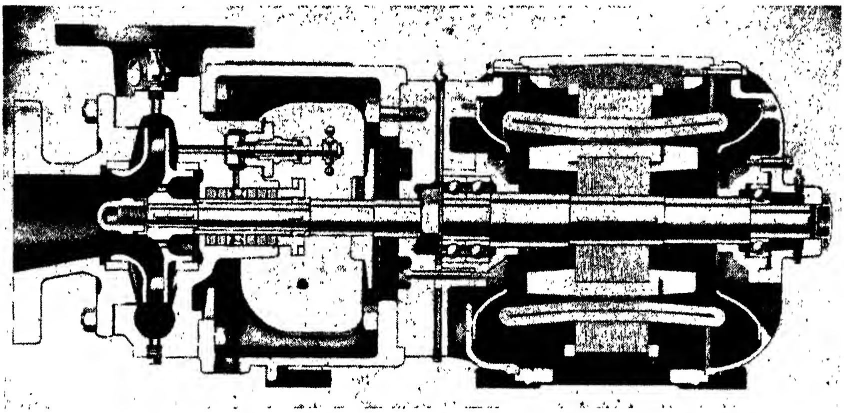


FIG. 10.—Cross section of Ingersoll-Rand single-stage Motorpump. (*Ingersoll-Rand Co.*)

Figure 10 illustrates the appearance and general construction of typical Motorpumps. One of the most useful advantages of this type of pump for air-conditioning work is that the pump may operate in any position. Pumps of this type may be bolted *to the floor, ceiling, or wall at any angle.* The only foundation required is a flat surface with four or six properly located anchor bolts. The adaptability and ease of installation is an important factor in air-conditioning applications. See Fig. 11.

Pump Selection. In order to select a suitable pump, the following information should be provided accurately to the pump manufacturer:

Gallons per minute	——	Liquid (water or brine)	
Total Head	——	temperature	——
Suction lift	——	*If brine, give also*	
Net positive suction head	——	Specific gravity	——
		Viscosity	——

Current conditions:

AC————Volts————Phase————Cycles

DC————Volts

The pump manufacturer will guarantee the pump to perform in accordance with this information, but of course is *not responsible* for an error that you *might make in furnishing the data.* You already know how to determine the electric-power characteristics, or "current conditions" as they are called in the above list. You have already made calculations regarding the gallons per minute and the chilled water required in air-conditioning installations. As you will not often encounter chilled-brine-pumping, you will rarely require specific gravity or viscosity.

In the use of fans we do not take account of the work required to lift air upward because of the lightness of the air. Furthermore, air handled by the fan and duct systems is ordinarily at substantially the same weight as atmospheric air. In the case of water, however, if a pump lifts water

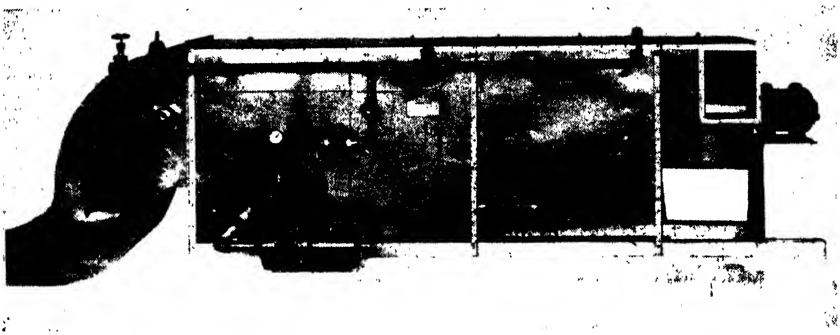


FIG. 11.—Air-cooling unit equipped with Ingersoll-Rand Motorpump. (*Ingersoll-Rand Co.*)

from one point and discharges it at a higher point, we must take account of the work required to lift this water. Fortunately, it is not necessary to use our formula $W = F(D)$ from Chap. V. Instead, we simply measure the difference in height between the water level in the tank to which the pump suction connects and the water level against which the pump discharge operates. This distance represents "unbalanced water leg." This distance is measured in feet; the frictional resistance of the piping is calculated in feet. We add these two and obtain *total head in feet.* In other words, frictional head plus unbalanced water leg equals total head.

The items "suction lift" and "net positive suction head" in the preceding list are for information of the pump manufacturer. It is not within the scope of this chapter to explain these factors or to explain the method of designing water-piping systems and calculating the friction head or friction resistance of such systems. This material will be covered in the latter part of Chap. XXIV.

CHAPTER XXIV

DUCTS AND PIPING. DESIGN AND CALCULATIONS OF RESISTANCE

General. This chapter is divided into three main parts: one regarding air ducts, the second concerning water piping, and the third giving brief pressure-drop information regarding Freon refrigerant lines. In each a recommended procedure is given, in definite steps, by means of which we can design a duct or piping system and calculate its resistance. Little of a theoretical nature will be mentioned in this chapter. It is necessary that you understand Chap. XXII thoroughly before proceeding with this chapter. After finishing this chapter, review Chap. XXIII.

PART 1

Ducts for Handling Air

Friction Losses. In Chap. XXII we saw that the head losses in fluid flow are divided into friction losses and impact losses. We shall first consider friction losses, in connection with air ducts. This means that our discussion under this heading will concern only *straight* ducts of *uniform* sizes. (As you know from Chap. XXII, changes in sizes and elbows or turns cause *impact* rather than *friction* losses.)

In designing a duct system, we have little control over the total length of the duct. Usually, a duct system must extend from a fresh-air intake through certain conditioning equipment, then through a fan, and finally through the necessary outlets in the conditioned space. A recirculation duct is also usually involved. Thus the place from which we take our air, the location of the fan and conditioner, and the size and shape of the conditioned space will determine the length of our duct. Naturally *the longer the duct, the greater the friction loss.*

We also have little control over the air quantity that must be handled. This quantity is determined by our heating-load or cooling-load calculations, *whichever requires the greater amount of air.* (The summer operating conditions ordinarily require a greater air volume than the winter conditions.) In Chap. XXII it was brought out that the higher the velocity of air in a certain duct, the greater the impact losses of a certain elbow. Similarly, the greater the velocity in any duct or pipe, the greater the friction losses. Since we have seen that we have little control over duct length and air capacity, what can we control and select in order that we may have some control over the friction (and impact) losses of a duct system?

If our duct length and air quantity are both fixed, we can vary the cross-sectional *area* of the duct. For a fixed air quantity, the greater the area, the lower the velocity. Obviously then the smaller the duct area, the higher the air speed, if a fixed quantity of air is to be carried. We have just pointed out that frictional resistance is greater at higher air velocities. For practical purpose, we may say that *frictional resistance varies directly as the square of the velocity.* The frictional resistance offered by a duct will always vary depending upon the smoothness of the inner surface of the duct. These two factors that affect frictional resistance may be expressed together in a single equation. Actually, we shall *not use the equations* in practical problems regarding duct work. We are giving you formulas (1) and (2) because the *charts that we are going to use shortly* have been based upon them. The proof of these formulas and the use of them is beyond the scope of this text.

For round ducts:

$$h_L = \frac{L}{CD}(h_v) = \frac{L}{CD}\left(\frac{V}{4{,}005}\right)^2 \tag{1}$$

For rectangular ducts:

$$h_L = \frac{L}{C}\left(\frac{a+b}{2ab}\right)(h_v) = \frac{L}{C}\left(\frac{a+b}{2ab}\right)\left(\frac{V}{4{,}005}\right)^2 \tag{2}$$

where h_L = head loss due to friction, *inches of water*

$h_v = \left(\frac{V}{4{,}005}\right)^2$ = velocity head, *inches of water*

V = velocity of air flow, *fpm.*

L = length of duct, *ft*

D = diameter of round duct, *ft*

a and b = the two sides of a rectangular duct, *ft*

C = a constant depending upon the nature of the inside duct surface [1] (see Table 1)

Note that the term C appears in the bottom of the fraction in both of the above equations. We have pointed out that C is a constant depending upon the nature of the inside surface of the duct. Values for C are given

[1] For students familiar with physics, C for round pipe may be accurately defined thus (this is not within the scope of this text):

$$C = \frac{1}{\text{coefficient of friction}} = \text{length of duct } \textit{in diameters} \text{ for one velocity head loss}$$

in Table 1. We give these values not because you will be required to substitute them in Eq. (1) or (2), but instead for reasons that are explained in the next paragraph.

For practical duct-design work, we shall use the chart shown in Fig. 1. This chart is based upon ducts having an inside surface such that the value of C is 50. This value applies for ordinary sheet-metal heating and ventilating ducts of good construction, as you can see from Table 1. In the event that you have ducts with rough inner surfaces, you will have to choose the value of C (*lower* than 50) from Table 1. (It is obvious from Table 1 that you will not encounter ducts for which C is greater than 50.)

TABLE 1.—VALUES OF C FOR EQ. (1) AND (2)

Nature of Inner Duct Surface	Value of C
Perfectly smooth surface (no seams)	60
Very smooth surface, as in air-conveying factory-dust exhaust systems	55
Ventilating and air-conditioning ducts, sheet metal, of good construction	50
Tile, brick, or concrete—smooth	45
Tile, brick, or concrete—rough	40
Ducts made of wire lath, plastered *outside*, *no inner finish*, *approximately*	35

Let us suppose that you have a large rough tile duct. From Table 1 we see that $C = 40$ for this duct. We then take our duct resistance as determined by the chart in Fig. 1 and multiply it by the fraction.

$$\frac{50}{40} = \frac{\text{standard duct } C}{\text{actual duct } C}$$

In other words, our rough tile duct would have a frictional resistance considerably greater than an ordinary sheet-metal duct.

Figure 1 is also based upon standard air at 70°F and a barometric pressure of 29.92 in. Hg. In air-conditioning work we may have occasion to handle air from 40 to 150°F, but the temperature correction is so slight that we may still use Fig. 1 for air within the temperature range given above.

If the barometric pressure is greater than standard, the air is somewhat compressed and hence higher in density. Theoretically, frictional resistance varies directly with air density, *i.e.*, it increases as the air density becomes greater. Even though the chart in Fig. 1 is exactly correct only for air at standard barometric pressure, we may use it without making any density correction provided the absolute pressure inside the duct is *not more than* 1 *in. Hg different* from atmospheric pressure. (If you wish to set up a density-correction proportion based upon air at an absolute pressure of 30.9 in. Hg you will see that the frictional resistance would be increased only about 3 per cent.) In other words, if the static pressure

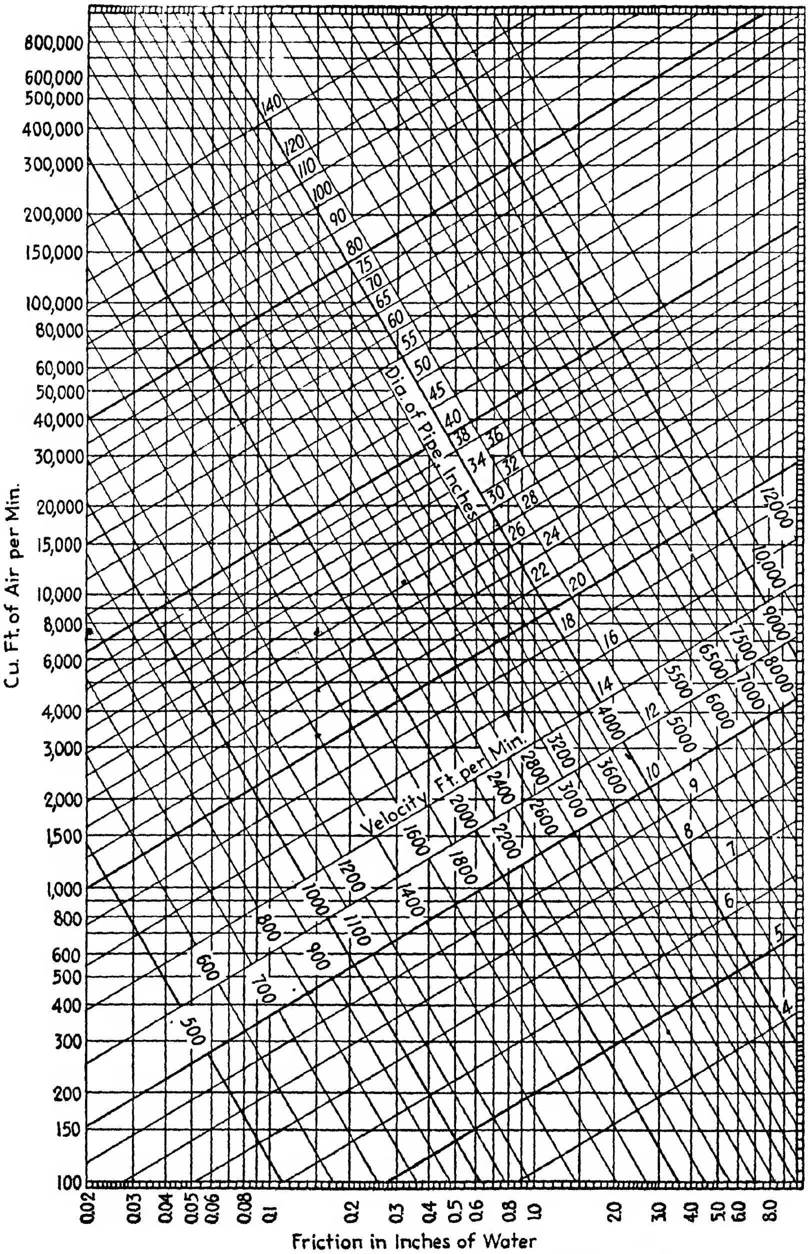


FIG. 1.—Friction of air in pipes in inches of water per 100 feet. (*Courtesy of A.S.H.V.E.* "Guide.")

within a duct is 1 in. Hg or less, we need make no correction regarding air density. Actually, in air-conditioning work, *positive* static pressures—pressures *greater than* atmospheric pressure outside the duct—almost never exist.

Use of the Friction Chart. We are now ready to explain the chart in Fig. 1. The bottom scale in Fig. 1 is labeled "friction in inches of water per 100 ft." This means the frictional resistance each 100 *ft of straight duct* would offer. Note the diagonal sloping lines in Fig. 1 which are marked *in inches.* These lines give the diameter of a *round* pipe or *round* duct. This means that using Fig. 1 we will base all our duct-design calculations on round ducts. We shall soon see an easy method of changing this information to fit ordinary practical rectangular ducts. Now notice the left-hand vertical scale of Fig. 1. The meaning of this scale is perfectly clear, as it is simply "cubic feet of air per minute," or "cfm."

For our duct-design work we shall standardize on a frictional resistance of 0.1 *to* 0.2 *in. of water per* 100 *ft of straight duct.* This means that when we are designing a new duct system we shall use only that portion of the chart lying between the vertical lines marked 0.1 and 0.2 in. friction.[1] Ducts designed on the basis of the 0.1 in. friction per 100 ft are in general larger and more costly to build, but because of the low resistance require less fan power. As we approach the higher limit of our design resistance standard, *viz.*, 0.2 in. per 100 ft, our duct work is smaller and cheaper to build, but requires greater fan power because of frictional resistance.

Let us take some examples involving the use of our friction chart. Suppose that we wished to handle 6,000 cfm through a duct having a frictional resistance of 0.2 in. per 100 ft. Follow the horizontal line marked "6,000 cfm" to the point where it crosses the vertical friction line marked "0.2." From the diagonal lines you see that a duct between 24 and 26 in. in diameter is required. Actually, the chart seems to give a duct just less than 25 in. in diameter, so that we would use a 25 in. diameter round duct to carry this 6,000 cfm.

Suppose that you have a 40 in. diameter round duct and wish to find how much air this duct can carry. If the frictional resistance is to be only 0.2 in. per 100 ft of length according to the standard that we have set, find on Fig. 1, the intersection of the "0.2" friction line and the sloping line marked 40 in. diameter pipe. Read horizontally to the scale at the right, and you will see that this duct can carry about 23,000 cfm. Suppose

[1] There is one important exception: Note in Fig. 1 that a 50 in. diameter pipe operating to have a frictional resistance of 1 in. per 100 ft is carrying about 27,000 cfm. Velocity of the air is 2,000 fpm. Never design duct work involving velocities higher than 2,000 fpm except on special advice from competent engineers, even though your frictional resistance per 100 ft of length must be made less than 0.1 in. of water. You will only run into this problem in the case of duct systems carrying more than 27,000 cfm.

that our duct, in the problem which we have just discussed, *must* carry 40,000 cfm of air. Start at the scale on the left, at the point marked 40,000; read horizontally to the right until you meet the line marked 40 in. diameter of pipe; then read vertically downward and you see that the duct would have a resistance of 0.6 in. of water per 100 ft of straight duct.

If you have followed closely the preceding two paragraphs and thoroughly understand them, you know all that you need to know regarding *determining friction* of duct work from Fig. 1. However, Fig. 1 represents for onvenience one more item of information. Suppose that you plot a point on Fig. 1. Turn Fig. 1 sideways so that you can read the figures labeling the diagonal lines marked "velocity in feet per minute." If we take our point representing a 40 in. diameter duct carrying 23,000 cfm and having a frictional resistance of 0.2 in. of water, we see that the velocity is almost 2,600 fpm. As closely as we can read this velocity from the chart, we might call it a velocity of 2,550 fpm. Take our other point, where our 40 in. diameter duct carries 40,000 cfm. You see that the velocity is just over 4,500 fpm. From the chart our best reading of this velocity would be 4,600 fpm.

Throughout the use of Fig. 1 described above, we are concerned with *round ducts* of various diameters. Do not worry about this as all duct design and friction calculations are based on round ducts. Later in this chapter you will learn how to translate information from terms of round ducts to terms of *equivalent rectangular ducts.*

Impact Losses. In Chap. XXII we learned that static pressure losses caused by impact result when air passes through an elbow or through any piece of apparatus such as a conditioning unit. We shall always obtain the static pressure through apparatus directly in terms of inches of water. This information depends upon the velocity at which we pass air through the apparatus and as obtained from manufacturer's data. The impact loss caused by elbows is obtained from charts.

Determining Pressure Losses in Round and Square Elbows. Figure 2 illustrates the chart showing pressure loss in round elbows. Figure 3 gives the chart showing pressure loss in square elbows. The information in Fig. 3 is sufficiently accurate for rectangular duct work, although the chart is absolutely exact only for ducts having sides of equal length. *Rectangular duct work should never be constructed with the long side more than five times the short side, otherwise the pressure-loss calculations from both Figs.* 1 *and* 3 *will not be correct.*

In Chap. XXII, we learned how to calculate the velocity head of any flowing fluid. The velocity head of air in inches of water for various air velocities is shown in Tables 3 and 4 of Chap. XXII. The elbow-pressure loss charts in Figs. 2 and 3 show the elbow-pressure loss in *per cent of velocity heat lost.* The impact loss through an elbow depends upon the radius of the elbow. The shorter the radius, the greater the pressure loss. Note that in Figs. 2 and 3 we speak not of the inside radius of the duct surface, but instead of the *radius of the center line of the duct.* This dimension

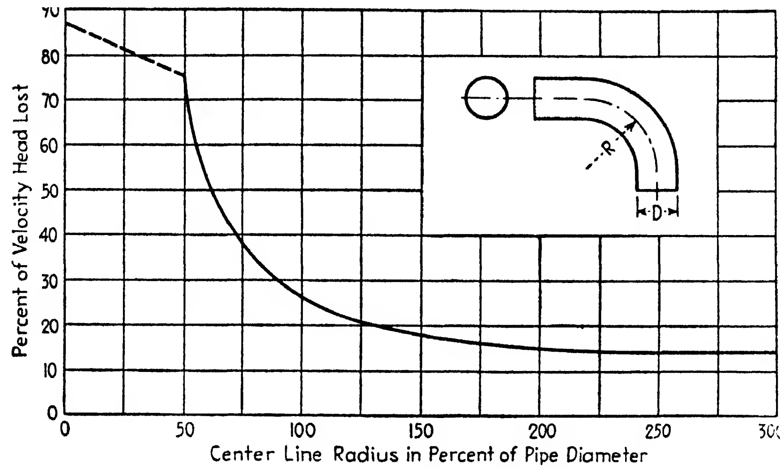


FIG. 2.—Curve showing loss of pressure in round elbows. (*Courtesy of A.S.H.V.E. "Guide."*)

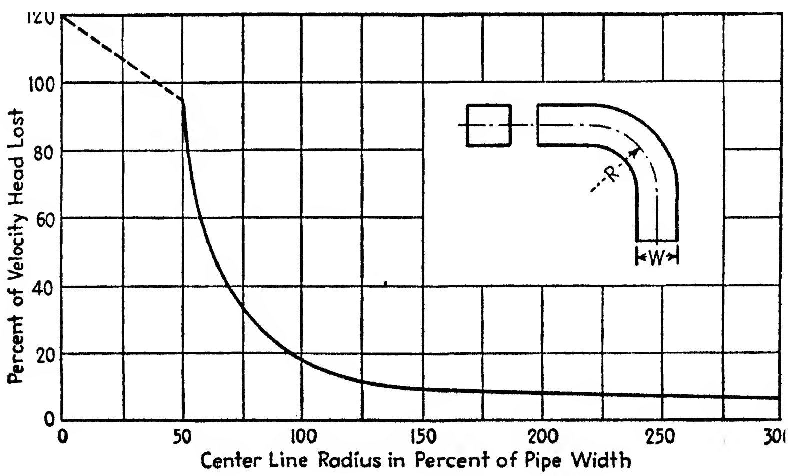


FIG. 3.—Curve showing loss of pressure in square elbows. (*Courtesy of A.S.H.V.E. "Guide."*)

is marked R in Figs. 2 and 3. If R is just equal to the diameter of a round duct or just equal to the width of a rectangular duct, we would say that the center-line radius is 100 per cent of pipe diameter or pipe width. Referring to Figs. 2 and 3, you will see that center-line radius is plotted, in per cent diameter or pipe width, along the bottom axis of the graph. As a standard of good duct design, the center-line radius should always be made 150 per cent of the pipe diameter or pipe width. From Figs. 2 and 3 you can see that this results in a pressure loss of about 17.5 per cent of a velocity head for a round duct and less than 10 per cent of a velocity head for rectangular ducts. *Note the standard just given, and use it in designing duct elbows.*

It is recognized that in some circumstances the available space will make necessary the use of elbows having a shorter radius than the design standard we have just given. In ordinary rectangular duct work, *do not use a simple elbow having a center-line radius less than the duct width.* In cases where it is necessary to use a shorter radius than this, use an absolutely *square elbow* in which are installed "duct turns" or vanes made up by a reliable manufacturer. In this case, the pressure loss caused by the elbow will be not more than 20 per cent of a velocity head. *Do not use homemade sheet-metal vanes in square elbows as they may actually increase the head loss.* Manufacturers of duct turns have available curves for square elbows equipped with their products, the curves being similar to Fig. 3.

The following data concern Ducturns manufactured by Tuttle and Bailey, Inc.

Ducturns are composed of scientifically designed turning blades for use on any air-conditioning or ventilating system to eliminate the necessity of long radius turns and allow the use of right-angle elbows throughout. Ducturns greatly simplify the layout of duct work and permit the use of easily fabricated right-angle turns that not only require less room, but furnish a more attractive and finished installation; straight runs of duct work and right-angle elbows can be painted to match surrounding trim and made to resemble a part of the building structure itself. Ducturns represent a remarkable achievement in duct design.

There are two standard Ducturn blades: type B for smaller elbows and type D for larger turns. The blade is made of steel heavily coated both inside and out with a special red lead paint. Special attention is called to the fact that Ducturn blades are tubular in construction, tapering down gradually to a very fine edge; the blades are so shaped that when they are placed in a right-angle turn any point on one blade will be equally distant from an opposite point on an adjacent blade, thus eliminating the changing in pressures as the air passes through the blades and minimizing the friction loss (see Fig. 4).

Resistance to Air Flow. The chart in Fig. 5 indicates the resistance loss for various air velocities in a right-angle elbow equipped with Ducturns compared with the loss through a long-radius elbow constructed to common trade practice; it will be noted that the friction loss in the right-angle turn is slightly less although practically the same as that encountered in a long-radius turn.

Ducturns are primarily designed for use in right-angle turns where the duct width and height is the same beyond the elbow turn as it is directly before it. In order that Ducturns shall function properly, the unit must

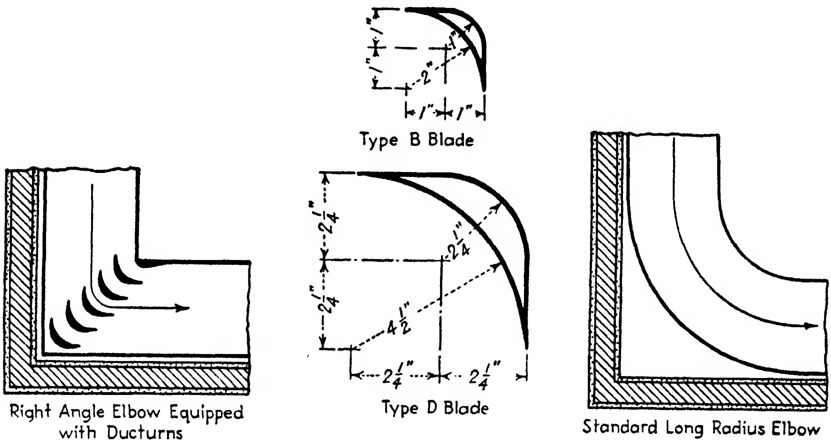


Fig. 4.—Application details for Ducturns. (*Tuttle & Bailey, Inc.*)

be placed at an angle of 45 deg. Each edge of a blade must be parallel to the direction of air flow to have a minimum of resistance and to avoid air eddies at the turn.

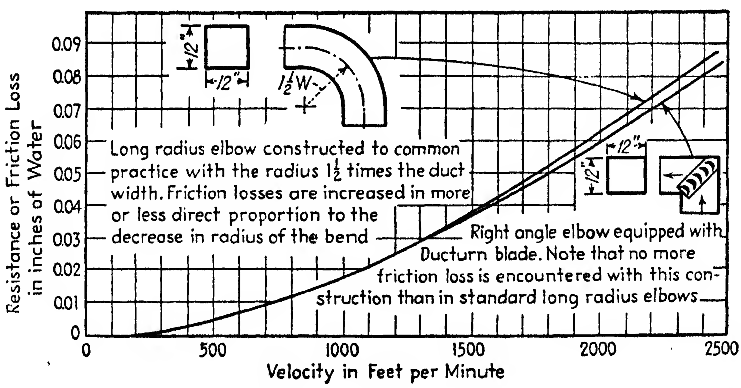


Fig. 5.—Resistance to air flow for square elbow equipped with Ducturns as compared with long radius elbow under same conditions. (*Tuttle & Bailey, Inc.*)

In the construction of duct work it is often necessary to go from a larger to a smaller size duct on a particular run, or vice versa, in which case the fixed Ducturn could not effectively be placed from corner to corner in the elbow.

An adjustable Ducturn is available which can be moved and set in an elbow at a position so that the unit will be at an angle of 45 deg. This construction can also be used at the junction of a right-angle take-off from a horizontal trunk line where the main trunk is sized down after each branch run (see Fig. 6).

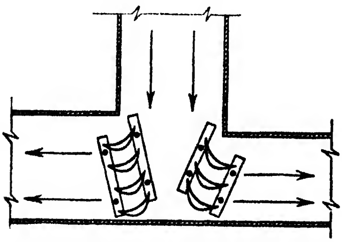


FIG. 6.—Main-trunk-line duct divided into several size branch ducts, showing application of Ducturns. (*Tuttle & Bailey, Inc.*)

Having determined the pressure loss through an elbow from Fig. 2 or 3, as a per cent of velocity head lost, determine the velocity head of the air flowing and express the elbow-pressure loss as *static pressure*. This involves multiplying the velocity head, as obtained from Table 3 or 4 in Chap. XXII, by the per cent of this velocity head which is lost in the elbow.

Branch Ducts. In many cases it is desirable to distribute air by means of a duct system composed of a trunk duct from which various branches of different lengths are taken. In one method of duct design which we are soon to consider, branch ducts must be chosen *to have the same friction resistance per* 100 *ft of length as the trunk duct out of which the branches run.* It would be lengthy and difficult to figure the size of these branch ducts mathematically, so charts have been prepared from which the sizes of branch ducts, based upon equal friction per feet of length, are quickly and easily obtained.

Suppose that you have a trunk duct leaving a fan carrying 10,000 cfm of air. Suppose that you wish to break up this trunk duct at one place into four branches carrying the air quantities given below:

Branch 1	4,000 cfm
Branch 2	3,000 cfm
Branch 3	1,800 cfm
Branch 4	1,200 cfm

Our first step is to express the air quantities carried by these branches in per cent of the air quantity carried by the main duct or trunk duct.

$$\text{Branch 1}\ \frac{4{,}000}{10{,}000}\ (100) = 40\%$$

$$\text{Branch 2}\ \frac{3{,}000}{10{,}000}\ (100) = 30\%$$

$$\text{Branch 3}\ \frac{1{,}800}{10{,}000}\ (100) = 18\%$$

$$\text{Branch 4}\ \frac{1{,}200}{10{,}000}\ (100) = 12\%$$

First we must determine the size of the round duct required as a *main duct* (or *trunk duct*) to carry our 10,000 cfm. To do this we use Fig. 1 and decide upon a frictional resistance per 100 ft of length (see pages 447 to 452). Let us take a frictional resistance of 0.1 in. of water per 100 ft of length. Turn to Fig. 1; starting from the bottom scale at the point marked 0.1, go vertically upward until you cross the horizontal line marked 10,000 cfm. On the diagonal pipe size lines, you see that a round pipe 34 in. in diameter is required. For further reference, note that the velocity is about 1,575 fpm.

Now turn to Figs. 7 and 8. Branch 1 carries 40 per cent of the capacity of the main duct. Branch 4 cannot be located in Fig. 7 because this includes only branches carrying 1 to 20 per cent of the main duct capacity. Hence look in Fig. 8. The sloping or diagonal lines represent the diameter of the *main duct* or trunk duct; the farthest of these lines to the right is labeled 100 (all dimensions are in inches). Follow to the left until you find the diagonal line representing the 34 *in. diameter main pipe.* Follow along this diagonal line to the point where it crosses the *horizontal line marked* 40 on the "per cent capacity" scale. From this point, read *straight up* to the top of the chart and you see that *a branch pipe* 24.5 *in. in diameter* is required for branch 1. Similarly from Fig. 5, branch 2 must be 22 in. in diameter. Turning now to Fig. 7, which is used in the same way, branch 3 requires an 18 in. diameter duct and branch 4 needs a round duct 15.5 in. in diameter.

We now know the size of each branch and the air volume flowing in each. Use the method discussed earlier, and determine the *velocity* in each branch. *Note that the velocity is lower in the smaller branch ducts and that the highest branch-duct velocity is lower than the main-duct velocity.* This will *always* be the case when you proportion branches, using Figs. 7 and 8, such that each branch has the same friction loss per 100 *ft of length* as the main duct. These branch-duct velocities are tabulated in Fig. 9.

We would not round off figures obtained in the preceding paragraph to the nearest inch. The work that we have done in our branch and trunk-duct problem would be the complete solution if the duct work were as shown in Fig. 9. However, an arrangement like this is extremely unlikely —ordinarily the branches would leave the trunk duct at various points, as illustrated in Fig. 10. Let us now proceed with the design of the practical trunk-duct system illustrated in Fig. 10. Starting in the main duct at point A, we have 10,000 cfm. We shall make this duct the same size as the main duct in Fig. 9, which has already been correctly designed, and start at the point where branch 1 leaves the main duct. We must really consider the main duct as *dividing into two branches—branch* 1 *and branch B.* Branch 1 is to carry 4,000 cfm so that the remaining 6,000 cfm must be carried by branch B. We realize that branch B is really a continu-

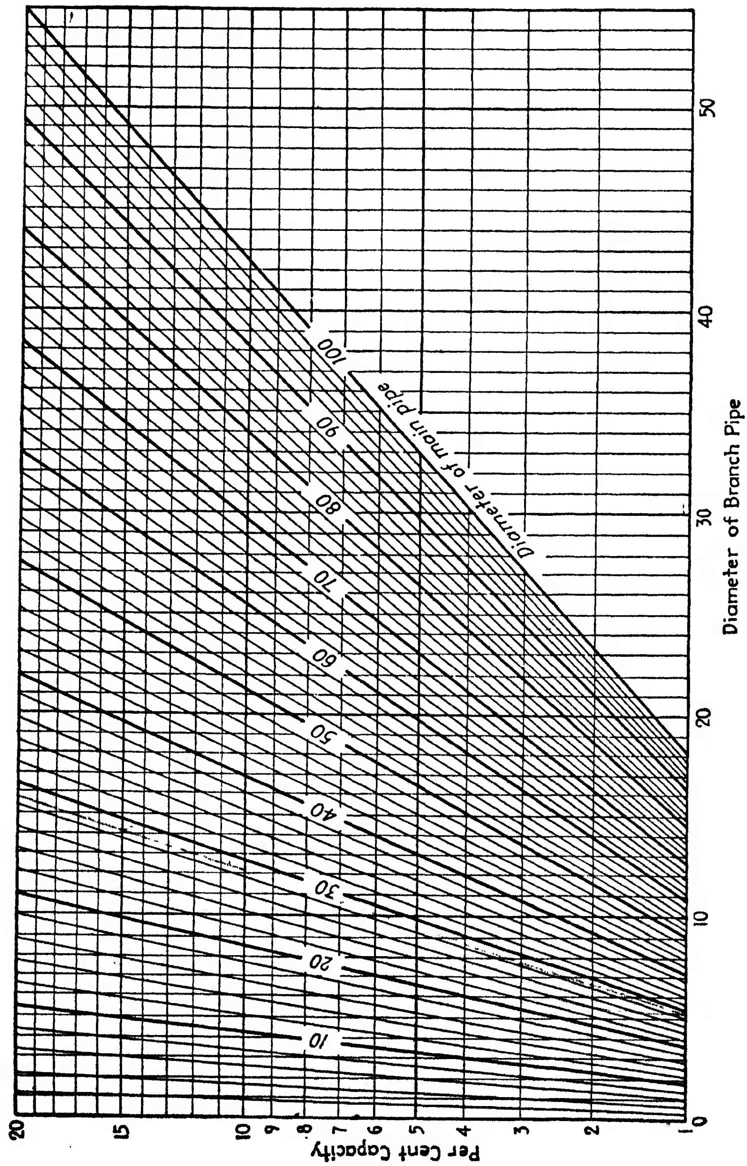


FIG. 7.—Main and branch pipes for equal friction per foot of length (1 to 20 per cent capacity). (*Courtesy of A.S.H.V.E. "Guide."*)

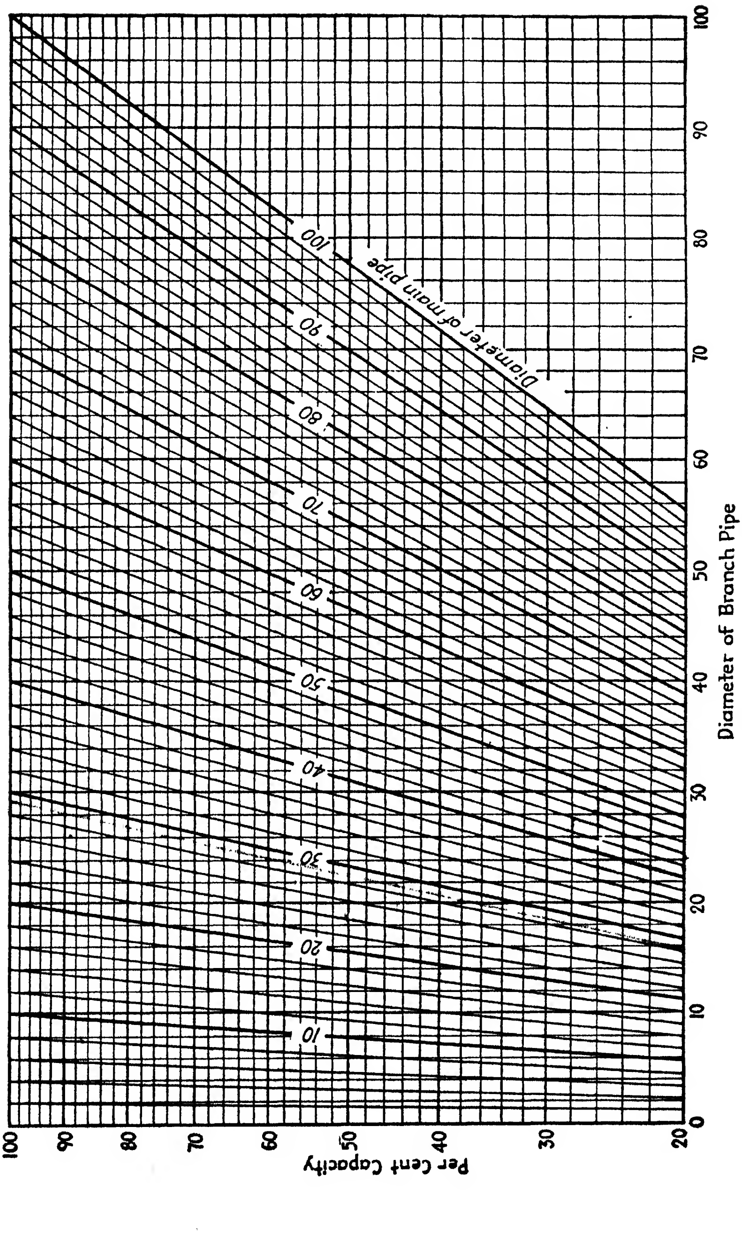


FIG. 8.—Main and branch pipes for equal friction per foot of length (20 to 100 per cent capacity). (*Courtesy of A.S.H.V.E. "Guide."*)

ation of the main duct; nevertheless since it is carrying only a portion of the air originally carried by the full-size main duct, we consider the duct run marked B as a branch. The 6,000 cfm carried by branch B now di-

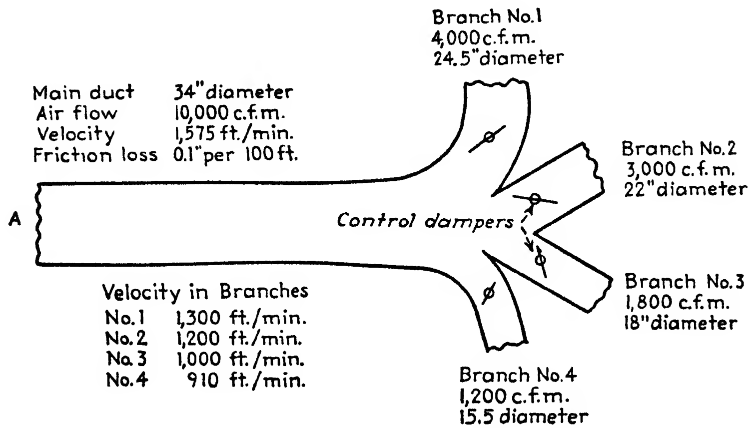


FIG. 9.—Several branch ducts leaving a main duct at one junction.

vides between branch 2 and branch C. Because branch 2 is known to carry 3,000 cfm, branch C carries the remaining 3,000 cfm. Finally, branch C divides into branch 3 carrying 1,800 and branch 4 carrying 1,200 cfm.

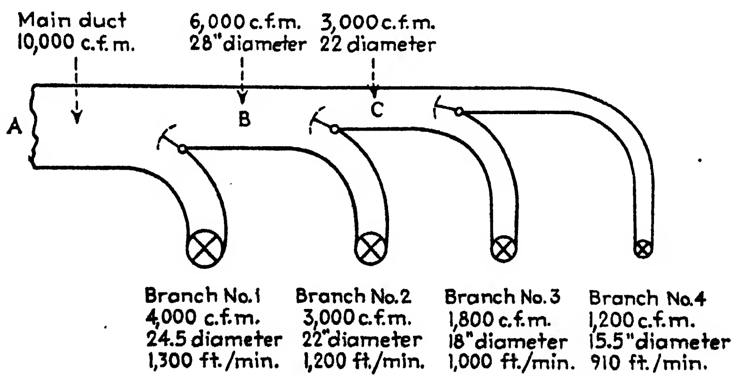


FIG. 10.—Trunk duct system, branches leaving at different points. Flow is from left to right.

As we analyze the practical duct system of Fig. 10, there are not just four branches, but really six branches. In other words, we have branches 1, 2, 3, and 4, and also branches B and C. To design this trunk-duct system, proportioning all branches so that they have the same friction *per foot of length*[1] we must repeat for all six branches the procedure that we followed on page 456 regarding Fig. 9. We shall not repeat the arithmetic of figuring the percentages; this information is given in Table 2. We now select the required size of round duct from the information in Table 2. Use Figs. 7 and 8, and follow the method described on page 457. The correctly chosen round duct sizes are all marked plainly on Fig. 7.

TABLE 2.—BRANCH-DUCT ANALYSIS, FOR USE WITH FIG. 7

	Air quantity, cfm	Air flow, % of that in main duct
Main duct *A*	10,000	100
Branch *B*	6,000	60
Branch 1	4,000	40
Branch *C*	3,000	30
Branch 2	3,000	30
Branch 3	1,800	18
Branch 4	1,200	12

Converting Round Ducts to Equivalent Rectangular Ducts. We have tables by means of which we can determine the several differently shaped rectangular ducts that are equivalent to a round duct of any diameter. By means of the same tables we can take any rectangular duct that we have and determine what diameter round duct is equivalent to it. If we are analyzing a duct system of rectangular ducts that has already been constructed, remember that we must convert such duct sizes to *equivalent round ducts.* This is necessary because our friction chart in Fig. 1 and our branch-duct charts in Figs. 7 and 8 are all based upon round ducts. What do we mean by the word "equivalent" as we have used it here? We do *not* mean equivalent in cross-sectional area; instead we say that a round duct and a certain rectangular duct are equivalent, *if when carrying the same quantity of air their friction loss per foot of length is the same.*

Perhaps charts can be drawn for use in converting from rectangular ducts to equivalent round ducts. The standard form of presenting this conversion information is, however, in the form of tables. These tables are based upon a mathematical formula that is too difficult to use every time we wish to convert from a rectangular duct to the proper diameter equivalent round duct. In case you are interested in this formula, and

[1] The chart in Fig. 1 is based upon a certain pressure loss per 100 ft of length. If two ducts have the same pressure loss per 100 ft of length, obviously each 50 ft, 10 ft, or each one foot of length would have the same friction loss in the case of both ducts.

to convince you that it really is too difficult to use, we present it below:

$$d = 1.265 \sqrt[5]{\frac{(ab)^3}{a + b}} \tag{3}$$

where a = one side of rectangular duct,
b = other side of rectangular duct,
d = diameter *in inches* of round duct for equal friction per 100 ft length when carrying the same number of cubic feet per minute

Those wishing to use formula (3) may do so. We feel sure that one trial will convince you that it is a waste of time. We shall proceed now to learn the use of the tables based upon formula (3).

How to Use Table 3. Note that Table 3 is in four parts. Each part is used in the same manner. Part 1 is for the smallest ducts, part 2 for the next larger, and so on, until part 4 covers the largest ducts. If we learn how to use any section of Table 3, we shall know how to use the entire table. Let us learn the use of this table by obtaining the various rectangular ducts that are equivalent (as to friction losses) to the 34-in. round duct that we have been using as a main duct in Figs. 9 and 10. Look at part 2 of Table 3. The numbers in the small squares across the top of the table represent the dimensions (in *inches*) of *one side* of the rectangular duct. The column of figures at the extreme left, in even inches, represents the dimension of the *other side* of the rectangular duct. The figures in the main body of the table are the diameters of various round ducts. These are arranged in columns underneath each of the numbers along the top of the table and opposite each of the numbers in the extreme left-hand columns. Note that the diameters of the round ducts are given to the *nearest tenth of an inch.*

Before we determine the various rectangular equivalents to a round duct 34 in. in diameter, let us find round ducts equivalent to the following three rectangular ducts:

$$\left.\begin{array}{l} 16 \times 66 \text{ in.} \\ 17 \times 60 \text{ in.} \\ 18 \times 56 \text{ in.} \end{array}\right\} \text{dimensions in inches}$$

To find the equivalent round duct for the first rectangular duct listed, start at the rectangular duct side "16 in." at the top of part 2 of Table 3. Run down the column under this number 16 until you are opposite the dimension "66 in." for the other side of a rectangular duct. This happens to be the bottom of the column under 16. Note that the diameter of the equivalent duct is 34.2 in. Similarly, solving the other two rectangular ducts listed above, we get the following result:

16 × 66 in. rectangular duct – equivalent to 34.2 in. diameter
17 × 60 in. rectangular duct – equivalent to 33.8 in. diameter
18 × 56 in. rectangular duct – equivalent to 33.9 in. diameter

Draw circles in pencil around the three round ducts you have just selected. Note that the numbers circled are *the nearest* in their respective columns *to being exactly* 34.0 *in. in diameter.* In other words, we have found *three ducts* that are the nearest to being exactly equivalent to a round duct 34 in. in diameter. Now let us find the others. Continue in Table 3, looking in the columns to the right of the columns that you have already used. Find the round duct diameter nearest to 34.0 in. After finding all these in part 2 of Table 3, proceed to part 3 of the table. In the first three columns here you will also find round ducts almost exactly 34 in. in diameter. If we list all these, we see that there are a total of 11 *different rectangular ducts given, all of which are substantially equivalent to a round duct* 34 *in. in diameter.* They are as follows:

16 × 66	19 × 52	22 × 44	28 × 34
17 × 60	20 × 50	24 × 40	30 × 32
18 × 56	21 × 46	26 × 36	

TABLE 3.—CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION

Part 1

Side Rectangular Duct	8	8.5	9	9.5	10	10.5	11	11.5	12	12.5	13	13.5	14	14.5	15	15.5	16
3	5.2	5.4	5.5	5.7	5.8	5.9	6.0	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0	7.1
3.5	5.7	5.9	6.0	6.2	6.3	6.5	6.6	6.7	6.9	7.0	7.1	7.3	7.4	7.5	7.6	7.7	7.8
4	6.1	6.3	6.5	6.7	6.8	7.0	7.1	7.2	7.4	7.5	7.7	7.8	7.9	8.1	8.2	8.3	8.4
4.5	6.5	6.7	6.9	7.1	7.2	7.4	7.6	7.7	7.9	8.0	8.2	8.4	8.5	8.6	8.7	8.9	9.0
5	6.9	7.1	7.3	7.5	7.7	7.8	8.0	8.2	8.3	8.5	8.7	8.8	8.9	9.1	9.2	9.4	9.5
5.5	7.3	7.5	7.7	7.8	8.1	8.3	8.5	8.6	8.8	9.0	9.2	9.4	9.5	9.6	9.8	9.9	10.1

Study the listing of rectangular ducts at the end of the preceding paragraph. Note that it starts with a very flat duct, *viz.*, 16 × 66. The last duct listed is almost exactly square, being 30 × 32. In other words, for the diameter of a round duct in some column of Table 3, the following holds true:

The farther to the *right* the column we use, the *nearer exactly square* is the rectangular duct selected.

TABLE 3.—CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION

Part 2*

Side rectangular duct	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	8.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.8	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.5	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	2[illegible].3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	24.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	15.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	27.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

* Additional sizes: 4 × 5 = 4.9; 4 × 6 = 5.4; 4 × 7 = 5.8; 5 × 5 = 5.5; 5 × 6 = 6.3; 5 × 7 = 6.5.

TABLE 3.—CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION

Part 3

Side rectangular duct	26	28	30	32	34	36	38	40	42	44	46	48
26	28.6											
28	29.7	30.8										
30	30.7	31.9	33.0									
32	31.7	32.9	34.1	35.2								
34	32.7	33.9	35.1	36.3	37.4							
36	33.7	34.9	36.1	37.3	38.5	39.6						
38	34.6	35.9	37.1	38.4	39.5	40.7	41.8					
40	35.3	36.7	38.0	39.3	40.5	41.7	42.9	44.0				
42	36.0	37.6	39.0	40.3	41.5	42.7	44.0	45.1	46.2			
44	36.9	38.5	39.9	41.2	42.5	43.7	44.9	46.1	47.2	48.4		
46	37.8	39.3	40.8	42.2	43.5	44.8	46.0	47.2	48.4	49.5	50.6	
48	38.5	40.0	41.5	43.0	44.4	45.6	46.9	48.1	49.3	50.5	51.6	52.8
50	39.2	40.8	42.3	43.8	45.2	46.5	47.9	49.1	50.4	51.6	52.9	54.0
52	40.0	41.6	43.1	44.7	46.1	47.5	48.9	50.1	51.3	52.5	53.8	55.0
54	40.7	42.4	44.0	45.5	47.0	48.4	49.9	51.1	52.3	53.5	54.8	56.0
56	41.3	43.0	44.6	46.2	47.7	49.1	50.6	52.0	53.3	54.6	55.9	57.0
58	42.1	43.8	45.4	47.0	48.5	50.0	51.5	52.9	54.2	55.5	56.8	58.0
60	42.7	44.5	46.1	47.8	49.3	50.9	52.3	53.8	55.0	56.4	57.7	58.9
62	43.4	45.1	46.8	48.4	50.0	51.7	53.0	54.5	55.9	57.2	58.5	59.7
64	44.0	45.8	47.5	49.2	50.9	52.4	53.9	55.4	56.8	58.1	59.4	60.6
66	44.7	46.5	48.2	50.0	51.6	53.1	54.7	56.2	57.6	59.1	60.4	61.6
68	45.3	47.2	48.9	50.7	52.2	53.8	55.5	56.9	58.4	59.9	61.3	62.6
70	46.0	47.8	49.5	51.3	52.9	54.5	56.2	57.7	59.1	60.6	62.1	63.5
72	46.5	48.4	50.1	51.9	53.7	55.4	57.0	58.7	60.0	61.3	63.0	64.5

Part 4

Side rectangular duct	50	54	60	66	72	78	84	88
50	55.0							
52	56.1							
54	57.2	59.4						
56	58.3	60.5						
58	59.3	61.6						
60	60.3	62.7	66.0					
62	61.3	63.7	67.1					
64	62.2	64.7	68.2					
66	63.2	65.7	69.3	72.6				
68	64.1	66.6	70.3	73.7				
70	65.0	67.6	71.3	74.8				
72	65.9	68.5	72.3	75.9	79.2			
74	66.8	69.4	73.3	76.9	80.3			
76	67.6	70.3	74.2	77.9	81.4			
78	68.4	71.2	75.2	78.9	82.5	85.8		
80	69.2	72.1	76.1	79.9	83.6	86.9		
82	70.1	73.0	77.1	80.9	84.6	88.0		
84	70.9	73.8	78.0	81.9	85.6	89.1	92.4	
86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
92	74.1	77.1	81.4	85.6	89.5	93.2	96.7	99.0
94	74.8	77.8	82.2	86.5	90.4	94.2	97.8	100.1
96	75.5	78.7	83.0	87.4	91.3	95.2	98.8	101.2

SOURCE: A.S.H.V.E., "Heating, Ventilating, and Air Conditioning Guide," Chap. 29, 1939

The farther to the *left* the column we use, the *more flat* is the rectangular duct selected.

As a matter of fact in part 2 of Table 3 some equivalent rectangular ducts are given which are considerably more flat in shape than the design standard which we have decided upon. On page 452 we stated that a rectangular duct should never be constructed with the long dimension more than five times the short dimension. The heavy stepped line across the lower left portion of part 2 of Table 3 marks ducts too flat to come within our design standard. Do not select rectangular ducts below and to the left of this line unless absolutely necessary.

TABLE 4.— MOST SQUARE AND MOST FLAT EQUIVALENT RECTANGULAR DUCTS FOR THE ROUND DUCTS IN FIG. 7

	Diameter of round duct, in.	Equivalent rectangular duct *nearest to being square.*	Equivalent rectangular duct, *most flat conforming to good design**
Main duct *A*	34	30 × 32	16 × 66
Branch *B*	28	26 × 26	12 × 60
Branch 1	24.5	21 × 24	11 × 50
Branch *C*	22	20 × 20	10 × 44
Branch 2	22	20 × 20	10 × 44
Branch 3	18	16 × 17	8 × 38
Branch 4	15.5	14 × 14	7 × 32

* See p. 452 for the design standard of flattest ducts permitted. See also p. 462 for the easy method of picking the flattest duct permissible.

Let us now consider a practical example on obtaining the equivalent rectangular ducts for a duct system that has been designed on our usual basis of round ducts of the correct diameter. In Fig. 10 we have such a duct system correctly designed and marked with the diameter of the round duct required for each branch. On page 463 we listed not one but 11 different rectangular ducts, any one of which might be used for the main duct in Fig. 10. Remember that the main duct in Fig. 10 was 34 in. diameter. The best rectangular duct to select is of course the duct nearest to being perfectly square. In many cases, however, we have space available which requires the use of a rather flat duct. For this practical reason and for practice in using Table 3, let us select two rectangular ducts for each branch in the duct system of Fig. 10. Let us select the rectangular duct *nearest square,* and the *most flat* rectangular duct that conforms to our design standard regarding flat ducts. Make this selection, and check your answers with the complete selection presented in Table 4.

Two Methods of Designing Duct Systems. The first method of designing duct systems is known as "the equivalent-friction method." This is the exact method of designing duct work. When duct work is designed on this basis, no matter how complicated the duct system—no matter how many and how long the branches—it is easy to calculate the over-all static resistance of the entire system. Remember that this must be calculated accurately because the fan must produce at its discharge this static pressure if the desired amount of air is to flow through the duct system. For short and simple duct systems, this accurate method is very little easier than for a very large duct system. However, this accurate method should be used for any trunk-duct system having five or more branches, especially if the branches are fairly long, *i.e.*, over 25 ft from the point where they leave the trunk duct.

The second method of duct design is called the "velocity method." It is less accurate than the equal-friction method, but for small duct systems it is quicker and easier. The velocity method should not be used for large duct systems for two reasons. (1) It does not result in a well-designed duct system, and the static resistance calculations may be slightly inaccurate. (2) While this method is the shorter and easier for small duct systems, it actually becomes more lengthy and difficult if you try to apply it to a large extensive duct system with many branches.

Regardless of which method we use, we want to know if a duct system is of good design, and we want to be able to calculate the over-all static resistance of this duct system with accuracy. As its name might imply, the equal-friction method of design is entirely based upon correct proportioning of the duct sizes to assure equal friction loss per foot of length. When using this method, we do not discover the size of rectangular ducts to be used until near the end. In the velocity method, we do not start by careful determinations regarding friction. Instead we take a guess and pick some rectangular duct size for the various branches by a method soon to be explained. Then at the end of our work we calculate the resistance and hope that we find our resulting resistance satisfactory.

The Equal-friction Method of Duct Design. This method of duct design is really already familiar to you. It consists of doing for an entire duct system exactly what you have done in the duct system in Fig. 10. In other words, its consists of proportioning the various branch sizes, using Figs. 7 and 8 and the method described on page 457. For your convenience, we present the entire method in the series of steps listed below. The long list of steps looks as though this might be a lot of work, but as you will see when you study the steps, you are already familiar with most of them. Let us say that the steps listed below apply to a supply-air duct system, *i.e.*, from *the fan outlet to the supply grilles.*

1. In the course of making your air-supply calculations,[1] you have determined the number of cubic feet per minute of air required at various points throughout the conditioned space.

For instance, where the conditioned space is divided into more than one zone, you know the air quantity that must be supplied to each zone. Within each zone, now locate your outlets or grilles and decide how many cubic feet per minute you desire at each grille, in order to ensure proper distribution of conditioned air throughout the space. It must be assumed that you have already found a satisfactory location for your conditioning unit and fan. Now make a sketch, drawing a single light line to represent the center line of your duct system. Start with your main duct at the fan outlet, and indicate the various branches leading to each grille. Give due consideration to the plan of the building, and make sure that you are not planning to run a duct through a steel column or some other obstacle. At the same time, maintain as simple a layout as possible, avoiding too many elbows and elbows that are too sharp.

2. Staying inside the standard we have set up for friction loss (0.1 to 0.2 in. per 100 ft of duct length), choose a value on which to base your design.

3. Check the air velocity you are using at the start of the run (just beyond the fan discharge, like point *A* in Fig. 10). Observe the following general standards:

Nearest grille over 100 ft from fan outlet—maximum velocity 2,000 fpm

Nearest grille 50 ft from fan outlet—approximate maximum velocity 1,500 fpm.

Nearest grille less than 25 ft from fan outlet—maximum velocity 1,000 fpm.

4. In accordance with 2 and 3, select from Fig. 1 the proper size of round duct for your main duct, as determined by the air quantity it must carry. Note the velocity while working on Fig. 1.

5. Get the proper size of round duct for all branch ducts, using the method explained on page 460 regarding Fig. 10. (Use the charts in Figs. 7 and 8,)

6. Convert all round ducts selected to the best rectangular ducts that will fit the space available (see page 461 for method, and use Table 3).

7. Select the radius you are to use for each elbow. Try to make them

[1] If the installation is for an all-year-round air conditioning you must calculate the air volume required for the summer according to Chaps. XIX and XXI. You must then also calculate the winter air volume required, based upon the design information in Chap. XVII. Whether or not radiators are used in the wintertime, the winter air volume will usually be found lower than the summer air volume. Of course, if the system is for summer only, you simply calculate the air volume according to Chaps. XIX and XXI.

all the same for convenience in figuring. Note the "per cent of velocity head lost" for the elbow design you choose (see pages 452 to 454 for method; see Fig. 3).

Except for selecting grilles, this duct system is now designed, and it will be a good one if you have followed the above procedure. We are now ready to calculate its resistance, which will be surprisingly easy. Our fan must create at its discharge a static pressure great enough to drive the desired amount of air to the grille *farthest from the fan.* The static-pressure loss *from fan outlet to the farthest grille* will be greater than the static loss to any grilles less far from the fan, because the ducts are designed to have *the same static pressure loss per* 100 *ft of length, which means also the same loss per foot of length.* This will be true unless a branch to some grille nearer the fan has many bad elbows, or some special apparatus, resulting in a greater *impact loss* than occurs in the longest duct run.

Let us now proceed to this step-by-step procedure for calculating the resistance of our supply-duct system. Since it is really a continuation of the design procedure on page 468, we shall continue the step numbers where we left off.

8. Trace the longest duct run, *i.e.*, from fan outlet along the duct to the farthest grille. Compute the number of feet of straight duct. (For illustration let us say this came to be 340 ft.)

9. Let us say that in step 2 you based your design on a friction loss of 0.15 in. per 100 ft of length. Now we have a simple formula

$$\left(\frac{L}{100}\right) K_f = \text{static pressure loss due to friction in straight duct} \qquad (4)$$

where L = length of straight duct in feet, from step (8)
K_f = friction loss factor chosen in step (2) and mentioned above

For the values assumed in step (8) and above, we would have

$$\left(\frac{340}{100}\right)(0.15) = 0.51 \text{ in. static pressure loss}$$

10. Obtain the air velocity through each elbow in the longest duct run, which you have traced in step 8. Obtain the velocity head in each of these elbows; see Tables 3 and 4, Chap. XXII. Obtain the "per cent of velocity head lost" in each elbow, from step 7; this will be the same for regular elbows[1] if you have made them as instructed in step 7. For illustration, let us suppose you have six elbows, with velocity heads as shown in the table below. The table shows how you would calculate static head loss due to elbows.

[1] Excepting square elbows with duct turns, if any.

TOTAL STATIC LOSS DUE TO ELBOWS

Elbow	(1) Velocity head in elbow, in.	(2) Per cent of velocity head lost, from step 7	(3) Col. (1) × col. (2) static pressure loss through elbow, in.
1	0.20	10	0.020
2	0.15	10	0.016
3	0.13	10	0.013
4	0.11	30 bad elbow	0.033
5	0.10	10	0.010
6	0.08	10	0.008

Note: When more experienced, you may take a rough short-cut for step 10 without serious error, thus:

Rough average velocity head, 0.13 in.

Five elbows, all with loss of 10 per cent of a velocity head.

Then

$$5(0.10)(0.13) = 0.065 \text{ in.}$$

Plus bad elbow, calculated

$$\text{value} = \underline{0.033} \text{ in.}$$

$$\text{Total elbow static loss} = 0.098 \text{ in.}$$

or approximately 0.10 in.

11. From manufacturer's data obtain the static loss through any equipment. Usually there would be none in a *supply* duct, but we shall *pretend* to have a one-row reheat coil, static drop given as 0.12 in., at our velocity.

12. Get static drop required to overcome resistance of supply grilles, obtained from grille manufacturer. Let us pretend this is 0.04 in. of water.

13. *Summary*. Add the statics found in steps 9 to 12. This is the answer as to the static needed at the fan outlet for this duct system, although we have only calculated the longest branch.[1] For the values we have been taking for illustration, we would have

Friction loss in straight ducts....	0.51 in.
Elbow loss....................	0.10 in.
Equipment....................	0.12 in.
Grilles.......................	0.04 in.
Total static resistance........	0.77 in. of water

The Velocity Method of Duct Design. On page 467 we hinted that this method of duct design does not start with any considerations regarding

[1] If a shorter branch has exceptional impact losses, its over-all loss may be greater than that calculated for the longest branch. This can be found only by calculating the resistance of such other branch.

friction loss. This is true, and this means that we do not use Figs. 1, 7, and 8 in the first steps with this method. In fact, we do not use Figs. 7 and 8 at all. The proper selection of the various velocities that we choose, based upon a carefully considered guess as indicated in step 2, page 468, is of the utmost importance in this method of duct design. This is especially true because this method of duct design usually applies to smaller systems, and these smaller systems usually have smaller ducts and involve smaller air quantities. From Fig. 1, you know that a small duct may have as high a pressure loss due to friction at a velocity of 1,000 fpm as a larger duct would have at a velocity of 2,000 fpm. Therefore be careful with step 2, page 468, to avoid extensively high friction losses. Care with regard to this point will also be helpful in avoiding undesirable noise, because in these small duct systems the fan outlet is usually quiet near the first supply grille.

Now for our procedure of duct design based on the velocity method.

1. This is exactly the same as the first step in the equal-friction method. You will find this on page 468.

2. If your fan is not located inside a factory-assembled conditioner, you have something to say about the fan-outlet velocity selected. Choose a fan with an outlet velocity not higher than 1,400 fpm, if possible. Choose rectangular duct sizes such that you reduce these velocities immediately beyond the fan outlet, by enlarging the duct size, to about 1,200 fpm. At each point where the duct divides into two branches, reduce the velocity somewhat, 5 to 10 percent. For very simple systems, reduce to a velocity of 800 to 900 fpm directly beyond the fan and select rectangular ducts such that the velocity remains constant throughout the system.

3. Select the duct run, from the fan outlet to a remote grille, which looks as though it will have the highest friction loss. Select also the run that looks as though it will have the next highest friction loss. Both of these must be calculated for friction loss according to the method given in the steps to follow. *Usually*, the two ducts chosen as apparently having the greatest pressure loss will be *longest* and *next longest duct run*. However, a shorter run of small duct, at high velocity, having several short radius elbows, might be found to have a greater pressure loss than the longest runs. If so, you must discover this fact because our aim is to calculate the pressure loss for the run having the greater pressure loss. *Compare this method thus far with the equal-friction method. In this method we already have determined our duct sizes but know nothing about friction losses. In the other method, we started out working on a basis of friction losses consistent with good design and did not discover the resulting duct sizes until later on.*

4. Convert the straight duct sections of the runs that you have selected in step 3 to equivalent circular ducts (use Table 3).

5. From Fig. 1 calculate the frictional resistance of each of these straight runs per 100 ft of length. Measure the actual length of each run, and get the actual static-pressure drop due to friction in the straight run.

6. Calculate the resistance of elbows in each of the runs upon which you are working (see step 10 of the equal friction method, page 469).

7. From manufacturer's data obtain the static loss through any equipment.

8. Add the static-pressure drops you have determined in steps 5 to 7. (We need not worry about static-pressure drop due to equipment as there will be nothing on the discharge side of the fan in the case of a small simple duct system.) Do this for both duct runs that you selected as possibly having the highest over-all static resistance. Use the larger value as the static pressure which the fan must produce at its outlet.

Requirements for Dampers. In order to balance air flow throughout the system and to be sure of getting the desired air flow in each branch, it is absolutely essential to provide dampers or splitters in each branch. These may be *center*-pivoted volume dampers located at any point in the branch, such as those shown in Fig. 9. They may be *splitter-type dampers,* as shown at each junction point of the ductwork in Fig. 10. Only by the use of such dampers, while measuring the air flow with an anemometer or velometer, can the design air flow be achieved.

Design of Ductwork on Inlet Side of Fan. The recirculation duct connects to the inlet side of the fan. The recirculation air duct should be designed, based upon the same methods used for designing a supply-air duct. This means that if the recirculation duct is short and has few branches we may use the velocity method. We may do this even though we have designed the supply ductwork by means of the equal-friction method—supply-air duct systems are more extensive than return-air-duct systems. If, on the other hand, a return-air-duct system is sufficiently complicated, it is best to design it using the equal-friction method.

The duct work connecting from the fresh-air intake to the conditioner and on to the fan is almost always very short. It may be based upon a velocity of 900 to 1,200 fpm and may be designed on the velocity method. If this duct is less than 50 ft long, we often neglect the resistance of the duct itself. This is done without appreciable error because we often have the following equipment and apparatus along this duct:

	Static-pressure Drop
Inlet louvers	0.1–0.3
Filters	0.1–0.6
Steam-heating coils, if any	0.1–0.3
Air washer or cooling coil	0.2–1.5

As you can see, the resistance of a 50 ft long fresh-air duct would be small compared with equipment resistances, even for the minimum values tabulated above. *If the resistance of the fresh-air-intake duct, up to the point where it joins the recirculation duct, is greater than that of the recirculation duct, a splitter or other damper is required at that point to create increased resistance in the path of the recirculated air.*

Effect of Duct Design upon Fan Selection. On page 443 we saw that we needed the following information in order to purchase a fan:

1. Number of cubic feet per minute of air to be handled.
2. Over-all static resistance of duct system.

Having calculated the static resistance of our supply-air-duct system and also of our duct system on the suction side of the fan, *we add these two values and we have the grand total over-all static resistance against which the fan must move air.*

In addition to items 1 and 2 in the preceding paragraph, we must bear in mind two other facts in selecting a fan. (1) Any fan should be selected such that it operates near its maximum efficiency. If you select a fan and it is at an inefficient point of operation, you will find that another size of fan will operate at a more efficient rating point. (2) The outlet velocity of the fan must be determined, consistent with the duct velocity in your main duct just beyond the fan outlet. It would be ideal to have the fan-outlet velocity exactly the same as your main-duct velocity. However, this would usually result in a large-sized uneconomical fan. Therefore, try to observe the following standards regarding the outlet velocity of the fan you select:

Make the fan-outlet velocity as near the main-duct velocity as is consistent with fan efficiency and cost of the fan, in no event having an outlet velocity more than twice main-duct velocity.

PART 2 PIPING FOR PUMP-CIRCULATED WATER

Friction Losses. In previous chapters you have had experience in calculating the number of gallons per minute of chilled water required for air washers and other air-conditioning applications. Ordinarily chilled water, cooling tower water, and even warm water for heating must be moved about through a piping system by means of a pump. For any certain flow of water, a wide variety of pipe sizes may be used, the disadvantage of too small pipes being that the pumping power required will be very great. This pumping requirement, which is determined by the amount of friction and impact loss in the piping system, is of importance not because it affects the practical operation of a water circulating system, but because it affects the cost of the pump and the cost of the power to run the pump.

No exact standard may be laid down with regard to permissible friction loss in water piping, as we did in the case of duct work. The total length of piping connected to this pump was probably less than 15 ft. Very small pipe could have been used, involving a very high friction loss *per foot of pipe;* yet the friction loss against which the pump would have to operate would be very small because of having so few feet of pipe length. On the other hand, with a cooling-tower piping system involving perhaps 800 ft of pipe and a water flow of 800 to 1,500 gpm, it would be essential that the piping be selected with a fairly low frictional resistance to keep down the cost of buying and operating the circulating pump.

Friction loss in air ducts was found expressed as static-pressure loss in inches of water, per 100 ft of straight duct. In the case of piping, the friction loss is expressed sometimes in pounds per square inch. In piping design work and friction calculations, however, friction loss is always expressed in static pressure loss *in feet of water, per* 100 *ft of straight pipe.*

Let us set up an approximate design standard for water piping. *For water flows less than* 100 *gpm use a friction head loss per* 100 *ft of pipe not greater than* 10 *ft of water column; better if not more than* 6 *ft per* 100 *ft; for water flows greater than* 100 *gpm, use a friction head loss per* 100 *ft of not more than* 7.5 *ft of water column; better if not more than* 4.5 *ft per* 100 *ft.* In applying these design standards, stay close to the lower limit if you have a great length of pipe with many fittings. The upper limits set may be exceeded a little when you have a total pipe run of less than 100 ft, with a minimum of pipe fittings and valves. On page 481 a practical pump problem is solved two ways, one based on too small pipe, the other based on properly sized pipe. After you have completed this chapter and studied the solution of these typical problems, you will see clearly the importance of using adequately large pipe.

Use of Pipe Friction Tables. Table 5 gives the friction loss, at various gallons per minute, for 1- to 6-in. pipes. Between the two the velocity head is given in *feet of water.* (For definition of velocity head, see Chap. XXII. Velocity head has the same meaning for any flowing fluid.) Suppose that you have to pump 140 gpm through a pipe 600 ft long. Under 3 in. pipe in Table 5, find 140 gpm. To the right of figure 140, read

Velocity head...... 0.63 ft of water
Friction loss....... 9.2 ft per 100 ft of straight pipe

For a single pipe of this length, this friction loss would be permissible. However, a 4-in. pipe would result in a friction loss of only 2.28 ft per 100 ft of pipe. The over-all friction loss for the two cases would be found as follows:

$$\text{3-in. pipe } \frac{600}{100}(9.2) = 55.2 \text{ ft friction loss}$$

$$\text{4-in. pipe } \frac{600}{100}(2.28) = 14.7 \text{ ft friction loss}$$

TABLE 5.—PIPE FRICTION AND VELOCITY HEAD

Corresponding to '17-year pipe." For new and smooth iron pipe the head loss will be 0.7 of that shown

U.S. gpm	Velocity Head	Head loss, ft per 100 ft	U.S. gpm	Velocity head	Head loss, ft per 100 ft	U.S. gpm	Velocity head	Head loss, ft per 100 ft	U.S. gpm	Velocity head	Head loss, ft per 100 ft	U.S. gpm	Velocity head	Head loss, ft per 100 ft	U.S. gpm	Velocity head	Head loss, ft per 100 ft	U.S. gpm	Velocity head	Head loss, ft per 100 ft	U.S. gpm	Velocity head	Head loss, ft per 100 ft
1-in. pipe			1½-in. pipe			2-in. pipe			2½-in. pipe			3-in pipe			4-in. pipe			5-in. pipe			6-in. pipe		
3	0.02	1.26	4	0.01	0.26	6	0.01	0.20	8	0.00	0.11	10	0.00	0.07	20	0.00	0.06	30	0.00	0.04	40	0.00	0.03
4	0.03	2.14	5	0.01	0.40	8	0.01	0.33	10	0.01	0.17	15	0.01	0.15	25	0.01	0.09	40	0.01	0.08	50	0.01	0.04
5	0.05	3.25	6	0.01	0.56	10	0.02	0.50	12	0.01	0.24	20	0.01	0.25	30	0.01	0.13	50	0.01	0.11	60	0.01	0.06
6	0.08	4.55	7	0.02	0.74	12	0.20	0.79	14	0.01	0.23	25	0.02	0.38	35	0.01	0.17	60	0.02	0.16	70	0.01	0.08
8	0.14	7.8	8	0.02	0.95	14	0.03	0.94	16	0.02	0.41	30	0.03	0.54	40	0.02	0.22	70	0.02	0.21	80	0.01	0.11
10	0.22	11.7	9	0.03	1.18	16	0.04	1.20	18	0.02	0.50	35	0.04	0.71	50	0.03	0.34	80	0.03	0.27	90	0.02	0.14
12	0.31	16.4	10	0.04	1.43	18	0.05	1.49	20	0.03	0.61	40	0.05	0.91	60	0.04	0.47	90	0.03	0.34	100	0.02	0.17
14	0.42	22.0	12	0.06	2.01	20	0.06	1.32	25	0.04	0.92	50	0.08	1.38	70	0.05	0.63	100	0.04	0.41	110	0.02	0.21
16	0.50	28.0	14	0.08	2.68	25	0.10	2.73	30	0.06	1.29	60	0.12	1.92	80	0.06	0.81	120	0.06	0.58	125	0.03	0.26
18	0.70	35.0	16	0.10	3.41	30	0.15	3.84	35	0.08	1.72	70	0.16	2.57	90	0.08	1.00	140	0.08	0.76	140	0.04	0.32
20	0.86	42.0	18	0.13	4.24	35	0.20	5.1	40	0.11	2.20	80	0.20	3.28	100	0.10	1.22	160	0.11	0.98	160	0.05	0.40
25	1.39	64.0	20	0.16	5.2	40	0.26	6.6	50	0.17	3.32	90	0.26	4.08	120	0.15	1.71	180	0.13	1.22	180	0.07	0.50
30	1.92	89.0	22	0.19	6.2	45	0.38	8.2	60	0.24	4.65	100	0.32	4.96	140	0.20	2.28	200	0.17	1.48	190	0.07	0.55
35	2.95	119.0	24	0.22	7.3	50	0.40	9.9	70	0.33	6.2	120	0.46	7.0	160	0.26	2.91	220	0.20	1.77	200	0.08	0.61
40	3.42	152.0	26	0.26	8.4	55	0.49	11.8	80	0.43	7.9	140	0.63	9.2	180	0.33	3.61	240	0.24	2.08	220	0.09	0.73
1¼-in. pipe																							
4	0.01	0.57	28	0.30	9.7	60	0.58	13.9	90	0.54	9.8	160	0.82	11.8	200	0.41	4.4	260	0.28	2.41	240	0.11	0.87
5	0.02	0.84	30	0.35	11.0	65	0.68	16.1	100	0.66	12.0	180	1.04	14.8	220	0.49	5.2	280	0.33	2.77	260	0.13	1.00
6	0.03	1.20	35	0.47	14.7	70	0.79	18.4	120	0.95	16.0	200	1.28	17.8	240	0.58	6.2	300	0.37	3.14	280	0.16	1.14
7	0.03	1.59	40	0.62	18.8	75	0.91	20.9	140	1.30	22.3	220	1.55	21.3	260	0.69	7.2	320	0.42	3.54	300	0.18	1.30
8	0.05	2.03	45	0.78	23.2	80	1.04	23.7	160	1.70	29.0	240	1.84	25.1	280	0.79	8.2	350	0.51	4.19	320	0.20	1.47
10	0.07	3.05	50	0.96	28.4	90	1.31	29.4	180	2.15	35.7	260	2.16	29.1	300	0.91	9.3	300	0.66	5.4	350	0.24	1.70
12	0.10	4.3	55	1.17	34.0	100	1.62	35.8	200	2.66	43.1	280	2.51	33.4	320	1.04	10.5	450	0.84	6.7	380	0.28	2.00
14	0.14	5.7	60	1.39	39.6	110	1.96	42.9	220	3.22	52.0	300	2.88	38.0	340	1.17	11.7	500	1.04	8.1	400	0.32	2.20
16	0.18	7.3	65	1.62	45.9	120	2.33	50.0	240	3.82	61.0	320	3.28	42.8	360	1.31	13.1	550	1.26	9.6	450	0.40	2.74
18	0.23	9.1	70	1.88	53.0	130	2.73	58.0	260	4.48	70.0	340	3.71	47.9	400	1.62	16.0	600	1.49	11.3	500	0.50	2.90
20	0.28	11.1	75	2.17	60.0	140	2.17	67.0	280	5.20	81.0	360	4.15	53.0	450	2.05	19.8	650	1.75	13.2	550	0.60	3.96
25	0.45	16.6	80	2.46	68.0	150	3.64	76.0	300	5.98	92.0	380	4.62	59.0	500	2.53	24.0	700	2.03	15.1	600	0.72	4.65
30	0.65	23.0	85	2.78	75.0	160	4.14	86.0	320	6.80	103.0	400	5.11	65.0	550	3.06	28.7	750	2.34	17.2	700	0.98	6.21
			90	3.09	84.0	170	4.67	96.0	340	7.68	116.0	420	5.64	71.0	600	3.65	33.7	800	2.66	19.4	800	1.28	7.96
			95	3.47	93.0	180	5.23	107.0	360	8.60	128.0	440	6.20	77.0	650	4.28	39.0	850	2.99	21.7	900	1.62	9.92
																					1,000	1.99	12.02

SOURCE: Courtesy of the Ingersoll-Rand Pump Company, New York, N.Y.

Note that the 3-in. pipe has a friction loss almost four times as great as the 4-in. pipe.

Pressure losses through valves, elbows, tees, and other pipe fittings are actually *impact losses.* In water-pipe head-loss calculations, the loss through fittings is usually considered a friction loss and so named, even though it is really an impact loss. In the case of these pipe fittings (including elbows), we do not have to figure velocity head and then calculate the percentage of velocity-head loss, as we did in the case of ducts. Instead

Horsepower Variation With Specific Gravity. To obtain power required for pumping a liquid of specific gravity differing from that of water, multiply power required when pumping water by specific gravity of liquid being pumped.

Effect of Viscosity. Viscous liquids tend to increase pump horsepower, reduced efficiency, head and capacity.

Table 6 shows for common pipe fittings the friction of each fitting, as the *length of straight pipe to which its friction is equal.* In Table 6 the different fittings included are listed in the various columns. The column at the extreme left gives the different sizes. Let us consider the head loss caused by a gate valve, a very common fitting. For 1½-in. pipe, a gate valve has the same friction loss as 1.09 ft of straight 1½-in. pipe. A 6-in. gate valve has a friction resistance the same as 5.72 ft of straight 6-in. pipe. Note that for all fittings in Table 6 the larger the pipe, the greater the pressure loss through the fitting, in terms of number of feet of pipe whose resistance equals the resistance of the fitting.

Design of Water Piping Systems and Calculation of Friction. Below we shall give the procedure somewhat similar to that which has been given in the case of duct work. However, in the case of piping we have only one procedure. Before setting up this procedure, let us investigate the effect of so-called "static head" or "unbalanced water leg." Ordinarily pumps draw water from a tank or sump (or basin). The water passes through the pump, is forced through piping, and is discharged at some point that is almost always considerably higher in elevation than the surface of the water in the sump from which the pump draws its water. In simplest terms the net static head or unbalanced water leg simply means this vertical difference in water level between the suction sump and the discharge point. If at the discharge point the discharge pipe does not spill freely into atmosphere, but instead discharges through small openings such as spray nozzles, a certain back pressure is created. This back pressure has the same effect as unbalanced water leg; it must be expressed not in pounds per square inch, but in *feet of water,* and must be added to the static head.

An example of the meaning of static head (or unbalanced water leg) is clearly given in Fig. 11. Refer to the notation in the lower right portion

TABLE 6.—FRICTION LOSSES THROUGH SCREW-PIPE FITTINGS IN TERMS OF EQUIVALENT LENGTHS OF STANDARD PIPE

Nominal pipe size, in.	Actual inside diameter, in.	Gate valve	Long-sweep elbow or on run of standard tee	Medium sweep elbow or on run of tee reduced in size ¼	Standard elbow or on run of tee reduced in size ½	Angle valve	Close return bend	Tee through side outlet	Globe valve	Check valve (approx.) varies with type and make
Factor of resistance		0.25	0.33	0.42	0.67	0.90	1.00	1.33	2.00	
½	0.662	0.335	0.442	0.56	0.89	1.20	1.34	2.79	2.68	4.0
¾	0.824	0.475	0.627	0.79	1.27	1.71	1.90	2.52	3.80	5.7
1	1.049	0.640	0.844	1.07	1.72	2.30	2.56	3.40	5.12	7.7
1¼	1.38	0.902	1.19	1.51	2.42	3.24	3.61	4.80	7.22	11.0
1½	1.61	1.09	1.43	1.83	2.92	3.92	4.36	5.79	8.72	13.0
2	2.06	1.49	1.96	2.50	3.99	5.36	5.96	7.92	11.92	18.0
2½	2.46	1.86	2.46	3.13	5.00	6.72	7.47	9.93	14.94	22.0
3	3.06	2.46	3.25	4.11	6.66	8.87	9.86	13.11	19.72	30.0
4	4.026	3.44	4.53	5.77	9.22	12.37	13.70	18.28	27.50	41.0
5	5.047	4.57	6.00	7.68	12.20	16.47	18.30	24.33	36.60	55.0
6	6.065	5.72	7.55	9.61	25.30	20.61	22.90	30.45	45.00	65.0

Foot valve loss is zero, provided foot valve has an area of 150 per cent of suction pipe.
SOURCE: Courtesy of The Ingersoll-Rand Pump Company, New York, N.Y.

TABLE 7.—CHARACTERISTICS OF LIQUIDS

Liquid	Specific gravity at 60°F/60°F	Viscosity S. S. U.
Beer	1.01	32 at 68°F
Brine—calcium chloride	Up to 1.3	32 to 42 at 68°F
Brine—sodium chloride	Up to 1.2	32 to 36 at 60°F
Fuel oil—Nos. 1 and 2	0.825–0.95	35 to 45 at 100°F
Gasoline	0.721–0.731	30 at 68°F
Kerosene	0.81	35 at 68°F
Milk	1.03–1.04	32 at 68°F
Water, fresh	1.0	31.5 at 60°F

SOURCE: Courtesy of The Ingersoll-Rand Pump Company, New York, N.Y.

of Fig. 11. If the pump is connected to the suction well or sump *below the pump,* we have

$$\text{Static head} = H_D + H_S$$

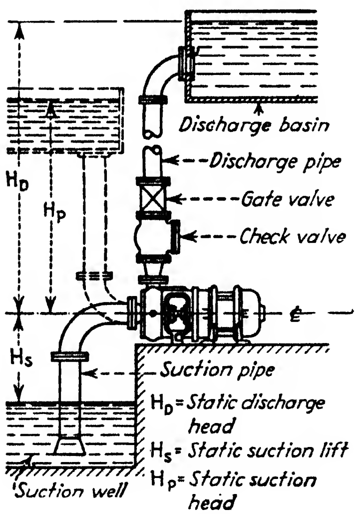


FIG. 11.—The meaning of head on a pump. (*Ingersoll-Rand Co.*)

If, on the other hand, the pump suction is connected as shown by the dotted pipe to a suction basin located above the pump, we have

$$\text{Static head} = H_D - H_P$$

The following is our procedure for designing piping systems and calculating their frictional resistance.

1. It is assumed that you have in earlier calculations determined the number of gallons per minute you must pump. Also you know the point of which the pump inlet must be connected. Now locate your pump at a convenient point.

2. Draw a light line representing the center line of the piping you are to run. This is much easier than in the case of duct work because elbows in piping are not such serious obstructions as in duct work, and because water piping is very small in diameter compared with duct work for the same air-conditioning job.

3. Choose the pipe size that you are going to use (see page 474 and Table 5). Note the head loss per 100 ft of straight pipe, and note also the velocity head

Start at the discharge head and proceed with steps 4 through 9.

4. Measure the length of straight pipe in feet.

5. Count the number of gate valves, elbows, and other fittings.

6. From Table 6 obtain the equivalent feet of pipe that would cause the same resistance as the fittings in the line (see page 477).

7. Add your answers to steps 5 and 6; the result is the *total length for figuring friction.* Multiply this total length of pipe by the friction loss per 100 ft; divide by 100; the result is the *total discharge friction loss.*[1]

8. Determine the static discharge head, *i.e.*, H_D (see Fig. 11).

9. Add the resistance figures determined in feet as a result of steps (7) and (8); the result is the *total discharge head.*

In steps 4 to 9 you have been working with the piping on the discharge side of the pump and have calculated total discharge head. In steps 10 to 16 an almost identical procedure is given for doing the same thing, starting at the beginning of the suction pipe and working toward the pump inlet.

10. Repeat step 4, working with the suction pipe.

11. Repeat step 5, working with the suction pipe.

12. Repeat step 6, working with the suction pipe.

13. Repeat step 7, working with the suction pipe. In this case we call the result *total suction friction loss.*

14. Rewrite the velocity head originally figured in step 3.

15. Determine the static suction lift. If the water level in the suction sump is below the pump inlet, the suction lift is added to the other losses in the suction line. If the water level in the suction sump is above the pump inlet, there is a positive suction pressure instead of a suction lift, and the value is subtracted from the other losses in the suction pipe.

16. Add the resistances determined (in feet) in steps 13, 14, and 15, taking care regarding the note in step 15. The result is called *total suction lift.* (If there was a positive suction head due to the water level in the suction sump being above the inlet of the pump, the total suction lift calculated according to this step will be negative, *i.e.*, will have a minus sign.)

17. Add the total discharge head from step 9 to the total suction lift from step 16, the result is called *total head.* (If the suction lift as calculated in step 16 has a minus sign, the total head will actually be the difference between the two values instead of their sum.)

[1] If in the course of its travel through the pump-discharge line, the water passes through the tubes of an extended surface coil or a refrigeration condenser or evaporator, or any other heat exchangers, an additional friction loss due to this equipment will occur. Manufacturers of such equipment give tables or graphs stating for various flows the friction drop in feet of water. Such friction loss, if any, must be added in the total discharge friction loss as calculated in step 7.

The above procedure must be followed exactly for all cases except those in which it is clear that there will always be a definite positive suction head at the pump inlet. The reason is that the pump manufacturer requires the suction lift as part of the information necessary to guarantee the pump for service under your operating conditions. In fact you should always go through the procedure as given above so that if the result of step 16 is a negative quantity you can inform the pump manufacturer of the exact value of this quantity. When this quantity comes out with a minus sign it may be called *net positive suction head* (see page 445).

It may have occurred to you that the friction losses in the straight pipe and fittings could be figures for the suction and discharge pipe lines *at the same time*, instead of separately as in our procedure. This is true. To these values we would then add the static head or unbalanced water leg, determined as described on page 478 and Fig. 11. This short-cut method may be used for obtaining approximate figures regarding pumps, or figures for your own information. When actually ordering a pump for a particular job, the calculations should be made according to our standard procedure.

Our standard procedure is that endorsed by the Cameron Pump Division of Ingersoll-Rand. You will note that their solutions to the pump problem follow exactly the procedure that we have given you. After following through the determination of total pumping head in solutions *A* and *B* of the pump problem, the manufacturer then selects the proper pumps for these two sets of conditions from the pump bulletin from which the problem is reproduced.

Problems. An industrial plant wishes to install a pump to lift 200 gallons of water per min at 72°F from a sump to a tank on the roof. The water is to be delivered into the tank at 10 lb pressure. The tank is 58 ft above the pump and the pump is 4 ft above the water level in the sump. The discharge pipe from the pump to the tank is 400 ft long and contains 4 standard elbows, 1 check valve, and 1 gate valve. A 2½-in. discharge line is already installed which the manager would like to use if possible. The suction pipe is 4 in. in diameter, 25 ft long and contains 2 elbows and a foot valve. The pump is to be driven by an electric motor. The current available is 220 volt, 3 phase, 60 cycle. A sketch of the layout is shown in Fig. 12.

The friction loss and velocity head can be obtained from the tables on the next page. For comparison two solutions are given: Solution *A* using 2½-in. discharge pipe and Solution *B* using 4-in. discharge pipe.

Solution A: shows 278.7 ft total head. A 2MRV25 pump would be required to handle 200 gal per min against this head.

Solution B: shows 1086 ft total head. A 2RV7½ pump will be required.

These two problems forcibly point out the savings that a discharge pipe of proper size make possible.

In solution *A* in which 2½-in. discharge pipe was used a 25-hp pump is required. In solution *B* in which 4-in. discharge pipe is used only 7½ hp is required to do the same job.

Discharge Head	Solution A 2½-in. discharge pipe 4-in. suction pipe		Solution B 4-in. discharge pipe 4-in. suction pipe	
Length of discharge pipe		400′		400′
4 ells—equivalent length of pipe	4×5′ =	20′	4×9.22′ =	36.9′
1 check valve equivalent length of pipe	1×22 =	22′	1×41 =	41′
1 valve—equivalent length of pipe	1×1.86 =	1.9′	1×3.44 =	3.4′
Total length for figuring friction		443.9′		481.3′
Friction loss per 100 ft	43.1		4.4	
Total discharge friction loss	$\frac{43.1 \times 443.9}{100}$ =	191.3	$\frac{4.4 \times 481.3}{100}$ =	21.2′
Static discharge head } pump to tank		58′		58′
Static discharge head } tank pressure	10×2.31 =	23.1′		23.1′
Total discharge head		272.4′		102.3′
Suction Lift				
Length of suction pipe		25′		25′
2 ells—equivalent length of pipe	2×9.22 =	18.4′	2×9.22 =	18.4′
Foot valve—equivalent length in feet		0		0
Total length for figuring friction		43.4′		43.4′
Friction loss per 100	4.4		4.4	
Total suction friction loss	$\frac{4.4 \times 43.4}{100}$ =	1.9′	$\frac{4.4 \times 43.4}{100}$ =	1.9′
Velocity head		0.4′		0.4′
Static suction lift		4′		4′
Total suction lift		6.3′		6.3′
Total Head				
Total discharge head		272.4		102.3
Total suction lift		6.3′		6.3′
Total head		278.7′		108.6′

Courtesy of Ingersoll-Rand Company, New York, N.Y.

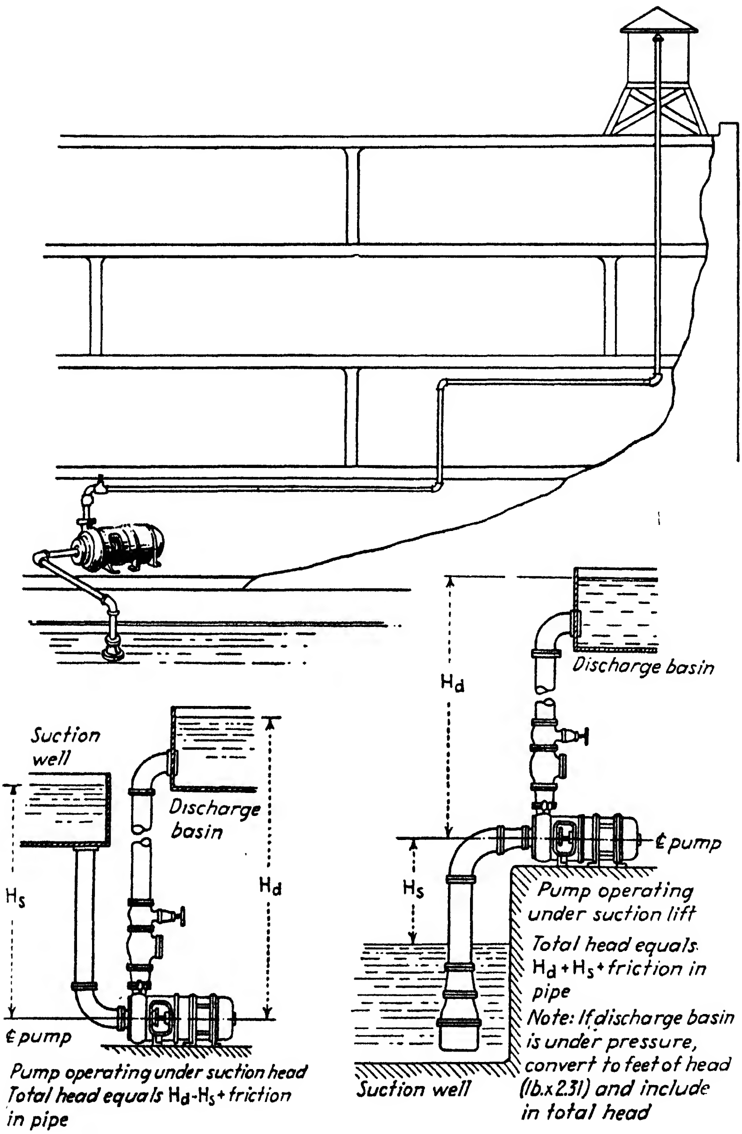


Fig. 12.—Upper section shows piping layout mentioned in problem. Lower left and right show possible variations and definitions. (*Ingersoll-Rand Co.*)

PART 3 FRICTION LOSSES IN REFRIGERANT PIPE LINES

In Chap. XXII we studied fluid flow, realizing that the fluid might be a liquid or a gas. Throughout parts 1 and 2 of this chapter we applied our theory of fluid flow to the design of air-duct and water-piping systems. In the case of refrigerant piping, the suction line and the hot-gas discharge line are really small round ducts carrying gases. However, they are so small that our duct-work design methods would not apply, even if the nature and pressure of the gases did not rule out the use of our duct-design procedure.

The liquid line in a refrigeration piping system is much the same as a water-pipe line, as to friction loss. However, the fluid is quite different from water, the velocities of flow are different, and usually the lines are short and pressure loss is of little importance. Therefore, we shall not attempt to apply our procedure for water-piping systems designed to the simple problem of figuring the proper size liquid line for a refrigeration installation.

Design Standard. As a general statement, we may say that suction-line piping should be selected so that the pressure drop is between 2 and 3 psi for the particular suction line, regardless of its length. Similarly, the pressure drop in the liquid lines should be held at not to exceed 4 to 5 psi. Hot-gas lines (discharge lines) should be limited to a pressure loss of not over 3 to 4 psi. All the foregoing design standards include pressure drops through any shutoff valves, strainers, dryers, etc. For suction lines the figure given includes the pressure drop from the expansion valve on through the evaporator. For hot-gas lines the pressure drop includes the pressure loss through the passages of the condenser.

Bearing in mind the above standards, use Tables 8, 9, and 10 for sizing refrigerant lines. Table 8 applies to hot-gas lines. Table 9 gives the pressure losses in liquid lines. Table 10 gives the pressure loss in suction lines. All three tables apply *exactly* to Freon 12 only, but may be used for methyl chloride without serious error. The use of Tables 8, 9, and 10 should be evident to you after your previous work in this chapter.

Problem. An evaporator coil in an air conditioner using Freon 12 as the refrigerant is located 10 ft above the condensing unit. The head or discharge pressure is 108 psig and the suction pressure at the intake to the compressor is 43 psig. The pressure drop through the evaporator is 14 psi and through the expansion valve 35 psi. What is the allowable pressure drop in pounds per square inch through the liquid line if a 4 psi pressure drop is considered as a good safety factor?

Solution. The difference between the head pressure and the suction pressure is the working pressure which must be dissipated among the various pieces of equipment and the liquid line. Thus,

$$108 - 43 = 65 \text{ psi}$$

TABLE 8.—PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE DISCHARGE OR HOT GAS LINES*

Capacity, Btu per hr	Pressure drop, per 100 ft†									
	Line sizes, in.									
	⅝	¾	⅞	1⅛	1⅜	1⅝	2⅛	2⅝	3⅛	3⅝
10,000	2.3	1.0	0.6							
15,000	4.9	2.0	1.0							
20,000	8.5	3.4	1.7	0.6						
25,000	...	5.3	2.6	0.9						
30,000	...	7.5	3.6	1.2	0.5					
40,000	...	...	6.4	2.1	0.7					
50,000	...	...	9.8	3.1	1.0	0.5				
60,000	...	...	...	4.4	1.3	0.7				
70,000	...	...	...	6.0	1.9	0.9				
80,000	...	...	...	8.0	2.5	1.1				
90,000	...	...	...	10.2	3.1	1.4				
100,000	...	...	...	...	3.8	1.7	0.5			
125,000	...	...	...	...	6.0	2.6	0.7			
150,000	...	...	...	...	8.5	3.8	1.0			
175,000	...	...	...	...	11.6	5.1	1.3			
200,000	...	...	...	...	...	6.7	1.7	0.6		
250,000	...	...	...	...	...	10.4	2.6	0.9		
300,000	...	...	...	...	...	...	3.7	1.2	0.5	
400,000	...	...	...	...	...	...	6.7	2.2	0.9	
500,000	...	...	...	...	...	...	10.5	3.5	1.5	0.7
600,000	...	...	...	...	...	...	...	5.0	2.1	1.0
800,000	...	...	...	...	...	...	...	9.0	3.8	1.8
1,000,000	...	...	...	...	...	...	...	...	5.8	2.9
1,250,000	...	...	...	...	...	...	...	...	9.5	4.4
1,500,000	...	...	...	...	...	...	...	...	...	6.4
2,000,000	...	...	...	...	...	...	...	...	...	11.3

* Soft annealed copper tubing up to and including ¾ in. outside diameter. Hard copper pipe ⅞ in. outside diameter and larger.

† Length of tubing includes the average number of fittings.

SOURCE: "Heating, Ventilating and Air Conditioning Guide," Chap. 23, 1939.

TABLE 9.—PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE LIQUID REFRIGERANT LINES

Capacity, Btu per hr	Pressure drop, psi per 100 ft*			
	Pipe sizes, in.			
	⅞	1⅛	1⅜	1⅝
100,000	0.6			
125,000	0.9			
150,000	1.3			
175,000	1.8			
200,000	2.3	0.6		
225,000	2.9	0.8		
250,000	3.6	1.0		
275,000	4.3	1.2		
300,000	5.1	1.4		
325,000	5.9	1.6		
350,000	6.9	1.8		
375,000	7.9	2.1		
400,000	9.0	2.3	0.8	
450,000	...	2.9	1.0	
500,000	...	3.5	1.3	
550,000	...	4.3	1.5	0.7
600,000	...	5.0	1.8	0.8
700,000	...	6.7	2.4	1.1
800,000	...	8.7	3.1	1.4
900,000	...	...	3.9	1.7
1,000,000	...	...	4.7	2.1
1,200,000	...	...	6.7	3.0
1,400,000	...	..	9.0	4.0
1,600,000	...	...	...	5.1
1,800,000	...	..	...	6.3
2,000,000	...	...	..	7.9
2,200,000	...	...	..	9.2

* Length of tubing includes the average number of fittings.
SOURCE: "Heating, Ventilating and Air Conditioning Guide," Chap. 23, 1939.

TABLE 10.—PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES

Copper pipe actual O.D., in.	Capacity, Btu per hr	Pressure drop, psi per 100 ft*						
		Refrigerant temperature, °F						
		−10	0	10	20	30	40	50
¾	2,000	0.3	0.3	0.2	0.2	0.2	0.1	0.1
	4,000	1.3	1.0	0.8	0.7	0.6	0.5	0.4
	6,000	2.8	2.2	1.8	1.5	1.2	1.0	0.9
	8,000	4.8	3.8	3.1	2.6	2.1	1.8	1.5
	10,000	7.4	5.8	4.8	3.9	3.3	2.8	2.3
	12,000	10.5	8.4	6.8	5.6	4.7	4.0	3.3
	14,000	14.0	11.0	9.1	7.6	6.4	5.4	4.5
	16,000	...	14.5	12.0	9.8	8.3	7.0	5.8
	18,000	...	...	15.0	12.3	10.4	8.7	7.2
	20,000	...	...	...	15.0	12.7	10.7	8.9
1⅛	7,000	0.4	0.3	0.3	0.2	0.2	0.2	0.1
	10,000	1.0	0.7	0.5	0.5	0.4	0.3	0.3
	15,000	1.9	1.5	1.2	1.0	0.8	0.7	0.6
	20,000	3.3	2.6	2.1	1.7	1.4	1.2	1.0
	25,000	5.0	4.0	3.2	2.7	2.2	1.9	1.6
	35,000	9.7	7.7	6.2	5.1	4.3	3.6	3.0
	45,000	15.8	12.6	10.0	8.4	7.0	5.9	4.9
	60,000	...	...	...	14.8	12.2	10.2	8.6
	70,000	...	...	...	...	...	14.0	11.7
1⅜	10,000	0.3	0.2	0.2	0.2	0.1	0.1	0.1
	15,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	20,000	1.2	0.9	0.7	0.6	0.5	0.4	0.4
	30,000	2.6	2.1	1.6	1.3	1.1	0.9	0.8
	40,000	4.6	3.6	2.8	2.3	1.9	1.6	1.4
	50,000	7.0	5.5	4.4	3.5	2.9	2.5	2.1
	60,000	10.0	7.8	6.2	5.0	4.2	3.5	3.0
	80,000	...	14.0	11.0	8.7	7.3	6.2	5.2
	100,000	...	...	...	13.5	11.3	9.5	8.2
1⅝	30,000	1.6	1.3	1.0	0.8	0.7	0.6	0.5
	40,000	2.7	2.1	1.7	1.4	0.1	0.9	0.8
	50,000	4.2	3.2	2.5	2.1	1.7	1.4	1.2
	60,000	6.1	4.5	3.6	2.9	2.4	2.0	1.7
	70,000	8.7	6.3	4.8	3.8	3.1	2.6	2.2
	80,000	...	8.4	6.3	4.9	4.0	3.3	2.8
	90,000	...	...	8.0	6.2	4.9	4.1	3.5
	100,000	...	...	10.0	7.6	6.1	5.0	4.2
	120,000	...	...	...	...	8.6	7.0	5.9
	140,000	...	...	...	...	...	9.5	7.9

* Length of tubing includes the average number of fittings.
SOURCE: A.S.H.V.E., "Heating, Ventilating and Air Conditioning Guide," Chap. 23, 1939.

TABLE 10.—PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES (*Cont.*)

Copper pipe actual O.D., in.	Capacity, Btu per hr	Pressure drop, psi per 100 ft* Refrigerant temperature, °F						
		−10	0	10	20	30	40	50
2⅛	50,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	100,000	2.6	1.8	1.4	1.1	0.9	0.8	0.7
	150,000	5.6	3.9	3.0	2.4	2.0	1.6	1.4
	200,000	9.8	6.7	5.2	4.1	3.4	2.8	2.4
	250,000	14.8	10.3	8.0	6.3	5.1	4.2	3.6
	300,000	...	14.5	11.3	9.0	7.2	6.0	5.0
	350,000	...	19.5	15.3	12.0	9.7	7.8	6.7
	400,000	...	...	19.6	15.3	12.5	10.0	8.5
2⅝	50,000	0.2	0.2	0.1	0.1	0.1	0.1	0.1
	100,000	0.7	0.6	0.5	0.4	0.3	0.2	0.2
	150,000	1.6	1.2	1.0	0.8	0.6	0.5	0.4
	200,000	2.8	2.1	1.7	1.4	1.1	0.9	0.7
	250,000	4.3	3.4	2.6	2.1	1.7	1.3	1.1
	300,000	6.1	4.5	3.7	3.0	2.4	1.9	1.5
	350,000	8.2	6.0	5.0	4.0	3 2	2.5	2.0
	400,000	...	7.8	6.5	5.1	4.2	3.3	2.7
	450,000	...	...	7.7	6.4	5.3	4.0	3.5
	500,000	...	...	...	7.8	6.4	5.0	4.2
	550,000	...	...	...	...	7.7	6.2	5.1
	600,000	...	...	...	...	...	7.4	6.2
3⅛	200,000	1.2	1.0	0.8	0.6	0.5	0.4	0.4
	300,000	2.6	2.0	1.6	1.3	1.0	0.8	0.7
	400,000	4.5	3.4	2.6	2.1	1.7	1.4	1.3
	500,000	7.3	5.4	4.1	3.3	2.7	2.2	1.9
	600,000	...	8.1	6.0	4.7	3.8	3.1	2.7
	700,000	...	...	8.4	6.5	5.2	4.2	3.5
	800,000	...	...	...	8.6	6.8	5.5	4.6
	900,000	...	...	...	...	8.7	7.0	5.9
	1,000,000	...	...	...	...	...	8.9	7.3
3⅝	300,000	1.2	0.9	0.7	0.6	0.5	0.4	0.3
	400,000	2.0	1.6	1.3	1.0	0.8	0.7	0.6
	500,000	3.2	2.5	1.9	1.6	1.3	1.0	0.9
	600,000	4.6	3.6	2.8	2.2	1.8	1.5	1.3
	700,000	6.4	4.9	3.8	3.0	2.5	2.0	1.7
	800,000	8.7	6.4	4.9	3.9	3.2	2.5	2.2
	900,000	...	8.2	6.2	4.9	3.9	3.2	2.7
	1,000,000	...	...	7.7	6.1	4.9	4.0	3.3
	1,100,000	...	...	9.4	7.3	5.8	4.8	4.0
	1,200,000	...	...	...	8.7	6.9	5.6	4.8
	1,300,000	...	...	...	...	8.0	6.6	5.6
	1,400,000	...	...	...	...	9.3	7.6	6.4

* Length of tubing includes the average number of fittings.
SOURCE: A.S.H.V.E., "Heating, Ventilating and Air Conditioning Guide," Chap. 23, 1939.

The drop through the evaporator is 14 psi, leaving a net at this point of $65 - 14 = 51$ psi.
The drop through the expansion valve is 35 psi, leaving a net of $51 - 35 = 16$ psi.
For every foot of height of liquid line above the condensing unit 0.6 psi is allowed, so that we have $0.6 \times 10 = 6$ psi, which must be subtracted from the net head of 16 psi.

$$16 - 6 = 10 \text{ psi}$$

With a safety factor of 4 psi, the allowable pressure drop through the liquid line would then be

$$10 - 4 = 6 \text{ psi} \quad \textit{Ans.}$$

CHAPTER XXV

AIR DISTRIBUTION IN CONDITIONED SPACE

General. We have already learned the importance of supplying the proper amount of conditioned air at the proper psychrometric conditions in order that a conditioned space may be kept at its design conditions. Important as this is, the proper distribution of the conditioned air is of at least equal importance. We quote a sentence from the A.S.H.V.E. "Guide."[1] "Correct air distribution contributes as much or more to the success of a forced air heating, ventilating, cooling or air conditioning system as does any other single factor."

The problem of air distribution is divided into two general problems. (1) There is a problem of dividing the supply of conditioned air between different zones, which we have already learned in Chaps. XIX and XXI. (2) There is a problem of distributing air within a single conditioned space This chapter concerns the second problem.

Definitions. The A.S.H.V.E. has standardized upon certain definitions to which most manufacturers of grilles, registers, and air-distribution equipment conform.[1] Some of the most important of these definitions are listed below:

Supply Opening. Any opening through which air is delivered into a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

Exhaust Opening. Any opening through which air is removed from a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

Outside Air Opening. Any opening used as an entry for air from outdoors.

Grille. A covering for any opening and through which air passes.

Damper. A device used to vary the volume of air passing through a confined cross-section by varying the cross-sectional area.

Multiple-louver Damper. A damper having a number of adjustable blades.

Single-louver Damper. A damper having one adjustable blade.

Face. A grille with provision for attaching a damper.

Register. A face with a damper attached.

Flange. The portion (either integral or separate) of a grille, face, or register extending into the duct opening for the purpose of mounting.

Frame. The portion (either integral or separate) of a grille, face, or register extending around the duct opening for the purpose of mounting.

Margin. The margin of a grille, face, or register is one-half of the difference between the duct dimension and overall dimension measured either horizontally or vertically.

[1] "Heating, Ventilating and Air Conditioning Guide," Chap. 28, p. 565, 1939.

Free Area. The total minimum area of the openings in the grille, face, or register through which air can pass.

Core Area. The total plane area of the portion of a grille, face, or register bounded by a line tangent to the outer edges of the outer openings through which air can pass (also called gross area or total area).

Duct Area. The area of a cross-section of the duct based on the inside dimensions at the point where the grille, face, or register is mounted.

Percentage Free Area. The ratio of the free area to the core area expressed in percentage.

Throw. The distance air will carry measured along the axis of an air stream from the supply opening to the position in the stream at which air motion reduces to 50 f.p.m.

Envelope. The outer boundary of an air stream.

Summer Problem Different from Warm-air Heating. A brief discussion of air distribution for warm-air heating systems in residences is given in Chap. XVII. At the end of Chap. XVII it was pointed out that buildings larger than residences rarely are entirely heated by means of a warm-air system. Since warm-air heating is virtually restricted to residences and other smaller buildings, it is true that, in general, rooms heated by means of warm air are fairly small. In Chap. XVII, it is explained that low supply outlets provide the best means of air distribution for warm-air heating.

Most summer air-conditioning work is in commercial buildings—stores, offices, restaurants, etc.,—where the average size of a single conditioned room may be quite large. Because of the larger sizes of the ordinary conditioned space, and also because conditioned air is usually 15 to 20° *colder* than room temperature, high supply outlets are required for factory distribution in summer air conditioning. It is essential that the cold conditioned air be mixed with the room air before striking occupants. This can be achieved only by having the supply-air outlets well above the breathing level.

When an air-conditioning system is to provide year-round conditioning high supply outlets must be used.[1] Chapter XVII shows the proper method of using high supply outlets if the conditioning system also supplies all the heat by means of warm air. If the conditioning system in wintertime supplies only clean humidified fresh air, warmed to room temperature, the

[1] With one impractical exception, a few elaborate year-round systems have been arranged to operate with high supply openings and low return openings during the summer season. By means of a complicated plenum chamber and dampering, the air flow is reversed during the winter season so that low supply openings and high return openings are then used. For its slight advantage (it is not perfect) this system is impractical because of the high cost and the space required.

special care that would be required to distribute warm air for heating may be omitted. Instead we may, in wintertime, use the distribution system as designed for summer air-distribution systems in accordance with the principles in this chapter.

Distributing Air throughout a Room. Throughout our discussion here we must remember that there are two entirely different air-distribution patterns to consider for every room. (1) We must *look down* upon the room (from the ceiling) and note the path followed by the conditioned air after it leaves the supply grille. In the language of the drafting room, this is called the *plan view.* (2) We must consider the room as if it were cut in half vertically, pretending that we are standing off to one side of the room. We might call this a "side view," or in the language of the drafting room, an *elevation view.* You can easily imagine air distribution in a conditioned space where we supply conditioned air at a high velocity just below the ceiling. If this room were narrow, and if we were to make the mistake of locating our return openings just below the ceiling on the wall opposite the supply openings, conditioned air would shoot across the ceiling of the room and out again. Quite possibly the air down at the breathing level would receive little or no effect from the conditioned air. The "plan view" of such a distributing system could look very nice, but it might only mean that we were blanketing the entire ceiling with conditioned air. The elevation view, however, would show the path of this conditioned air and would show that the conditioned air never had an opportunity to get more than 3 or 4 ft below the ceiling. Draw sketches based upon the description in this paragraph, and you will clearly understand the need for considering both the plan view and the elevation view of air distribution within a room.

The two great difficulties that may be encountered in distributing air are *dead spots* and *high-velocity areas.* Dead spots are those not reached by the conditioned air, and such spots will be too warm in a summer conditioning installation. High-velocity areas are locations in which a stream of cool conditioned air may strike occupants before the stream is slowed down to about 50 fpm. Occupants of high-velocity areas will complain of drafts and will experience a feeling of being too cold. Remember from your comfort-chart discussion in Chap. XIII that a high-velocity air current decreases effective temperature. Therefore, even if the dry-bulb temperature of a high-velocity air current may not be much below room temperature, the occupants will still experience drafts and will feel cold because of excessive air motion.

From the preceding paragraph you can see that our problem of air distribution may be stated somewhat as follows: Distribute air throughout all points of the conditioned space to avoid dead spots; do not have any occupants subjected to an air current of higher velocity than 50 fpm.[1]

The best way to illustrate general methods of ensuring good air distribution, both in plan and in elevation, is to consider a series of sketches. The discussion and the sketches must necessarily be of a *general* nature; every conditioned space must be carefully studied because it will almost always have peculiarities of its own.

Grille Locations. Supply and Return. It is not possible to discuss supply-air outlets without discussing the return openings at the same time. A room having air-supply grilles without any return grilles *must be* provided with some type of opening into a corridor or adjoining room. This opening is necessary to permit air to leave the conditioned room. If you continuously *supply* 3,000 cfm of air to a room, 3,000 cfm *must leave* the room *somewhere.* Some of this air will leak out of cracks around windows and doors if the room has no return-air register. However, this leakage or "exfiltration" is not sufficient, and a "relief opening" must be provided to assure the air a free path by which to leave the room. Once a relief opening is provided, it acts in much the same way as any opening in a recirculation air duct, except that no fan is drawing the air through the relief opening. Remember that you always must consider both the *means of air supply to a room* and the *means of air removal from the room.*

Supply-air *envelopes* as they appear in *plan view* are determined by the type of grille selected. Different manufacturers offer grilles having different standard envelope patterns in plan view. The plan-view envelope pattern is ordinarily called the "deflection" of the particular grille. The charts shown on page 503 illustrate standard deflections available in grilles made by the Hart and Cooley Manufacturing Company. Other manufacturers offer grilles having deflection patterns by means of which substantially the same sort of results may be obtained. However, each manufacturer has only certain deflections which are available as standard. Special deflections are available from most manufacturers, but at extra cost and with delayed delivery. Grilles are also available having their vertical vanes or bars adjustable either individually or in groups of five or six vanes. With such grilles, adjustments may be made in the field after installation to obtain any unusual deflection desired. Fixed deflection grilles cannot be tampered with and lack the flexibility of the adjustable type.

We defined "envelope" as "the outer boundary of an air stream." By this we mean the outer boundary of the stream of air discharged by the

[1] For some special installations having inside design conditions of high dry bulb and low relative humidity, such as 85° dry bulb, 35 per cent relative humidity, air currents of 100 fpm are not only permissible but desirable. Such design conditions are unusual and almost never encountered in ordinary comfort conditioning.

supply grille. The envelope of a supply grille is not a *sharp beam* like that from a focusing searchlight. Some air from the stream discharged by the grille leaves the envelope and mixes with the adjacent room air, causing eddy currents and air motion in the air adjacent to the envelope. Also the location of return-air outlets may affect the pattern of the supply-air envelope both in plan and elevation view. A thorough and reliable knowledge of air flow within a room can be obtained only by experience, but the sketches in Figs. 1 to 10 give you a basic knowledge of what you may expect.

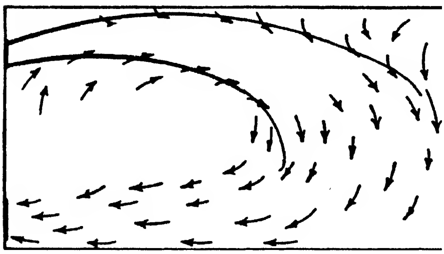


FIG. 1*a*.—Supply grilles high and returns low in same wall.

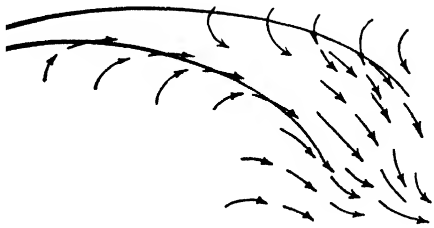


FIG. 1*b*.—Same as Fig. 1*a*, except supply and return in opposite walls.

Figure 1*a* illustrates an elevation (or side view) of a room having high supply outlets and low return openings. The return openings are *correctly* located on the same side of the room as the supplies. This ensures that the conditioned air will be thrown across the room above the breathing level, dropping to the floor at the opposite wall, and being slowly drawn across the room at the breathing level to the return registers. Figure 1*b* illustrates the *wrong* place for a return outlet. It clearly shows that half the room would be rather dead while occupants near the side wall having the return grilles would be subjected to a blanket of cold air. The cold air would not have a proper opportunity to mix with the room air and be warmed in picking up the heat from lights, occupants, transmission, etc. Figure 2 illustrates the room shown in Fig. 1*a*, but shows the *plan view*. In Fig. 2 we have shown three return-air registers, although the middle one might be omitted without serious difficulty.

In many cases, an odd-shaped room is encountered, and return grilles properly located may eliminate the possibility of dead spots in such rooms. Figure 3 illustrates a room, small enough to be covered by one supply grille, with the return grille strategically located to avoid a dead spot. In elevation the room shown in Fig. 3 would appear almost exactly like the one in Fig. 1*a*.

Sometimes it is necessary to supply conditioned air from outlets located in the ceiling. This is particularly true in restaurants, under theater balconies, and in similar locations where the room has masonry walls, but where there is a "hung ceiling" of plaster or sheet metal suspended from the beams. Figure 4 shows a room equipped with three ceiling outlets

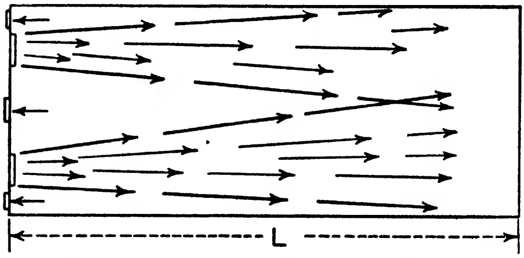


FIG. 2.—Plan view of distribution in Fig. 1*a*.

discharging air in all directions. Note that the coverage is complete in the plan view. Note that in the elevation view there appears to be a dead spot just below the outlets in the center of the room. This dead spot is, however, well above the breathing level, and distribution at the breathing level is complete and satisfactory. Note that although the air may leave the supply outlet at a fairly high velocity the

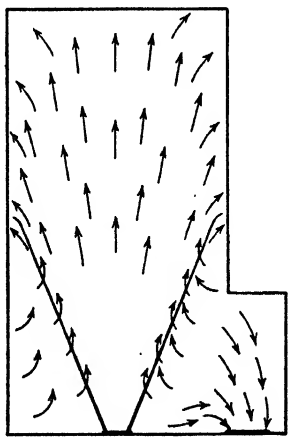


FIG. 3.—Supplying air to an odd shaped room.

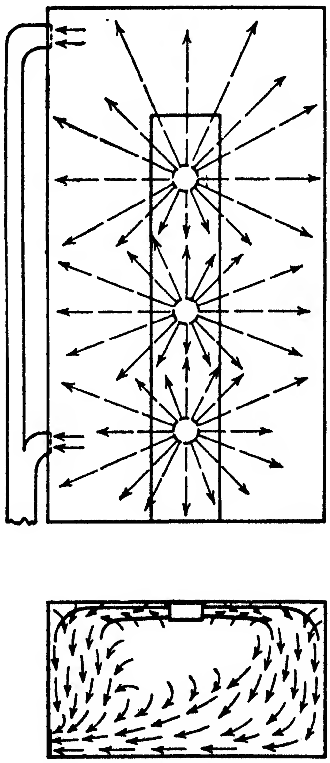


FIG. 4.—Plan and cross section. Room equipped with three ceiling outlets or plaques.

air spreads out and the air motion at the breathing level is very slight.

Figures 5 and 6 illustrate two different types of ceiling supply outlets. That shown in Fig. 5 is about the same as the one shown in plan and elevation in Fig. 4. The arrangement in Fig. 6, with the return outlet built into the supply duct and supply outlet, is unusual and not often used. It is interesting to note the air-flow pattern. The apparent "dead spot" near the floor is more theoretical than real.

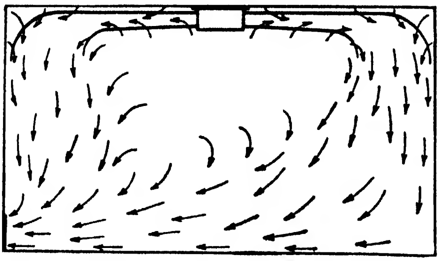


FIG. 5.—Elevation. Air distribution with ceiling outlets in center of room and returns at floor.

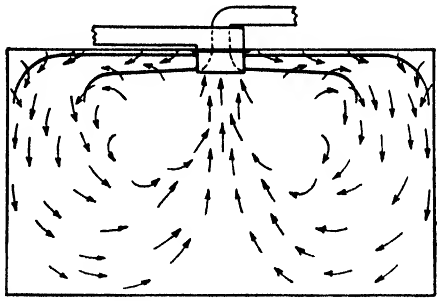


FIG. 6.—Unique method of supplying and returning air from one fixture.

In many cases a conditioned space is too long to be covered by means of supply grilles throwing air the full length of the space from one wall. In such cases the arrangement shown in Fig. 7 provides excellent air distribution. Note that only a single return opening is used. In many cases the distribution system shown in Fig. 7 is used for smaller spaces, because the short duct across the center of the ceiling may be finished off to look like a false beam. If a ceiling-suspended unit is used, it might be located directly above the return grille. The recirculation duct would then be only a short duct extending from the bottom of the unit to the floor. A supply duct would also be a very short duct. Many purchasers prefer this arrangement because of its low cost of ductwork and because unsightly ductwork is reduced to a minimum. This arrangement also has the advantage that it does not destroy top-shelf space on a wall, as would a duct running along the entire length of a wall.

Connections of Ducts to Grilles and Registers. In many cases we have a duct carrying air at a relatively high velocity (say 1,000 fpm) but we require a grille at the end of this duct through which the air must pass at a much lower velocity (say 500 to 700 fpm). In this case the duct must increase in area *very gradually,* or the undersirable effect shown in Fig. 8 will result. As Fig. 8 shows, the air will flow outward through the middle portion of the grille, while the outside of the grille will be dead or may even be subject to room air being drawn inward. Angle A in Fig. 8 should be as low as 5 to 7 deg to avoid the undesirable result shown in the figure. Deflecting vanes in the duct will help to distribute the air evenly over the face of the grille if angle A cannot be made as small as 5 to 7 deg.

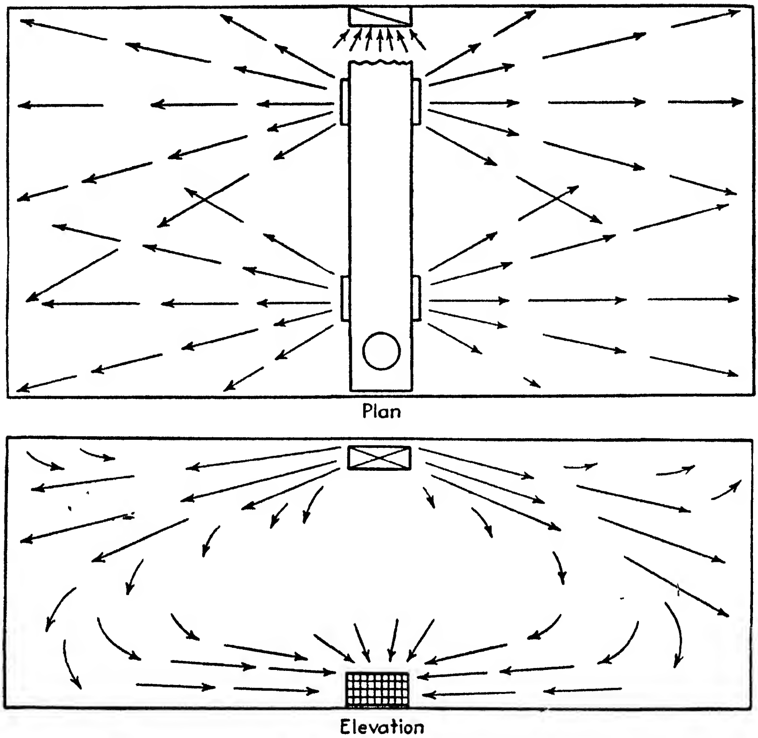


FIG. 7.—Another method of air distribution.

Frequently supply grilles are located at the top (or bottom) of a riser. (A riser you will recall is any vertical duct or pipe.) The simple construction shown in Fig. 9 for this type of installation gives the undersirable results shown. The square-elbow installation, shown in Fig. 10, utilizing a set of duct turn vanes is much more satisfactory.

Air-stream Drop. The various elevation views in Figs. 1 to 7 indicate that a supply of cool air, thrown horizontally from a high supply outlet, will *drop somewhat* as it *travels* farther from the grille and loses velocity. The amount of this drop is usually measured *in feet below the center line of the grille,* at throws of various length. The higher the velocity at which the air leaves the grille, the less it will drop for each foot of throw. The cooler the air compared with room temperature, the greater the drop per foot of throw. It is possible to calculate air-stream drop for any particular grille. For practical purposes however, a general chart may be prepared showing this information in tabular form. We present this as Chart 8, page 511.

Grille Velocity and Length of Throw. Refer to the deflection diagrams shown on page 503. Suppose that you wished to use type *A* deflection but have a room only 40 ft long. Under these conditions, you would not want to take advantage of the maximum possible throw of the grille, because if you did, the air would strike the opposite wall at too high a velocity. This would produce drafts and eddy currents near this wall. Instead it is necessary that the throw *be* 40 *ft,* the width of your room, such that the air will have slowed to approximately 50 fpm upon striking the wall opposite

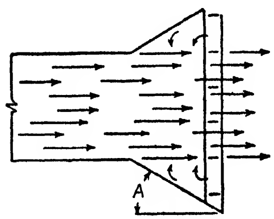


FIG. 8.—Angle *A* should not be greater than 30 degrees.

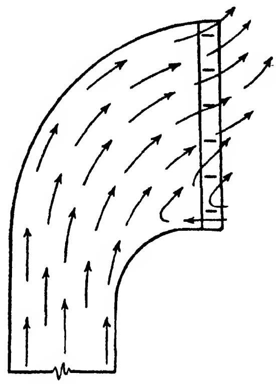


FIG. 9.—Deflecting vanes at grille face will evenly distribute air across the opening.

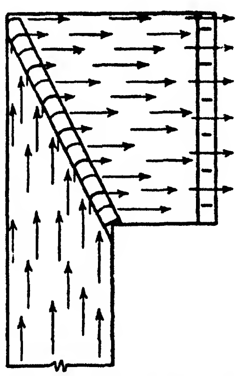


FIG. 10.—Method of distributing air around square elbows on approach to grille face.

the grille. The table covering the various types of deflection is shown on page 506. Data are given for *any volume* of air from 50 to 3,000 cfm through a single supply outlet. Various lengths of *throw from* 10 *to* 80 *ft* are shown. For every practical air quantity and throw, the table shows the required *free area of grille in square inches* and the *velocity in feet per minute* at which air must leave the grille to achieve the throw desired.

We do not know of any other manufacturer presenting this technical data in a manner as clear and easy to use as the Hart and Cooley data.

Grille Resistance. Air-conditioning supply grilles and registers are actually plates of metal in which are hundreds of openings, each of which acts just like the simple orifice shown on page 412. This means that there is a static pressure drop through the grille, this static pressure being used partly to overcome friction and partly to give increased velocity head to the air. Grille manufacturers present data regarding the friction of their grilles at various velocities, both in the form of tables and graphs. Since the tables involve interpolation, and because extremely accurate values are not necessary, the graphs are more convenient to use. In Chart 10, on page 513, we present the Hart and Cooley resistance chart relating to their current line of grilles. Refer to this chart, and note particularly that the design 88 register with built-in turning vanes smooths out turbulences so effectively that its resistance is less than that of an empty "stack head." (The construction shown in Figs. 9 and 10, without the grille, is called a "stack head.")

Air-outlet Noise. The measurement of noise is a technical subject about which no accepted engineering standards have yet been stated and agreed upon by all concerned. The American Standards Association has published bulletin Z24.3–1936, entitled "American Tentative Standards for Sound Level Measurements." The accepted technique of sound-level measurement and the type of instrument in accepted use conform to this A.S.A bulletin. Those interested in the technical data regarding noise level are referred to the A.S.A. and also the acoustical division of the Burgess Battery Company and the makers of Celotex acoustical material for publications regarding noise and the measurement of noise level.

In actual air-conditioning work, noise-level measurement is rarely used. Equipment such as compressors has a far more serious noise than air noise through grilles and registers. Ordinarily we try to make noise levels as low as possible. For special classes of work such as broadcasting studios and very fine theaters, consulting acoustical engineers are called in, taking the noise problem out of the hands of the air-conditioning people. These data are presented mainly for your interest. You should remember the maximum allowable velocities (from a noise-level standpoint) presented in Chart 9 on page 512. This chart applies to *Hart and Cooley* grilles, and similar information must be obtained from other manufacturers when their grilles are used.

Selection of Return-air Register. Return-air registers are usually simple stamped grilles, as they do not have to deflect air in any particular direction. The main purpose is simply to conceal a large duct opening. Although properly located return-air intakes may help your distribution, as pointed out in the discussion of Figs. 1 to 7, the location of return-air intakes is relatively unimportant *so long as they are not located so as to cause short-circuiting, as shown in the lower sketch in Fig.* 1.

The free area of return-intake registers may be chosen such that face velocities occur as high as 1,000 fpm. There is little danger of drafts, because with a velocity of 1,000 fpm at the grille face, experimental work indicates velocities below 75 fpm a few inches away from the register face. Care must be taken to prevent seated or standing occupants from being closer than about 18 in. to return openings.

Engineering Data for Grille Selection. Proper selection of size, fan velocity, and type of deflection is much more important in the case of supply grilles than with air-return registers. On the following pages, we present engineering data with explanatory discussion as presented

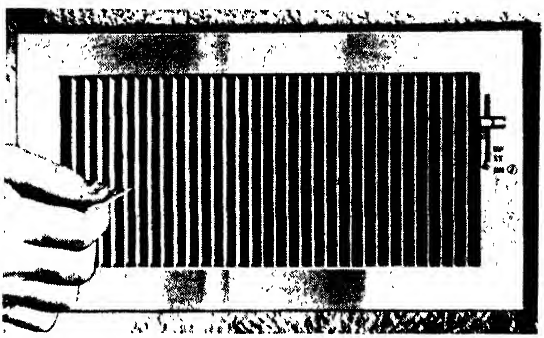


Fig. 11.—No. 88 design supply air grille. (*Hart & Cooley Mfg. Co.*)

by the Hart & Cooley Manufacturing Co. Bear in mind that most of this discussion is based upon air leaving the grilles in a horizontal path; the length of throw and the air stream drop may be varied by deflecting the air stream upward or downward from horizontal. While the adjustable double-deflection grille 88 Hart and Cooley design is exclusive because of several patented features, other manufacturers do offer grilles having adjustable vanes behind the grille face, by means of which the horizontal deflection can be changed. In selecting supply grilles for a carefully planned installation, you must obtain data as complete as that presented on the following pages, from the manufacturer whose grilles you are considering. When working with grille-selection data not familiar to you it is always advisable to have the manufacturer's representative verify your selections to assure that you are interpreting the data correctly. This will also ensure your obtaining the most *economical* grille selection which will perform satisfactorily in your service.

Figures 11 to 15 show various designs of Hart and Cooley grilles.

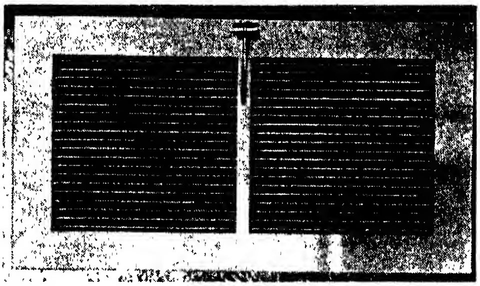


Fig. 12.—No. 74 design supply air grille. (*Hart & Cooley Mfg. Co.*)

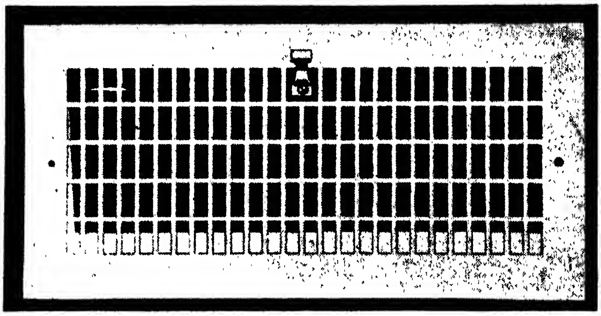


Fig. 13.—No. 69 design air register. (*Hart & Cooley Mfg. Co.*)

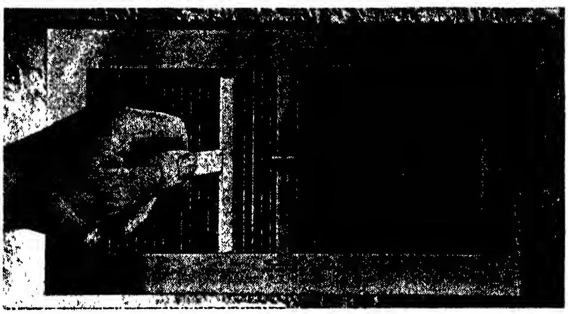


Fig. 14.—No. 75 design air supply grille. (*Hart & Cooley Mfg. Co.*)

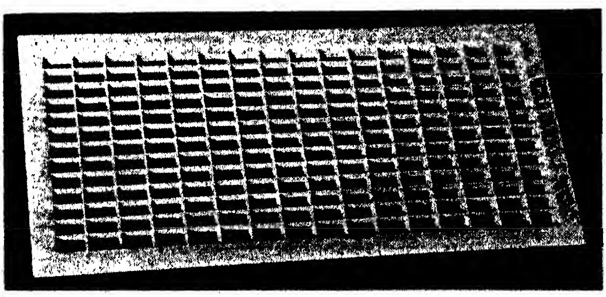


Fig. 15.—No. 265 design return air grille. For floor installation. (*Hart & Cooley Mfg. Co.*)

Register and Grille Selection for Residential Air Conditioning.[1] Proper air distribution is one of the most important factors in the success or failure of an air conditioning system. Therefore, too much emphasis can not be placed on the necessity of properly selecting the location, type, and size of registers used to distribute the air. This involves the following steps:

1. Calculate the temperature and volume of the air required to properly heat or cool each room.
2. Decide whether high sidewall, low sidewall, or baseboard locations will be used (other possible locations, such as floor, ceiling, window sill, and combinations are used so infrequently in domestic installations that they will not be considered.)
3. Decide on the location of the register in the room.
4. Select the type of register to be used.
5. Determine the proper air velocity.
6. Determine the proper size of register.

Inasmuch as most manufacturers of air conditioning equipment furnish sufficient information to enable the installer to determine the proper air temperature and volume of air required, no attempt will be made in this discussion to cover these points.

Sidewall or Baseboard Locations. By "high sidewall" is meant a location 6 ft. or more above the floor. By "low sidewall" is meant a location immediately above the baseboard, and by "baseboard" is meant a location in the baseboard. Since the results obtained from baseboard and low sidewall locations are practically identical, the term "baseboard" will hereafter be used to include both low sidewall and baseboard locations.

With Cooling Systems better results will be obtained with high sidewall than with baseboard locations, as it is almost impossible to avoid objectionable drafts if cold air is introduced into a room near the floor.

With Heating Systems satisfactory results may be obtained with either high sidewall or baseboard locations. The table on page 502 shows some of the items which should be considered in making the choice between high sidewall and baseboard locations.

Location of Register in Room. Ordinarily, the structural details of the building and economical considerations concerning the layout of the duct work are principal factors in determining the location of the registers. Frequently these must be located where conditions permit and not in what might be a more desirable location if there were no restrictions. In general, the following should be followed as closely as possible:

1. High Sidewall Locations. Locate the registers in such a position that the air can be directed to blanket the exposed walls.

2. Baseboard Locations. Locate the registers on an inside wall so that they will interfere as little as possible with the subsequent arrangement of the furniture, and in such a location that none of the occupants of the room will customarily be placed directly in front of the register.

Type of Register. 1. High Sidewall Locations. Cooling or Heating Systems. The air is generally brought to the register from directly below, and unless some

[1] Reprinted in part by courtesy of Hart & Cooley Manufacturing Co.

Item	Baseboard locations	High sidewall locations
Possibility of drafts in occupied zone	Difficult to avoid	May be entirely avoided by using proper air velocity
Arrangement of furniture	May interfere	Out of the way—no interference
Streaking of ceilings	Avoided	May be avoided by using proper type of register
Register temperatures	Must not be less than recommended	Any temperature can be be used
Temperature differential be-between floor and ceiling	Good, even if system is not properly engineered	Excellent if air velocities can be accurately predetermined
Accuracy of engineering required	More flexible, does not require exacting calculations	Correct calculations essential
Possibility of stratification	Slight	Unless system is properly engineered stratification may result
Cost of installation	Short runs, less labor to install stack	Smaller resisters and fittings

provision is made to change the direction of the air flow, the air will continue on its upward path after it leaves the register, and strike the ceiling immediately in front of the register, causing streaking. Some provision should therefore be incorporated either in the register itself or in the stackhead to straighten the air stream. It is also desirable to control the air sideways so that the air can be directed

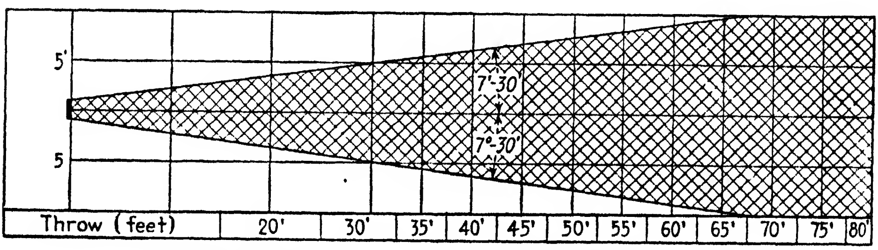


CHART 1A.—Type S deflection. (*Hart & Cooley Mfg. Co.*)

to the exposed portion of the room. A register such as No. 88 Design, combining control of the air flow on both a horizontal and vertical plane, is recommended. Charts 1–7 show the air distribution to the left or right with various types of deflections. From these charts determine the deflection that will most completely blanket the area to be conditioned. Even though the register is of the adjustable type, with which the deflection can be changed as desired after installation, it is advisable to pre-determine the general type of deflection which will be used, as this affects the velocity required, which in turn governs the size of the register used.

2. Baseboard Locations. *a.* **Cooling Systems.** Baseboard registers if used with cooling systems should have provision for directing the air upward as rapidly as possible so that the cool air will not strike the occupants of the room.

b. Heating Systems. Baseboard registers used with heating systems should have provision for directing the air toward the floor and thus diffusing it with the

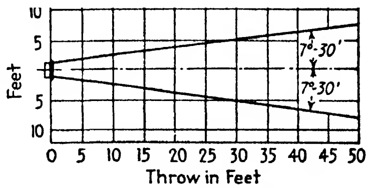


CHART 1. Type A.

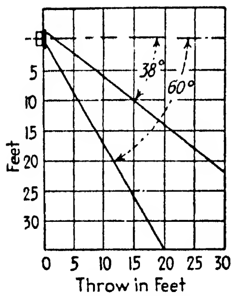


CHART 2A.—Type B.

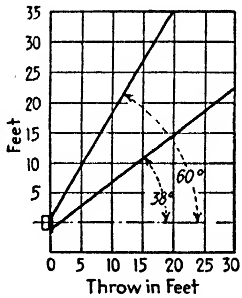


CHART 3A.—Type C.

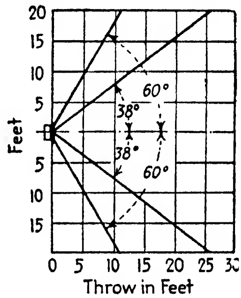


CHART 3.—Type D.

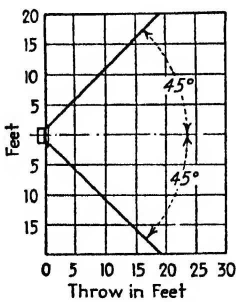


CHART 4.—Type E.

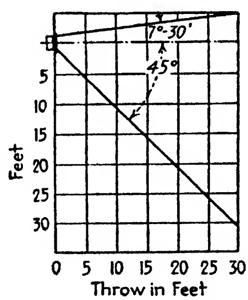


CHART 5.—Type F.

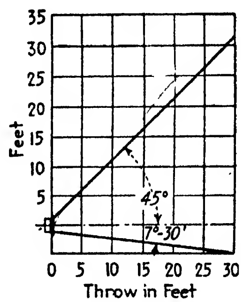


CHART 5A.—Type G.

CHARTS 1 to 5A DEFLECTIONS. (*Hart & Cooley Mfg. Co.*)

cooler room air, or else deflections should be used that will spread out the air stream sideways, thus accomplishing the same purpose. For best possible results a register such as No. 86 Design, with which the air may be directed toward the floor as well as sideways, should be used. If sideway deflections are used, precautions should be taken that the air is not directed against adjacent walls, furniture, curtains, etc. Charts 1–7 show the general pattern of the air stream with various types of deflections.

Air Velocity. 1. High Side-wall Locations. Charts 8–10 show the velocity required to project the air various distances with different types of deflection. From these charts select the velocity which will throw the required volume of air the required distance, *i.e.*, the distance from the register to the opposite wall. Velocities high enough to cause objectionable noise should not be used. Chart 10 shows the maximum velocities which can be used for satisfactory noise levels under the conditions shown. If the determined velocity exceeds the maximum shown in Chart 10, it is necessary, if objectionable noise is to be avoided, either to relocate the register or use more than one register so that the velocity can be reduced.

2. Baseboard Locations. Two opposing conditions govern the velocity to be used with baseboard registers. The velocity should be as high as possible in order to reduce the size of the registers required, but should be sufficiently low so that the air is not discharged far enough into the room to cause drafts. The velocity should never exceed 300 fpm with perforated or straight-flow, bar-type registers; 500 fpm with directional-flow registers, or 800 fpm with combination diffusing and downward deflecting registers, such as No. 86 Design.

Register Size. Knowing the volume of air required and having determined the proper velocity as outlined above, refer to Chart 6 for the proper size of register for various velocities and cfm requirements. This chart applies to either side-wall or baseboard registers.

Return-air Intakes. Exhaustive research to date has failed to disclose any effect that the location of return air intakes has on the temperature distribution in a room. Therefore, it is recommended that the intakes be installed in the most economical and convenient location unless it is considered important to adhere to established tradition, in which case the intakes should be installed in or near the outside walls.

Unless an unusually large volume of air is returned through any one intake, no drafts are caused by the air returned through intakes, regardless of the velocity. This is due to the fact that the air entering the intakes is drawn from so many directions that the velocity rapidly diminishes. With a velocity of 1,000 fpm at the face of the intake, for example, the velocity is only 90 fpm 12 inches away.

Size of Intakes. The size of intakes is governed by the volume and velocity of the air drawn through the intakes. As mentioned in the preceding paragraph, high velocities through intakes do not cause drafts but might result in excessive resistance. Velocities of 300–500 fpm through intakes may be used without appreciably increasing the resistance of the system as a whole. If it is desired to use higher velocities, the resulting resistance may be determined from Chart 11.

Special Applications and Precautions. 1. Do not use baseboard registers with air temperatures less than 135° in bathrooms, where the occupants are usually more susceptible to drafts, and 125° in other rooms. Air of lower temperatures coming in contact with the occupants of a room will create the impression of drafts. Low register temperatures will result from long runs of duct and from continuous fan operation.

2. Avoid the use of baseboard registers with cooling systems if possible, as it is extremely difficult to prevent drafts with such installations. If baseboard registers must be used with cooling systems, use a type of register with which the air can be deflected abruptly upwards.

3. On high sidewall installations there must be some provision either in the register or in the stackhead to prevent the air from striking the ceiling and causing discoloration. If the register is located within 12 in. of the ceiling, additional precautions must be taken to avoid streaked ceilings caused by the secondary air motion set up by the air leaving the registers. In such cases it is not sufficient to discharge the air on a horizontal plane; it must be discharged at a slightly downward angle.

4. Return air intakes located in the floor tend to collect dust and dirt and should not be used except where solid walls, such as concrete, brick, etc., make it impossible to install intakes in the sidewall or baseboard.

5. In rooms from which it is desired to shut off the heat during certain hours, the return air intakes should be provided with valves.

6. Bathrooms and kitchens sometimes become airbound when the doors are closed, with the result that the air flow is so retarded that the rooms do not properly heat. Vents, either to the attic or to the basement should be installed to overcome this condition. Attic vents should extend at least six inches above the attic floor and should be equipped with some type of hood or back draft damper so as to prevent backdrafts of cold air. Where the attic floor is well insulated and there is considerable moisture in the bathroom, the moisture is apt to condense in the cold attic space. Where vents to the basement are used, the vent pipe should extend to within a few inches of the floor to prevent backdrafts into the bathroom or kitchen. Basement vents should terminate in the furnace room so that the vented air will be used in the combustion of the fire, causing a suction and increased circulation in the vented rooms.

7. When garages are to be heated, the supply registers should be equipped with backdraft dampers to prevent cold air reversing into the house when the blower is not operating. Air introduced into garages must not be recirculated.

Air Distribution. The following information applies particularly to installations for cooling rather than for heating. Inasmuch as the grille requirements for cooling are much more severe than they are for heating installations, satisfactory results will be obtained if the information given is used for either heating or cooling installations.

To secure proper air distribution, it is necessary to comply with the following requirements:

1. The air should be projected so as to thoroughly blanket the room, and strike the opposite wall with a velocity not in excess of 40 fpm. (In unusually large spaces, this may not be possible. In such cases, satisfactory results may be obtained by projecting two or more streams of air toward each other so that they will meet with a velocity not in excess of 20 fpm.)

2. The air should not enter the occupied zone with a velocity in excess of 40 fpm. Higher velocities will cause drafts.

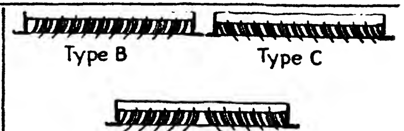
3. When two or more grilles are located on one wall, their air streams should not cross before the air has traveled at least two-thirds of its throw.

4. The largest possible amount of induced air movement should be obtained, so that the incoming air will be thoroughly mixed with the room air, resulting in an even temperature distribution. If the grilles are located at least a foot below the

Chart 6

For Determining Required Velocity and Free Area for Various Volumes and Throws

Type A



Type B Type C

Type D

Volume, cfm		Type A deflection, Throw, ft: 10	15	20	25	30	35	40	45	50	Type B, C, or D deflection, Throw, ft: 10	15	20	25	30
50	Velocity*	480	...	...	...	...	...	...	...	...	...	...	...	...	...
	Area†	15	...	...	...	...	...	...	...	...	...	...	...	...	...
75	Velocity	330	720	...	...	...	...	...	...	...	635	...	...	...	...
	Area	33	15	...	...	...	...	...	...	...	17	...	...	...	...
100	Velocity	245	555	960	...	...	...	...	...	...	480	...	...	...	...
	Area	59	26	15	...	...	...	...	...	...	30	...	...	...	...
150	Velocity	165	365	655	1,030	1,440	...	..	...	...	320	720	1,270	...	...
	Area	132	59	33	21	15	...	...	...	...	67	30	17	...	...
200	Velocity	120	280	480	755	1,110	1,510	1,920	...	...	245	540	960	1,530	...
	Area	235	103	60	38	26	19	15	...	...	118	53	30	19	...
250	Velocity	95	220	390	600	880	1,160	1,565	2,000	2,400	195	440	765	1,240	1,710
	Area	370	165	93	60	41	31	23	18	15	185	82	47	29	21
300	Velocity	...	180	320	500	735	980	1,310	1,660	2,050	165	365	645	1,005	1,440
	Area	...	238	135	86	59	44	33	26	21	265	118	67	43	30
350	Velocity	...	155	280	435	615	840	1,120	1,400	1,740	140	315	550	870	1,260
	Area	...	322	180	116	82	60	45	36	29	363	160	92	58	40
400	Velocity	...	135	245	385	560	740	980	1,225	1,510	125	275	490	740	1,090
	Area	...	420	237	150	105	78	59	47	38	470	210	118	78	53
450	Velocity	...	120	215	340	485	660	865	1,080	1,350	...	245	430	675	970
	Area	...	535	300	190	133	98	75	60	48	...	265	150	96	67
500	Velocity	...	...	195	305	435	600	780	975	1,200	...	220	390	615	880
	Area	...	...	370	235	165	120	92	74	60	...	325	185	117	82
550	Velocity	...	...	175	275	395	540	700	880	1,090	...	200	350	550	785
	Area	...	...	450	285	198	145	112	89	72	...	395	225	142	100

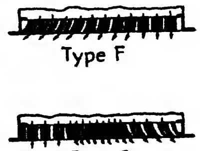
Source: Hart & Cooley Mfg. Co.

CHART 6. — (*Continued*)

		10	15	20	25	30	35	40	45	50	10	15	20	25	30
600	Velocity	...	...	160	255	370	495	650	825	1,020	...	185	325	515	730
	Area	...	...	535	340	235	175	133	105	85	...	470	265	168	118
650	Velocity	...	...	150	235	335	455	600	750	935	...	170	300	470	675
	Area	...	...	630	400	280	205	155	125	100	...	550	310	198	138
700	Velocity	...	...	135	220	315	430	560	695	870	...	160	280	440	630
	Area	...	...	730	465	320	235	180	145	116	...	640	360	230	160
750	Velocity	...	...	...	200	280	390	520	660	810	...	145	260	410	585
	Area	...	...	...	535	370	275	208	164	133	...	740	415	265	185
800	Velocity	...	...	...	190	275	370	490	615	765	...	...	245	385	550
	Area	...	...	...	610	420	310	235	187	150	...	...	470	300	210
850	Velocity	...	...	...	180	260	350	460	585	725	...	...	230	365	525
	Area	...	...	...	685	470	350	266	210	170	...	...	540	338	235
900	Velocity	...	...	...	170	245	335	435	555	675	...	...	215	340	490
	Area	...	...	...	770	530	390	300	235	192	...	...	600	380	265
950	Velocity	...	...	...	160	235	310	410	520	635	...	...	205	320	465
	Area	...	...	...	860	590	440	335	264	215	...	...	670	425	295
1,000	Velocity	...	...	...	150	220	295	390	495	615	...	...	195	305	435
	Area	...	...	...	950	660	485	370	290	235	...	...	740	470	330
1,100	Velocity	...	...	...	...	205	270	360	445	555	...	...	175	280	395
	Area	...	...	...	...	780	580	440	355	285	...	...	900	570	400
1,300	Velocity	...	...	...	...	170	230	300	380	470	...	...	150	235	340
	Area	...	...	...	...	1,100	820	620	495	400	...	...	1,250	800	555
1,500	Velocity	...	...	...	...	145	195	260	330	410	...	...	...	205	290
	Area	...	...	...	...	1,500	1,100	830	660	530	...	...	...	1,060	740
1,750	Velocity	...	...	...	...	...	170	225	280	345	...	...	...	175	250
	Area	...	...	...	...	...	1,480	1,130	900	730	...	...	...	1,450	1,010
2,000	Velocity	...	...	...	...	...	145	195	245	305	...	...	...	150	220
	Area	...	...	...	...	...	1,950	1,480	1,170	950	...	...	...	1,900	1,320
2,500	Velocity	...	...	...	...	...	...	155	195	245	...	...	...	...	175
	Area	...	...	...	...	...	...	2,300	1,830	1,480	...	...	...	...	2,070
3,000	Velocity	...	...	...	...	...	...	...	165	200	...	...	...	...	145
	Area	...	...	...	...	...	...	...	2,650	2,150	...	...	...	...	2,960

* Velocity in feet per minute.
† Area, grille free area, square inches.
SOURCE: By Permission of the Hart & Cooley Manufacturing Co.

CHART 6. — (*Continued*)



Volume, cfm		Type *E* deflection, Throw, ft					Type *F*, *G* deflection, Throw, ft					Type *S* deflection, Throw, ft					
		10	15	20	25	30	10	15	20	25	30	55	60	65	70	75	80
50	Velocity*	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	Area†	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
75	Velocity	...	...	...	...	...	720	...	...	...	...	...	...	...	...	...	...
	Area	...	...	...	...	...	15	...	...	...	...	...	...	...	...	...	...
100	Velocity	625	...	...	...	...	555	...	...	...	...	...	...	...	...	...	...
	Area	23	...	...	...	...	26	...	...	...	...	...	...	...	...	...	...
150	Velocity	415	1,140	...	...	...	370	980	...	...	...	...	...	...	...	...	...
	Area	52	19	...	...	...	58	222	...	...	...	...	...	...	...	...	...
200	Velocity	310	850	1,700	...	...	280	222	1,515	...	...	...	...	...	...	...	...
	Area	52	19	...	...	...	58	720	19	...	...	...	...	...	...	...	...
250	Velocity	250	680	1,385	...	...	225	580	1,200	2,550	...	...	...	...	...	...	...
	Area	144	53	26	...	...	160	62	30	14	...	...	...	...	...	...	...
300	Velocity	210	555	1,165	2,400	...	190	480	1,005	2,160	...	...	...	...	...	...	...
	Area	208	78	37	18	...	230	90	43	20	...	...	...	...	...	...	...
350	Velocity	180	485	990	2,020	...	160	445	885	1,870	...	2,800	...	...	...	...	...
	Area	282	104	51	25	...	315	114	59	27	...	18	...	...	...	...	...
400	Velocity	155	425	860	1,800	...	140	385	740	1,600	3,030	2,500	3,030	...	...	...	...
	Area	370	136	67	32	...	412	149	78	36	19	23	19	...	...	...	...
450	Velocity	140	375	775	1,580	3,080	125	345	670	1,410	2,700	2,240	2,700	3,250	...	...	...
	Area	468	172	84	41	21	520	188	97	46	24	29	24	20	...	...	...
500	Velocity	...	340	700	1,400	2,770	...	310	600	1,285	2,400	2,000	2,400	2,880	3,330	...	...
	Area	...	213	103	51	26	...	232	120	56	30	36	30	25	21	...	...
550	Velocity	...	305	625	1,260	2,530	...	280	540	1,150	2,180	1,780	2,180	2,600	3,010	...	...
	Area	...	258	125	62	31	...	280	145	63	36	44	36	30	26	...	...

CHART 6. — (*Continued*)

		10	15	20	25	30	10	15	20	25	30	55	60	65	70	75	80
600	Velocity	...	280	575	1,185	2,340	...	260	500	1,055	2,110	1,660	2,010	2,400	2,790	3,320	...
	Area	...	308	150	73	37	...	333	173	82	43	52	43	36	31	26	...
650	Velocity	...	260	535	1,085	2,120	...	240	465	985	1,830	1,530	1,830	2,220	2,600	3,020	3,460
	Area	...	360	175	86	44	...	392	202	95	51	61	51	42	36	31	27
700	Velocity	...	240	500	1,020	1,980	...	220	430	925	1,715	1,425	1,715	2,060	2,400	2,800	3,260
	Area	...	420	202	99	51	...	455	235	109	59	71	59	49	42	36	31
750	Velocity	...	225	470	950	1,860	...	205	400	865	1590	1,320	1,590	1,930	2,250	2,630	3,090
	Area	...	480	230	114	58	...	525	270	125	68	82	68	56	48	41	35
800	Velocity	...	210	435	885	1,740	...	195	375	805	1,495	1,240	1,495	1,800	2,130	2,450	2,880
	Area	...	545	265	130	66	...	595	308	143	77	93	77	64	54	47	40
850	Velocity	...	200	410	840	1,640	...	180	335	760	1,415	1,180	1,415	1,710	1,980	2,320	2,670
	Area	...	620	300	146	75	...	675	345	162	87	104	87	72	62	53	46
900	Velocity	...	190	390	780	1,550	...	170	335	715	1,340	1,120	1,340	1,610	1,880	2,200	2,550
	Area	...	690	335	165	84	...	755	390	182	97	116	97	81	69	59	51
950	Velocity	...	180	370	750	1,460	...	165	315	675	1,270	1,050	1,270	1,525	1,800	2,075	2,400
	Area	...	770	372	183	94	...	840	432	203	108	130	108	90	76	66	57
1,000	Velocity	...	170	350	715	1,400	...	145	300	640	1,210	1,000	1,210	1,450	1,700	1,975	2,290
	Area	...	850	415	202	103	...	990	480	225	119	144	119	99	85	73	63
1,100	Velocity	...	115	315	645	1,265	...	...	270	585	1,090	900	1,090	1,320	1,550	1,610	2,080
	Area	...	1,040	500	245	125	...	...	580	270	145	175	145	120	102	98	76
1,300	Velocity	...	...	265	545	1,070	...	...	230	490	935	770	935	1,125	1,320	1,520	1,760
	Area	...	...	700	342	175	...	...	810	380	200	244	200	166	142	123	106
1,500	Velocity	...	...	230	475	940	...	...	200	430	830	665	830	980	1,140	1,310	1,540
	Area	...	...	940	455	230	...	...	1,080	505	260	325	260	220	190	165	140
1,750	Velocity	...	...	200	405	800	...	...	170	370	690	575	690	840	975	1,130	1,320
	Area	...	...	1,270	620	315	...	...	1,470	685	365	440	365	300	258	223	192
2,000	Velocity	...	...	175	335	695	...	...	150	320	600	495	600	730	855	990	1,150
	Area	...	...	1,660	810	415	...	...	1,930	900	480	580	480	395	337	290	250
2,500	Velocity	...	...	140	285	580	...	...	...	255	480	400	480	580	680	780	925
	Area	...	...	2,600	1,260	620	...	...	...	1,400	750	900	750	620	530	460	390
3,000	Velocity	...	...	...	235	465	...	...	...	215	405	335	405	485	570	660	770
	Area	...	...	...	1,830	930	...	...	...	2,030	1,075	1,300	1,075	890	760	660	565

* Velocity in feet per minute.
† Area, grille free area in square inches.

Chart 7

To select a grille having a certain free area (as determined from Chart 6), decide on the desired height of the grille. The correct width of the grille will be found in the column under the height and opposite the required free area.

Free area, sq in.	Grille height, in.												
	4	5	6	8	9	10	12	14	16	20	24	30	36
15	8	..	..	..	..	..	..	..	..	..	..	..	..
25	10	8	..	..	..	..	..	..	..	..	..	..	..
35	12	10	8	..	..	..	..	..	..	..	..	..	..
45	16	14	10	..	..	..	..	..	..	..	..	..	..
55	20	16	12	10	..	..	..	..	..	..	..	..	..
65	24	18	14	..	10	..	..	..	..	..	..	..	..
75	26	22	18	12	..	10	..	..	..	..	..	..	..
85	30	24	20	14	12	..	..	..	..	..	..	..	..
100	36	30	24	16	14	12	..	..	..	..	..	..	..
125	..	36	30	20	18	16	14	..	..	..	..	..	..
150	..	40	32	24	22	18	16	14	..	..	..	..	..
175	..	..	40	26	24	22	18	16	..	..	..	..	..
200	..	..	..	32	26	24	20	18	16	..	..	..	..
225	..	..	..	36	32	..	24	20	18	..	..	..	..
250	..	..	..	40	36	30	26	22	20	..	..	..	..
275	..	..	..	44	40	32	..	24	22	..	..	..	..
300	..	..	..	..	..	36	30	26	..	..	..	..	..
325	..	..	..	..	44	40	32	..	24	20	..	..	..
350	..	..	..	..	48	44	..	30	26	..	..	..	..
375	..	..	..	..	..	..	36	32	..	22	..	..	..
400	..	..	..	..	..	48	40	..	30	24	..	..	..
425	..	..	..	..	..	..	..	36	32	..	..	..	..
450	..	..	..	..	..	..	44	..	..	26	..	..	..
475	..	..	..	..	..	..	48	40	..	..	..	..	..
500	..	..	..	..	..	..	..	..	36	..	24	..	..
525		..	..	..	..	..	..	48	..	30	26	..	..
550		..	..	..	..	..	54	..	40	32	..	..	..
600			..	..	..	..	..	..	44	..	..	..	..
625			..	..	..	..	..	..	..	36	30	..	..
650			..	..	..	..	..	54	48	..	..	..	..
675			..	..	..	..	..	..	..	..	32	..	..
700	..		..	..	..	..	..	..	..	40	..	..	..
750		..	..	..	..	..	..	..	54	44	36	..	..
775		..	..	..	..	..	..	..	..	..	..	30	..
825	..	..	..	..	..	..	..	..	..	..	40	32	..
850	..	..	..	..	..	..	..	..	..	48	..	..	..
925	..	..	..	..	..	..	..	..	..	..	44	..	..
950	..	..	..	..	..	..	..	..	..	54	..	36	..
1,000	..	..	..	..	..	..	..	..	..	..	48	..	..
1,100	..	..	..	..	..	..	..	..	..	60	54	40	36
1,200	..	..	..	..	..	..	..	..	..	..	..	44	..
1,300	..	..	..	..	..	..	..	..	..	..	60	48	40
1,400	..	..	..	..	..	..	..	..	..	..	..	..	44
1,500	..	..	..	..	..	..	..	..	..	..	..	54	48
1,600	..	..	..	..	..	..	..	..	..	..	..	60	..
1,700	..	..	..	..	..	..	..	..	..	..	..	..	54
1,900	..	..	..	..	..	..	..	..	..	..	..	..	60

Source: Hart & Cooley Mfg. Co.

Chart 8.* — Air-stream Drop

Showing the Drop, in Feet, of the Air Stream with a Grille Having a Throw as Shown at the Velocity Indicated

Throw, ft	Velocity, fpm													
	200	300	400	500	600	700	800	900	1,000	1,100	1,200	1,300	1,400	1,500
10	6	3.8	2.9	2.2	..	..	..	..	..	..	..	..	..	..
15	8.5	5.6	4.2	3.4	2.7	2.4	2.1	..	..	..	..	..	..	..
20	16.0	10.5	8.0	6.2	5.2	4.5	3.9	3.5	3.1	2.8	2.6	2.4	2.2	..
25	..	14.6	11.0	8.9	7.3	6.2	5.5	4.9	4.4	4.0	3.6	3.4	3.2	2.9
30	..	..	14.5	11.7	9.5	8.2	7.2	6.4	5.8	5.3	4.8	4.4	4.1	3.8
35	..	..	..	14.2	12.0	10.2	9.0	8.0	7.2	6.5	6.0	5.5	5.2	4.9
40	..	..	..	..	14.3	12.5	11.0	9.6	8.6	8.0	7.3	6.6	6.2	5.8
45	..	..	..	..	..	14.2	12.5	11.1	10.1	9.1	8.4	7.7	7.1	6.7
50	..	..	..	..	..	..	14.5	12.9	11.7	10.6	9.7	9.0	8.3	7.8

* Drop as shown is based on 15° temperature differential.
For 10° temperature differential, deduct 33 per cent from drop shown
For 20° temperature differential, add 33 per cent to drop shown.

Source: Hart & Cooley Mfg. Co.

ceiling and if the three preceding requirements are met, a sufficient percentage of induced air movement will automatically result.

Selection of Grille. The proper location, deflection, velocity, free area, and size of grille are determined as follows:

1. *Location of Grille.* Ordinarily the structural details of the building and economical considerations concerning the layout of the duct work are principal factors in determining the location of the grilles—that is, the grilles must be located where conditions permit and not in what might be a more desirable location if there were no restrictions.

2. *Type of Deflection.* Refer to Charts 1–5, which show the air distribution, on a horizontal plane, with various types of deflections. From these charts determine the deflection that will most nearly blanket the area to be conditioned Our Charts Nos. 1-A through 5, pages 502 and 503.

3. *Velocity and Free Area.* Knowing the volume of air required from each grille and the required length of throw, and having determined from the above the correct type of deflection, the proper face velocity and the required free area of the grille can be determined from Chart 6.

4. *Grille Size.* Having determined the required free area from Chart 6, the correct grille size can readily be determined from Chart 7.

5. *Minimum Grille Level.* Chart 8 shows the drop in the air stream under various conditions. From this chart it is a simple matter to determine the minimum height above the floor at which the grille should be located to prevent the air entering the occupied zone with a velocity in excess of 40 fpm. It is sometimes necessary to use higher velocities, relocate the grilles, or use a different type of grille to prevent drafts.

Chart 9. — Resistance

Showing Resistance (Static Pressure, Inches Water Gauge) of Hart & Cooley Grilles at Various Velocities

Velocity, fpm	Design 71	Design 72	Design 76	Designs 77 and 78	Designs 84 and 85		Design 90	
					Deflection *A*	Deflection *E*	Deflection *A*	Deflection *E*
50	0.002	0.001	0.004	0.001	0.001	0.002	0.001	0.001
100	0.003	0.002	0.005	0.002	0.003	0.004	0.003	0.003
150	0.006	0.003	0.007	0.003	0.005	0.006	0.005	0.005
200	0.009	0.005	0.011	0.005	0.007	0.008	0.007	0.007
250	0.014	0.007	0.016	0.007	0.009	0.011	0.010	0.010
300	0.019	0.010	0.021	0.010	0.011	0.014	0.013	0.014
350	0.024	0.013	0.027	0.013	0.015	0.018	0.016	0.018
400	0.030	0.017	0.033	0.017	0.020	0.022	0.020	0.023
450	0.036	0.022	0.040	0.022	0.025	0.028	0.025	0.028
500	0.045	0.026	0.050	0.027	0.030	0.034	0.030	0.033
550	0.052	0.031	0.060	0.032	0.035	0.040	0.036	0.040
600	0.062	0.037	0.070	0.038	0.040	0.047	0.042	0.047
650	0.075	0.043	0.082	0.045	0.045	0.054	0.049	0.054
700	0.085	0.050	0.095	0.052	0.052	0.062	0.055	0.061
750	0.100	0.058	0.109	0.060	0.059	0.070	0.062	0.068
800	0.110	0.066	0.124	0.068	0.067	0.079	0.070	0.076
850	0.125	0.072	0.139	0.076	0.074	0.088	0.079	0.084
900	0.140	0.080	0.155	0.084	0.082	0.098	0.087	0.093
950	0.155	0.088	0.172	0.092	0.090	0.108	0.095	0.102
1,000	0.170	0.097	0.190	0.100	0.100	0.118	0.102	0.110
1,100	0.205	0.112	0.228	0.118	0.114	0.138	0.120	0.132
1,200	0.240	0.130	0.268	0.137	0.130	0.1[illegible]0	0.139	0.151
1,300	0.275	0.150	0.310	0.157	0.150	0.182	0.160	0.172
1,400	0.315	0.170	0.350	0.178	0.170	0.206	0.179	0.192
1,500	0.350	0.185	0.385	0.198	0.190	0.230	0.200	0.214
1,600	0.400	0.208	0.450	0.220	0.210	0.260	0.218	0.235

NOISE-LEVEL CHART 10

Showing Maximum Allowable Velocities to Avoid Excessive Noise Levels

Type of room	Maximum velocity, fpm						
	Register, grille, or intake design						
	No. 69	Nos. 72, 77, 78, 84, 85, and 86			No. 90		
		Deflection			Deflection		
		A	*B, C, D*	*E, F, G*	*A*	*B, C, D*	*E, F, G*
Bed rooms..........	1,300	750	900	750	1,000	1,150	800
Other rooms........	1,800	1,100	1,250	1,050	1,350	1,600	1,250

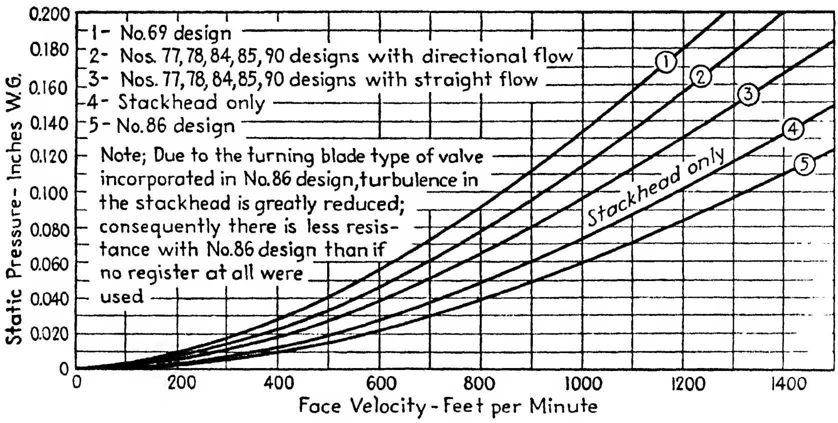
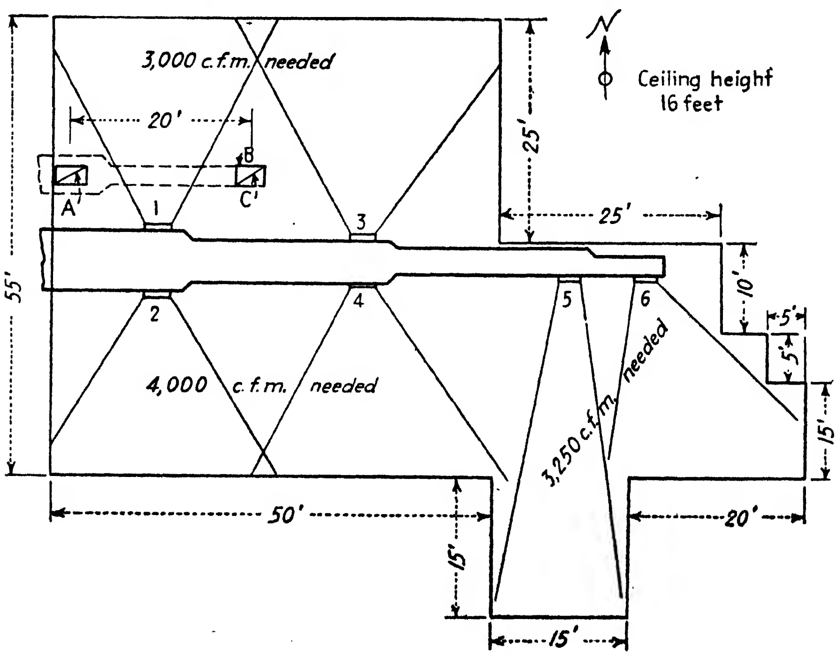


CHART 11.—Resistance chart, showing resistance (static pressure, inches water guage) of registers, grilles, and intakes at various velocities. (*Hart and Cooley Mfg. Co.*)

After studying the following simple grille-selection problem you are likely to have an idea as to a different method by which conditioned air could be distributed throughout the space illustrated. In our solution, where interpolations have been made in the various charts and tables, they have been made simply "by eye." The small differences in the various tables of data make accurate interpolation more trouble than it is worth.



Problem. Suppose that you have a conditioned space as illustrated by the sketch. Heat loads have been calculated and air quantities determined as indicated on the figure.

Solution. A study of the odd-shaped conditioned space, keeping in mind the plan-view grille deflections shown on page 494, indicates that a supply duct as shown in Fig. 11 will do. This duct is straight, simple, and reasonably short; it will not look badly and will be economical to build. Six grilles are required. Grilles 1, 2, 3, and 4 must be deflection *E*, grille 5 deflection *A*, and grille 6 deflection *G*. The approximate length of throw can be determined by measuring the throw length *from the center line* of the supply duct for simplicity. This gives a throw a foot or two longer than the actual throw. We now have information that we can tabulate in columns (1), (2), and (3) of the Table 1.

TABLE 1.—SUPPLY AIR GRILLES

Grille	(1) Cfm	(2) Deflec-tion	(3) Approx throw, ft	(4) Free area, sq in.	(5) Face velocity	(6) Approx air-stream drop, ft
1	1,500	*E*	25	342	545	8
2	2,000	*E*	30	415	695	8.2
3	1,500	*E*	25	342	545	8
4	2,000	*E*	30	415	695	8.2
5	1,500	*A*	45	330	660	14.1*
6	1,750	*G*	30	365	690	8.2

From Chart 6, we determine for each grille the required *velocity and free area*, tabulating these items in columns (4) and (5) of Table 1. We then turn to Chart 8 and determine the approximate *air-stream drop*. This is tabulated in column (6) of Table 1; note that grille 5 indicates a 14-ft air-stream drop (marked * in Table 1). This means that this grille would have to be either an 84 design or a 90 design. In either case, the grille center line would have to be 2 to 3 ft *below* the ceiling, the grille being arranged to *deflect the air upward* instead of horizontally. This vertical deflection would have to be ad-

TABLE 2.

Grille	Actual grille size	Grille resistance
1 and 3	14 × 30, or 16 × 26	0.04
2 and 4	16 × 30, or 16 × 32	0.062
5	12 × 32, or 16 × 24	0.045
6	12 × 36, or 14 × 32	0.062

justed experimentally after completing the installation to cut down the air-stream drop from about 14 ft to about 9 ft. This will ensure proper but not excessive air motion at the breathing level within the small 15-ft square alcove, near the end of the throw from grille 5.

We would now turn to Chart 7. Knowing the free area required, as shown in column (4) of Table 1, we would pick the nearest standard grilles from Chart 7. Finally from

Chart 9, based on the assumption of an 84 design grille having been selected throughout, we would determine grille resistance or grille friction. This information is shown in Table 2.

Return air can be taken through three grilles A, B, and C, as indicated in the drawing. The return-air duct could be run along the ceiling, dropping risers down along building columns to about the breathing level. Or, if the owner of this space controlled the floor below, the return-air duct could be run below the floor with short stubs coming up alongside building columns. Using this design for return-air grilles, our air quantities and grille sizes might be as follows:

Grille A	2,000 cfm
Grille B	2,500 cfm
Grille C	2,500 cfm
Total	7,000 cfm

Return grille A—3 sq ft area, 75% free area
Return grilles B and C—4 sq ft gross area, 75% free area

CHAPTER XXVI

AUTOMATIC CONTROL DEVICES AND SIMPLE SYSTEMS

General. Thermostats and humidistats in one form or other along with pressure-controlling devices form the basis of almost all control systems. In principle, all thermostats are basically the same—they are devices sensitive to temperature changes. In the same way, all humidistats are basically the same, although details of their construction and operation may be entirely different—they are devices sensitive to changes in relative humidity.

In the first section of this chapter it is our purpose to explain certain control instruments and devices. As a test of your knowledge, you should be able to give a complete explanation based upon the following outline:

1. Basic instruments
   - *a* Thermostats or temperature controllers
   - *b*. Humidistats or humidity controllers
   - *c*. Pressure controllers

How does each of the above work and what does it do?

2. Three different types of controls
   - *a*. Electric
   - *b*. Pneumatic (compressed air)
   - *c*. Self-contained (direct acting)

Compare briefly the three types listed above, including the basic advantages and disadvantages of each.

3. Positioning of control instruments
   - *a*. On-and-off or two-position type
   - *b*. Modulating (gradual acting or proportioning)

You must become familiar with the basic difference between these two possible control arrangements. (The distinction is simple. You need not be concerned with exactly how electric modulating controls operate.)

Automatic Controls. *Limit Controls.* Limit controls include a variety of controlling devices and may be practically any form of thermostat or pressure control. This term is applied to any controlling device, the primary function of which is to limit or prevent the operation of the system in such a manner that temperatures or pressures exceed predetermined limits. When the predetermined limit is a maximum temperature or pressure, such control is designated as a high-limit control. Where function of the control is to prevent temperature or pressures falling below a predetermined minimum, the control is designated as a low-limit control.

Humidity Controls. Humidity controls are defined as automatic devices reacting to change in humidity. Normally, such devices react to changes in relative humidity. Within this group the devices commonly called "humidity controls" operate to prevent relative humidity from exceeding a predetermined maximum. As such, they are a form of limit control, although when operating in control of humidity supplying equipment, they are regulating devices.

Furnace Fan Controls. In the operation of forced warm-air heating systems, it is usually necessary that some device prevent circulation of air until a sufficiently high temperature is attained in the furnace. Such a device, essentially a limit control, is defined as a furnace fan control or furnacestat.

Relays. A relay is defined as a unit installed between a controlling device and the controlled device for purposes of amplifying the capacity of the controller to the extent required by the load of the controlled device. For example, a thermostat in order to preserve its sensitivity may be constructed in such manner that it is not capable of handling the power required of a damper motor. It is, therefore, the practice to install a relay between the two and allow the thermostat to actuate the relay and the relay, in turn, to actuate the damper motor.

Principles of Control System. Control systems may be broadly classified into three general types as regards the characteristics of the motion imparted by the controls to the controlled equipment. These three classifications will be described in the following paragraphs. It must be remembered that often heating, ventilating, or air-conditioning systems under automatic control may make use of more than one and sometimes all of the three types of control in various phases or functions of the system. Following this broad discussion will be found pages outlining the actual basic circuits used in obtaining these actions.

Two-position Control. Two-position control is also referred to as "on-and-off" control or as positive-acting control. The first designation, however, is more descriptive of the principles underlying this form of control, since the result of the controlling action is always one of two possible positions. As an example, a simple thermostat which starts and stops an oil burner or a ventilating fan, or which opens and closes a solenoid valve, can merely select between starting and stopping of the fan or between opening and closing of the valve. There are no intermediate positions or degrees of motion between the two extremes of operation. Where the operation is primarily one of stopping and starting a motor-driven appliance, the term "off and on" is equally descriptive, but in the case of a louver damper, a two-position damper motor would require that the damper be moved from full closed to full open, or vice versa. Similarly, a two-position control valve would move between two fixed limits such as full closed and full open.

Two-position control is, in general, the simplest method of providing regulation. However, it has definite shortcomings in certain circumstances. For example, the operation of a louver-type face and by-pass damper in connection with a heating coil under two-position control would pass all the air through the heater or all the air around the heater. Thus the air discharged from such a system would vary in temperature sharply and would, unquestionably, be undesirable from a result standpoint. Such a system would be better handled by other forms of control to be later described. However, in a preheating coil, when steam is admitted, normal practice is to open the steam valve completely, and when tempering is not required, the steam valve is completely closed. This is a complete two-position control cycle and is adequate to the requirement.

Modulating Control. The modulating control system is also designated as "gradual or graduated acting control" or "proportioning control." These names are synonymous as applied to automatic control and are used to designate the type of system in which a control valve or damper motor modulates or proportions the flow of air, steam, or water in reacting to change of conditions at the controller. Modulating control causes motion in the controlled device in proportion to motion caused in the controller by fractional-degree variations in the medium to which the controller is responsive. After such fractional change has been measured at the controller and translated to terms of a new position of the valve or damper, the modulating system stands by awaiting further change at the controller before additional motion occurs. The extent of the motion is limited only by the limits of the controller and by the intensity of the change of conditions as measured; hence the valve or damper is repositioned as frequently as changes at the controller occur, but always in direct proportion to the amount of change.

The example of the heating coil with its face and by-pass damper, damper motor and thermostat will serve to illustrate this principle. Assuming the thermostat to be set to the range of 70 to 72°, at 72° the face damper will be completely closed, but if the temperature drops fractionally, for instance, 1/10 deg to 71.9°, the modulating damper motor will cause the face damper to open 1/20 or 5 per cent of its total movement. If the drop in temperature had been ½ deg, *i.e.*, to 71.5°, the damper would have opened 25 per cent of its total possible travel. After each such change, the damper will stand still awaiting the next movement at the thermostat.

Types of Controls. Another means of classifying control systems will be found in considering them in relation to the primary operating mediums, and such classification indicates three broad groups: *viz.*, electric control systems, pneumatic control systems, and self-contained control systems. A brief description of these three types of systems follows.

Electric Control Systems. In such control systems the primary medium utilized to provide for the operation is electricity, and the basic function of these controls consists of switching, or otherwise adjusting electric circuits to govern electric motors, relays, or solenoids. The individual units of this type of system are interconnected by line voltage or low-voltage wiring, and this wiring serves to complete the circuits carrying the commands of the controllers to the controlled valves or damper motors.

Pneumatic Control Systems. In the pneumatic control systems, the primary source of operation is obtained through a medium of compressed air, the pressure of which is varied by the controlling devices. In these systems one or more centrally located air compressors furnish a supply of compressed air which is distributed in special piping to the various controlling and controlled devices. By means of leak ports or orifices, the pressure of the air is varied in the branch lines and the changing pressures are utilized in air-operated damper motors or valves to obtain the movement necessary to the operation of valves and dampers.

Control Devices and Their Operation. *Why Not Use All Direct-acting Controls?* Direct-acting or self-contained controls are limited in their application. The power bellows of a thermostatic expansion valve (see Chap. XI) is an ideal example of a self-contained control. It is limited in two respects. (1) The bulb can be only a reasonable distance from the power bellows, say 6 to 10 ft. (2) The bellows can exert only a limited force, which is great enough to operate an expansion valve, but not great enough to operate many larger and heavier devices. For these reasons, of which the first is most important, we frequently must use electric or pneumatic controls. In the case of humidistats, the element being usually human hairs, animal gut, paper, or a small block of wood, so little power is available that direct-acting controls are almost impossible.

With either electric or pneumatic controls the slight power available from the thermostat element or humidity-sensitivity element is used to operate a small light electric switch or compressed-air needle valve. You can readily understand how the small electric switch can operate electric motors of any size. If the motor is very large, the control instrument would energize the magnetic coil of a relay, the relay contacts starting the electric motor.

Operation of Pneumatic Controls. How does the compressed-air control operate the large valve, damper, or other device requiring considerable power? A brief description of the operating principles of pneumatic controls is given on page 520. Refer to this in connection with the following text and with Fig. 1. You have seen the automobile hoist used in service stations. The attendant merely turns a little valve in a half-inch compressed-air line; the compressed air then enters a cylinder and lifts the large round shaft which carries the rack and automobile about 6 ft up in the air. In a compressed-air control, the needle valve operated by the humidistat or thermostatic element is like the small valve operated by the gasoline-station attendant. Opening this needle valve requires very little power. The compressed air is admitted to a large bellows or piston by means of which it can exert a large force. The bellows or piston connects to the stem of a valve to the louver or damper or other automatically controlled mechanism.

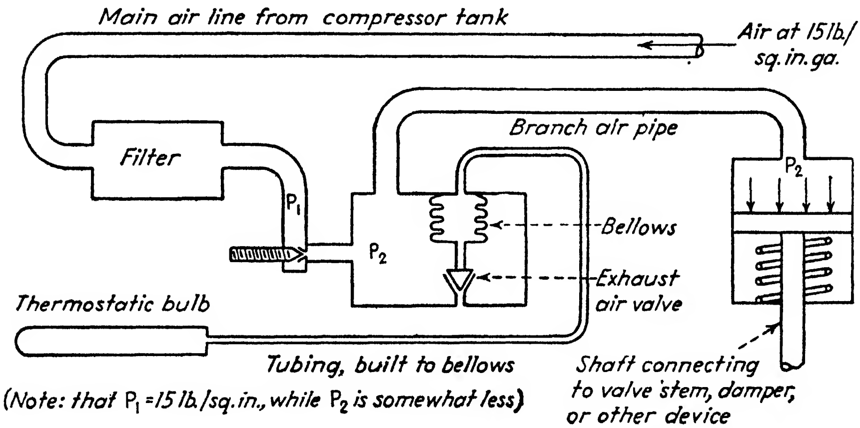


FIG. 1.—Pneumatic control mechanism sketch.

Actually, compressed-air controls of the on-and-off type may act exactly as just described. For modulating or proportioning pneumatic controls, the arrangement is that shown in Fig. 1. Compressed air is supplied from a line ordinarily at a pressure of 15 psi. This air is supplied to a small chamber through a needle valve, which may be of a fixed or variable setting. In this chamber an exhaust valve port is provided, out of which air may leak at a variable rate, under the control of an exhaust valve. This exhaust valve is operated by a small bellows in the case of a thermostat, or by the humidity-sensitive element in the case of a humidistat. (Figure 1 illustrates a thermostat type.) From the control chamber a pipe line connects with the bellows or piston which is to operate the valve or damper. You can readily see that, if the thermostat or humidistat causes the exhaust needle valve to close tightly, the pressure in the control chamber will become the same as the pressure of the compressed air supply, *viz.*, 15 psig.

This full pressure will be applied in turn to the large piston or bellows. As the exhaust valve opens, permitting air to *leak out* of the chamber, the pressure in the chamber will drop. How far this pressure drops will depend on how far the exhaust valve opens. This reduced pressure will be applied to the large piston or bellows making it possible to position the valve or damper to any midway position desired. The control is so arranged that when the exhaust air valve is *wide open* the large piston or bellows takes an extreme position, *opposite to* the extreme position taken when the exhaust air valve is *tightly closed.*

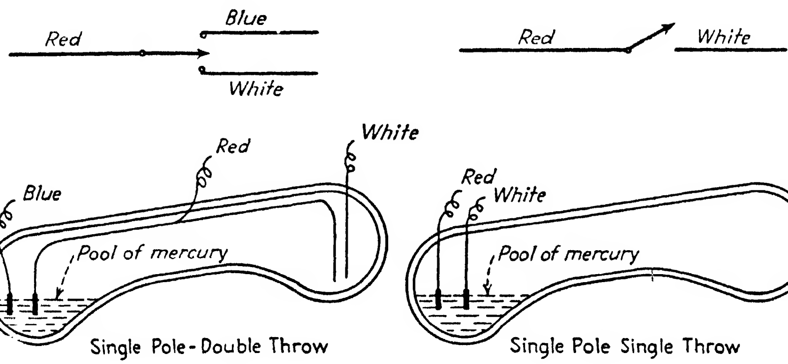


FIG. 2.—Typical mercury switches.

Mercury Switches. The metal contacts of a mercury switch are sealed into a closed tube. The center of the closed tube has a convex place or "bump" on the bottom. The pool of mercury slides over this raised portion of the glass in a single globule or pool. Since electricity flows readily through mercury, the circuit is closed when two of the metal contacts are immersed in a pool of mercury. Because of the raised portion in the glass, the pool of mercury does not tend to run back to the other end of the tube because of slight vibrations. Instead the tube must definitely tip back in the other direction. See Fig. 2.

Controls Classified As to Function. Later in this chapter you will encounter control instruments having certain special names based upon the purpose of the control. (For example, stoker switch, stack switch, aquastat, furnacestat, windowstat.) These special names do not mean that the controls are different in principle from those we have been discussing. For instance, an aquastat controls the temperature of water. ("Aqua" is the Latin word meaning water.) Nevertheless an aquastat is simply a thermostat because it is sensitive to temperature. Similarly, you will find that all controls having special names are really some form of thermostat, humidistat or pressure control.

Step Controllers. In some cases it is desirable to use modulating control where a modulating thermostat will start several compressors according to the need for cooling. In other cases it may be desired to operate several pumps, solenoids, or other devices *one after the other* in a *fixed sequence.* This is done by means of a step controller. Remember that in the case of electric modulating control, the thermostat might be set to modulate between any two temperatures such as 78 to 80°. With such a thermostat the modulating motor which it operates would be at one extreme position if the temperature at the thermostat were 80° and at the other extreme position if the temperature at the thermostat were 78°. If the temperature at the thermostat were exactly 79°, the modulating motor would be at exactly the middle position. As we have seen, such modulating motors ordinarily would operate dampers or valves.

In a step controller we simply have a *long extended shaft* on our modulating motor. Lugs or ears (cams) on this shaft operate mercury switches. The Minneapolis Honeywell Regulator Company makes step controllers as standard items having 5 or 10 such switches operated from the modulating motor shaft. At one extreme position of the controller, all the switches will be off; at the other extreme position, all the switches will be on. At intermediate positions of the "Modutrol" motor, one, two, three, or four of the switches could be on. If you wish a step controller having four switches, you simply buy a five-step controller and leave one switch unused.

Let us consider a summer cooling installation of 100 tons capacity with refrigeration supplied by four 25-ton compressors. Suppose that we wish to control these by a modulating thermostat with a range of 78 to 80°. At 78° dry bulb, in the conditioned space (at the thermostat) all the compressors would be off. If the temperature rose to 78.5°, the modulating motor would turn one-fourth of the way and the first switch on the shaft would start compressor 1. If this did not provide sufficient refrigeration and the temperature continued to rise, the second compressor would start when the temperature reached 79°. At 79.5° the third compressor would be started, and at 80° or above all four compressors would run. This action would be reversed if the temperature were to drop, compressors cutting out for each 0.05° of temperature drop. Similarly, step controllers can be used to operate a series of valves, solenoids, or motor starters for any purpose.

Differential. You will frequently encounter the word "differential" in connection with automatic controls. It has two different meanings, depending upon whether the control is the modulating or on-and-off type. In the case of a modulating control, the differential means the over-all range from one extreme position to the other. Modulating thermostats may be purchased having a differential of 1, 2, or 5° as required.

In the case of on-and-off controls, differential has a somewhat different meaning. Let us consider a thermostat that is to be set at 70°F. Obviously, this thermostat could not control the temperature to *exactly* 70.0°F. The thermostat operates only because of small *changes* in temperature. We could have a very exact thermostat that would turn on heat if the temperature drops to 69.9°, shutting off the heat if the temperature rises to 70.1°F. This thermostat would be said to have a differential of 0.2°F.

On most pressure controls the differential is adjustable, while on most temperature controls and humidity controls differential is not adjustable. Suppose that you had a pressure control on an oil-fired steam boiler, designed to prevent the steam pressure from exceeding 5 psi. This control would be arranged to shut off the oil burner or other fuel-burning device when the boiler pressure reached 5 psi. If set for a *differential* of ½ psi, this control would permit the oil burner to restart when the boiler pressure dropped to 4.5 psi. By setting the control differential at 2 psi, the boiler would not restart until the pressure dropped to 3 psi. The main control setting, which for the control we have been discussing was 5 lb, is ordinarily independent of the differential adjustment. In other words, we can leave the differential at 2 lb and move the main setting to 8 lb. The control would then cut off the oil burner when the boiler pressure reached 8 lb and would permit it to restart if the pressure dropped to 6 psi.

Control Systems. The following outline gives simple control systems which are standard practice in connection with domestic heating installations and the standard controls in connection with air conditioning refrigeration.

1. Burner or fire controls
   - *a.* Stoker
   - *b.* Gas
   - *c.* Oil
2. Control arrangements for particular systems
   - *a.* Steam
   - *b.* Hot water
   - *c.* Warm air (including humidifying)
   - *d.* Domestic hot water
3. Basic refrigeration controls
   - *a.* Expansion valves (this material is given in Chap. XI)
   - *b.* High- and low-pressure controls
   - *c.* Liquid line solenoids
   - *d.* "Unloading" devices, for multiple-cylinder compressors

Domestic-heating Controls. A single wall-mounted room thermostat is ordinarily used to control temperature in a domestic heating installation.

This thermostat acts in a manner to call for heat from the basement when heat is required. Just how it acts and what it does depends upon whether the system is warm air, hot water, or steam and whether the fuel-burning equipment is oil, gas, or stoker fired. Let us first consider the means of burner control in the case of stokers, oil burners, and gas-fired heating installations. Certain safety devices are required on each of these, regardless of the type of heating system.

The Stoker Switch. In the case of a stoker, a small fire must be maintained at all times even though in mild weather the thermostat may not call for any heat for several days. This fire must be so low that no heat is supplied to the heated space, but the small fire must be held; otherwise it would be necessary to light a new fire when heat was again needed.

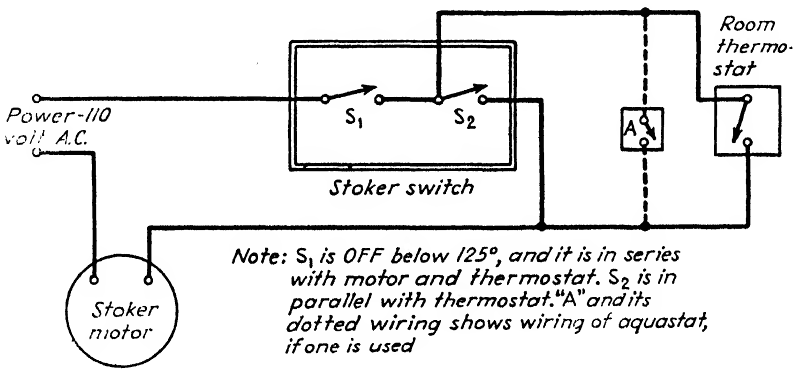


Fig. 3.—Diagram showing the wiring of a stoker switch.

Furthermore, it would be possible for the fire to go out by means of some accident even during cold weather. If the thermostat then called for heat, the stoker would run continuously, filling the entire furnace with unburned coal. This has occurred on installations not equipped with proper controls. In this event, the furnace door will be forced open and several hundred pounds of unburned coal will spill out over the furnace-room floor. The answer to this problem is a special control known as a "hold-fire" control or a "stoker switch." This control is really a high-temperature thermostat, with a bimetallic element that protrudes into the upper portion of the combustion chamber. Frequently these controls are mounted by cutting a hole in the larger door of the furnace. The thermostatic element of the stoker switch operates *two separate switches*, the switches usually being of the mercury-tube type. The wiring of a stoker switch is shown in Fig. 3, the two switches being labeled S_1 and S_2. Switch S_2 has an adjustable setting with a range of perhaps 200 to 400°. When the temperature within the combustion chamber is below this setting, the stoker switch will cause the stoker to operate even though the thermostat is not calling for heat. The switch should be adjusted with this setting as low as possible so that the smallest possible fire, with the lowest possible firebox temperature, will be maintained when the system does not require any heat output. Switch S_1 on the stoker switch has no adjustment and operates at about 125°F. If the fire should go out, the temperature will drop below this point. When this occurs, the stoker cannot operate even though the room thermostat may call for heat.

Ignition Failure Controls. Oil Burners. Oil burners are ignited by means of an electric spark of a gas pilot-light flame when the burner is of the on-and-off type. Some burners never shut off completely, but drop

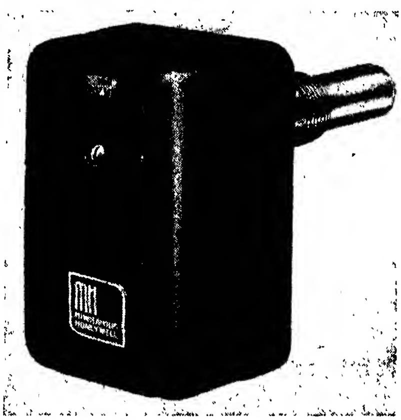


FIG. 4.—Immersion type aquastat. (*Minneapolis-Honeywell Regulator Co.*)

to a very low flame when the thermostat ceases to call for heat. In the first case it is possible that the ignition device may fail. In the second case it is possible that the burner may go out when operating on the low flame. If this occurs with the type of burner where oil is supplied into the combustion chamber by means of a pump, the combustion chamber can become filled with oil when the thermostat calls for heat. A protective device must be installed in the stack such that if the stack does not warm up within about 10 sec after the burner commences to operate the burner will be shut off. With the simplest type of protective switch in the stack, this leaves the house without heat, but it prevents the dangerous possibility of flooding the combustion chamber with unburned oil. Naturally if the burner starts properly, the stack warms up within 5 or 10 sec and this protection device does not have any cause to operate. Throughout our discussion of domestic controls, we shall show this protective device as a simple electric switch, which we shall call a "stack switch." See Fig. 9.

Aquastats. An aquastat is a thermostat with its temperature-sensitive element immersed in the water in the boiler. Aquastats may be used with either steam boilers or hot-water boilers. The original purpose of an aquastat was to permit the fuel-burning equipment to operate such that the boiler water temperature was kept at about 180°F, in order to heat domestic hot water during the summer. In modern hot-water heating systems as we shall see shortly in connection with Fig. 8, the burner is controlled all year round by the aquastat. Figure 4 shows an immersion-type aquastat, and Fig. 5 is an installation diagram.

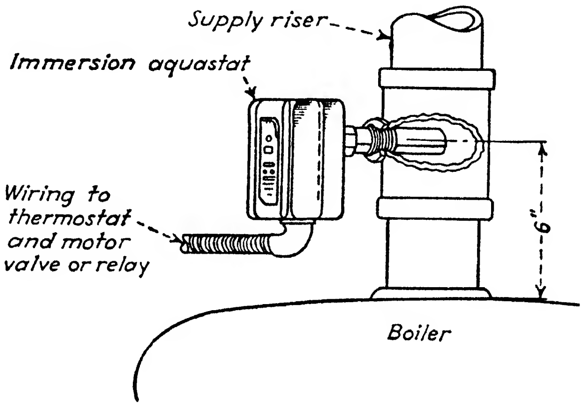


FIG. 5.—Showing installation of immersion-type aquastat. (*Minneapolis-Honeywell Regulator Co.*)

Low Water Cutoff. Steam boilers operate only partly filled with water. If the water level becomes too low, the boiler could be damaged by a hot fire. Therefore, with steam boilers, regardless of whether they are gas, oil or stoker fired, a *low water cutoff* is usually provided. This device is simply a float-operated switch that cuts out the fire if the boiler-water level becomes too low. Hot-water boilers require no water cutoff, because the boiler piping and radiators are entirely filled with water. Figure 6 shows a low water cutoff.

Domestic Oil-burner Controls. Steam and Hot Water. Figure 7 illustrates a frequently used control wiring circuit. Almost all modernizing installations, in which a new oil burner is installed to modernize a hand-fired steam boiler, require just this control arrangement. Note that the pressure control, stack switch, and low water cutoff are *all in series* with the oil-burner motor. In other words, any one of these devices could open the circuit and positively prevent the oil burner from operating. Current may flow through either the room thermostat or the aquastat in order to operate the oil burner. In other words, the thermostat and aquastat *are in parallel.* In summer if the house were at a temperature of 80°F, the thermostat would be in the off position. Nevertheless, if the boiler-water temperature dropped below 180°, the aquastat could operate the burner as required to increase the water temperature.

Figure 8 shows a wiring for oil-burner controls on a hot-water heating system. In Chap. IV we saw that the water could be circulated by con-

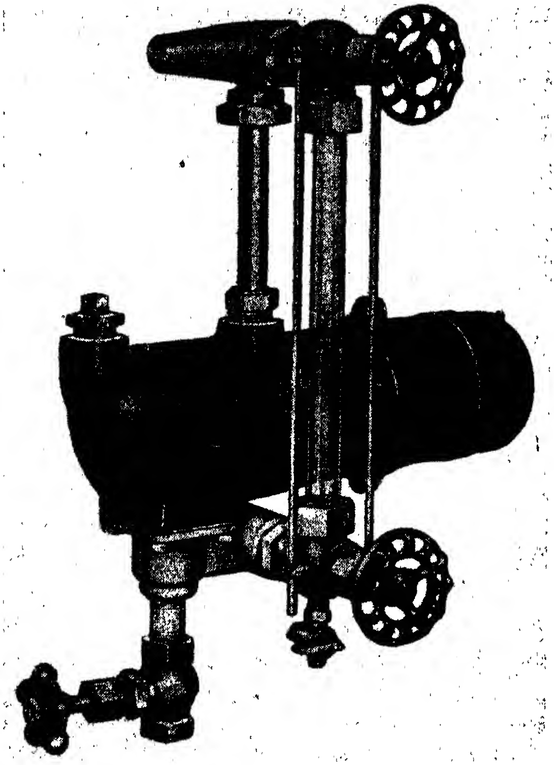


FIG. 6.—Low water cutoff. (*Minneapolis-Honeywell Regulator Co.*)

vection in a hot-water heating system. In Chap. XV we discussed in detail why these gravity-circulating hot-water systems are obsolete and showed the advantages of circulating the hot water by means of a pump. All modern systems use these circulators.

Note that we still have the stack switch, which we must have as a safety measure in connection with oil burners. The pressure control and low water cutoff are not required with hot-water systems. In Fig. 8 you can see that we control independently the oil-burner motor and the circulator pump motor. The circulator is controlled only by the room thermostat; when the thermostat calls for heat, the pump starts circulating water throughout the system immediately. Because the aquastat is the primary control of the oil burner, the boiler is filled with *water at* 180° *at all times.*

Let us suppose that the thermostat is located in the dining room and that someone closes all doors to the dining room and opens the windows. The thermostat would be subject to cold air and would call for heat. The

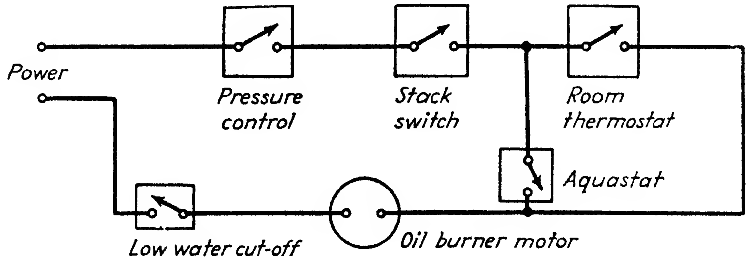


FIG. 7.—Standard control hookup of domestic oil-burner controls.

circulator would start to operate, and as soon as cold water from the radiators reached the boiler, the oil burner would start to operate. Because of the peculiar conditions we have assumed regarding the dining room, the circulator would run continuously until someone closed the windows. It is possible that all the water in the system would become so hot that water

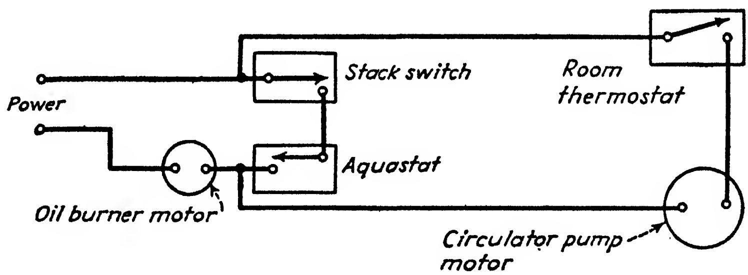


FIG. 8.—Wiring diagram for oil-burner controls on a hot-water heating system.

would be returning from the radiators at 183° or thereabouts. Under these conditions, the aquastat would stop the oil burner until the temperature of the water dropped a little below 180°.

Domestic Stoker Controls. Steam and Hot Water. In Fig. 3 you have already learned the action of the special stoker control known as the "stoker switch." To see how a stoker would operate in a steam system, study Figs. 3 and 7. In Fig. 7 remove the stack switch and oil-burner motor. Substitute the stoker switch and stoker motor, being careful to leave the room thermostat and aquastat in parallel as they are. This gives you the required arrangement, the pressure control and low water cutoff both remaining capable of stopping operation of the stoker. If the fire were to go out, switch S_1 in the stoker switch could also stop operation of the stoker. As indicated in Fig. 3, switch S_2 of the stoker switch, the aquastat, and the room thermostat would now be *all in parallel*; any one of the three could start the stoker.

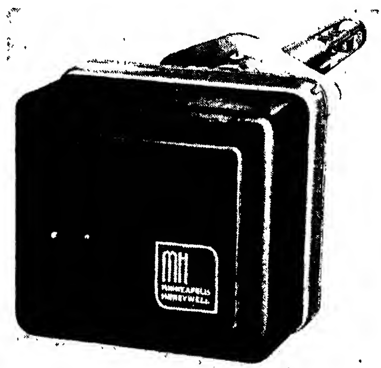


Fig. 9.—Stack control unit. Note stack element. (*Minneapolis-Honeywell Regulator Co.*)

How about stoker operation with a hot-water boiler? Again refer to Fig. 3, giving details of the stoker switch, but this time compare it with Fig. 8. The room thermostat in Fig. 8, controlling the circulator pump, has nothing to do with the method of firing the boiler. Hence this portion

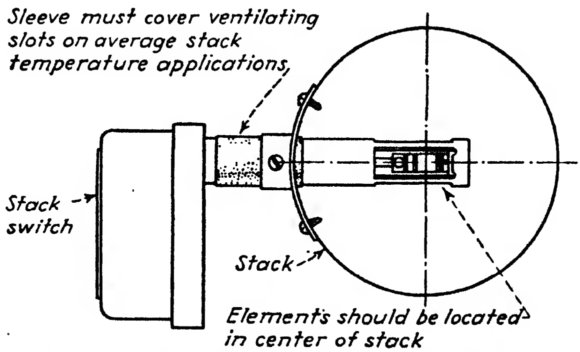


Fig. 10.—Installation of stack switch. (*Minneapolis-Honeywell Regulator Co.*)

of the circuit in Fig. 8 stays the same. Simply replace the stack switch and aquastat in Fig. 8 with the stoker switch and aquastat *as arranged in* Fig. 3. Also, of course, substitute a stoker motor for the oil-burner motor. You then have the control arrangement required for stoker-fired hot-water boilers. Switch S_2 of the stoker switch and the aquastat are in parallel, and either can start the stoker.

Control of Automatic Gas-burning Equipment. Even in large furnaces, gas is burned in a simple nonmechanical burner not very different from an ordinary gas-stove burner. A gas pilot remains continuously lighted, and there is only the problem of turning on and off the gas to start and stop the fire. The solenoid valve shown in Fig. 11 provides this automatic control of the gas.

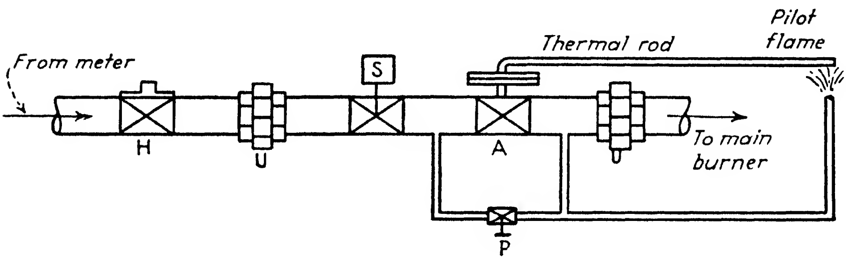


FIG. 11.—Diagram of control hookup of automatic gas-burning equipment.

Study Fig. 11 carefully. Gas-burning equipment, whether in steam or hot-water boilers or in warm-air conditioners, should always be arranged in a manner similar to Fig. 11. Instead of stack switch or stoker switch, the gas burner's safety device is the automatic shutoff marked *A* in Fig. 11. A few modern residences use gas as a fuel in direct-fired warm-air conditioners. The control arrangement required will be discussed in the next section.

Control of Automatic Gas-burning Equipment. Consider a gas burner installed in a steam boiler. Refer to Fig. 7. The stack switch is removed, and instead of the oil-burner motor the wires would connect to the gas solenoid valve shown in Fig. 11. The pressure control, low water cutoff, room thermostat, and aquastat would remain exactly as shown in Fig. 7. For a gas-fired hot-water boiler, turn to Fig. 8 and remove the stack switch. Substitute the solenoid valve of Fig. 11 instead of the oil-burner motor, and you have the completed control arrangement. *For both steam and hot water, the gas burner must be installed substantially as shown in Fig.* 11. Figure 12 shows a magnetic gas valve.

Control of Warm-air Conditioners. Let us imagine a house heated by a direct-fired warm-air conditioner. Let us again assume a thermostat on a dining-room wall and that the doors of the dining room are closed and all the windows opened. The thermostat would of course call for heat. All the house except the dining room would become overheated, perhaps to as high as 100°F. If the burner were permitted to operate continuously, it is possible that the entire conditioner might become *red hot.* If the V belt driving the fan were to break, it would be even worse. Certainly automatic controls must be provided to prevent this.

Let us assume another set of conditions for our warm-air-heated house. Suppose that in the spring we have several mild days during which no heat is required. With an oil or gas-fired conditioner, there will be no fire for these days and the entire conditioner will be at basement temperature, say 60°F. Suppose that it becomes cool some evening, the house temperature dropping low enough for the thermostat to demand heat. If the thermostat can start the blower *immediately*, what temperature air

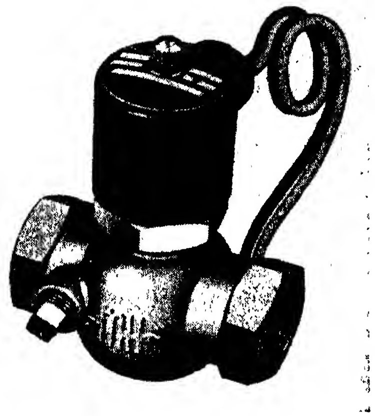


FIG. 12.—Magnetic gas valve. (*Minneapolis-Honeywell Regulator Co.*)

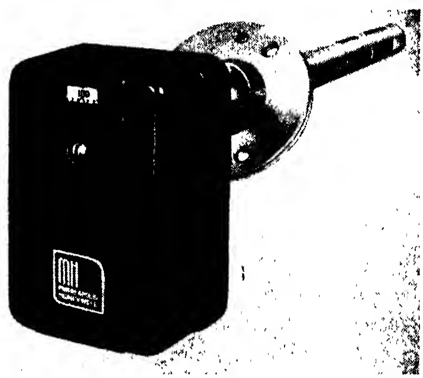


FIG. 13.—Airstat. (*Minneapolis-Honeywell Regulator Co.*)

would be discharged by the supply grilles? Since this air would come from the conditioner, which is at basement temperature, this air would be at about 60°F until the fire had been burning long enough to heat up all the metal in the conditioner. Obviously, we must provide automatic control to prevent this, because 60° air would create objectionable drafts and actually make the rooms colder.

Overheating of a direct-fired warm-air conditioner could be prevented by a single thermostatic control such as the Minneapolis Honeywell Airstat (see Fig. 13). This control is an on-and-off mercury switch and is in series with the burner. Thus, it can positively stop the burner if its temperature setting is exceeded. A separate control could be used to prevent fan operation when the bonnet of the warm-air conditioner is too cold. Such a control might be set at about 100°F, preventing the fan from operating if the temperature inside the conditioner were below this point.

In modern warm-air-conditioning installations, the functions of these two instruments are combined in a single thermostatic instrument with its temperature-sensitive element in the bonnet of the warm-air conditioner. It also may be located in the warm-air duct *very near* the conditioner.

Figure 14 illustrates the control wiring for a direct-fired warm-air conditioner using an oil burner and an air-conditioning furnacestat.

Trace the electric circuit in Fig. 14, neglecting for the moment the summer fan switch. Starting at power terminal A, note that the current must flow through the room thermostat first. In other words, the room thermostat is in series with all the equipment so that if it does not call for heat the entire equipment is shut down. Current then flows to the

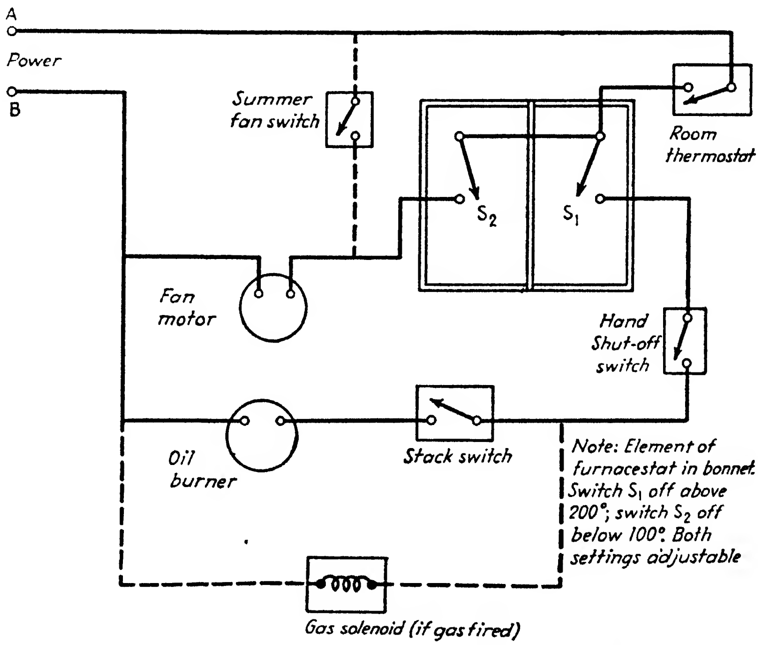


FIG. 14.—Schematic wiring diagram showing installation of gas solenoid valve.

furnacestat where it divides into two parallel circuits. Through switch S_2 current flows to the fan motor and back to the power line. Switch S_2 prevents the fan motor from running if the bonnet of the conditioner is cold.

From the thermostat, power flows through S_1, through a hand shutoff switch, through the stack switch, to the oil burner and back to the power line. Hand shutoff switches of this type are required on all oil-burner installations, usually being located at the top of the basement stairs. We have not shown these hand shutoff switches on previous control diagrams because, except on warm-air conditioners, they are for emergency use only. If the fan should cease to operate because of trouble with its V belt or motor, switch S_1 prevents the furnace bonnet from overheating.

For summer operation it is desirable to use the fan for circulating filtered air even though the installation is not equipped for summer cooling. In summer, the oil burner is shut off by means of the hand shutoff switch. The fan is operated by means of the summer fan switch. This fan switch is hand-operated and may be located at any convenient point in the house.

If the conditioner is gas fired instead of oil fired, the dotted wiring at the bottom of Fig. 14 shows how the gas solenoid would be connected.

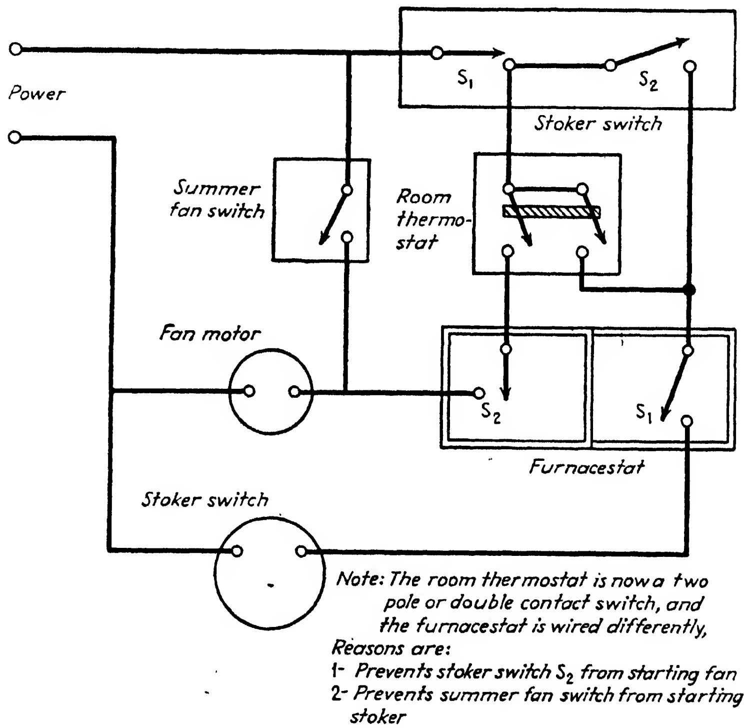


FIG. 15.—Schematic wiring diagram of stoker-fired conditioner.

The stack switch and oil burner are replaced by the dotted wiring. This solenoid would be complete with other gas-burner controls exactly as shown in Fig. 11. If the conditioner were stoker fired, the control wiring would be as shown in Fig. 15. If you are familiar with the stoker switch as first explained and shown in Fig. 3, and if you understand Fig. 14, Fig. 15 should be self-explanatory. Figure 16 is a room-type thermostat.

Control of Humidifiers. Many warm-air conditioners use pen or drip-type humidifiers. These are ordinarily controlled by a float within the conditioner, the float being subject to a manual adjustment outside the conditioner. When a humidifying device having a spray nozzle is used, a solenoid valve is ordinarily inserted ahead of the nozzle. A humidistat in the conditioned space energizes the solenoid when the relative humidity drops below the humidistat setting. As soon as the humidity in the conditioned space is brought up to that required to satisfy the humidistat, the circuit is opened and the solenoid shuts off.

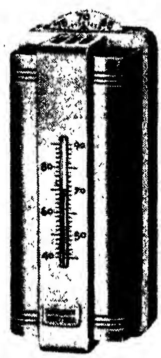


FIG. 16.—Room-type thermostat. (*Minneapolis-Honeywell Regulator Co.*)

An ordinary humidistat as just described has the disadvantage that in cold weather it must be manually set at a lower relative humidity to avoid condensation on windows. A special control instrument manufactured by Julian P. Friez and Sons, Baltimore, Md., known as the "windowstat" takes care of this automatically. This instrument is mounted near a window. The instrument consists of a humidistat, the setting of which is modified depending upon the temperature near the inside surface of the window. It functions to maintain as high a relative humidity as possible without permitting condensation on the window.

Domestic Hot Water. We have seen that with steam or hot-water boilers an aquastat can maintain a temperature of the water in the boiler always at 180°. With such an arrangement, domestic hot water can easily be obtained by locating a heat-exchanger coil outside the boiler. Such heat exchangers usually consist of a cast-iron casing inside of which there is a coiled copper tube. Hot water from the steam or hot-water boiler is inside the cast-iron casing, while the domestic hot water is inside the copper tube. Turn to Fig. 10 on page 87; imagine the coil shown at the left of this figure encased in a jacket of 180° water instead of heated by a flame, and you have the complete arrangement.

With a warm-air conditioner we have seen that we wish to use the fan to circulate air in the summer. Therefore we must have no fire in our conditioner. For this reason a separate domestic hot-water heater is always used with a direct-fired warm-air conditioner. Such domestic hot-water heaters may have gas burners or may burn a light fuel oil or range oil. In either case, a low pilot light burns all the time and an aquastat measuring the temperature below the middle of the storage tank controls the flame. This aquastat may have its temperature-sensitive element immersed in the tank, or it may be of the surface type which is simply clamped to the outside of the tank.

Some heating installations using steam or hot-water boilers also have separate domestic hot-water heaters, instead of using the heat-exchanger arrangement explained above. In this case the allowance for domestic hot-water heating, which is usually made in sizing the boiler, of course need not be made. While the use of a separate domestic hot-water heater increases installation cost, it is cheaper to operate than securing domestic hot water as an auxiliary from the heating boiler. The reason is that the hot-water tank operates at low temperature, is thoroughly insulated, and has small heat losses.

Basic Refrigeration Controls. Naturally one of the most important refrigeration controls is the expansion valve. Expansion valves are discussed in detail in Chap. XI. In the light of what we now know about

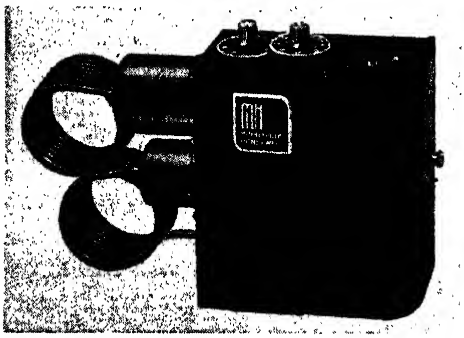


FIG. 17.—Polartron refrigeration control. (*Minneapolis-Honeywell Regulator Co.*)

controls, we can see that high- and low-side floats are simply float valves. Also an automatic expansion valve is a direct-acting modulating pressure controller, operating a refrigerant needle valve. Finally a thermostatic expansion valve is a combined pressure controller and direct-acting modulating thermostatic valve. There are a few other important fundamental refrigeration controls, some of which you may already have encountered in the shop.

High- and Low-pressure Cutoffs. Such instruments consist of two pressure controllers with their bellows connected by means of levers to the same electric switch. The switch may be a snap-acting metal contact switch or a mercury tube. One pressure bellows is connected (by quarter-inch copper tubing) with the low side of the refrigeration machine. When the low-side pressure drops *below* an adjustable setting, say 35 psi, this bellows *opens* the electric switch and *stops* the compressor. The differential setting on this low-pressure cutout is adjustable. It might be set for 5 psi, in which event the compressor would restart when the low-side pressure reached 40 psi.

The high-pressure bellows is connected by means of a small copper tube to the high side, usually right at the compressor discharge. This bellows acts to shut off the compressor (by tripping the same switch to which the low-side bellows is connected), if the high-side pressure *exceeds* the control setting. The high-side cutout pressure is adjustable and is usually set between 150 and 180 psi. In some instruments there is an adjustable differential on the high-side cutout, while in others the differential is fixed at 10 to 20 psig.

In the case of small single-phase motors requiring only two power wires to the motor, the single-pole single-throw switch in the high-low pressure cutout may be wired in series with the compressor motor. In this case, the full power drawn by the compressor motor flows through the control switch. For larger compressors, including all compressors driven by three-phase motors, the switch in the high-low pressure cutout is wired in series with the magnetic coil in the magnetic contactor that acts as motor controller for the compressor motor.

Liquid-line Solenoids. In many cases, more than one evaporator coil is handled by a single compressor. In this event if we desire to stop the refrigerating effect of one of the evaporators, we have only to shut off the supply of liquid refrigerant to this evaporator. Any humidistat or thermostat having a single-pole single-throw switching action can be wired in series with a solenoid valve located in the liquid line. Thus the thermostat or humidistat will control whether or not liquid is admitted to a certain evaporator.

In some cases, solenoids are used in liquid lines even when there is a single evaporator operated by the compressor. With these installations, as well as with some multiple-evaporator installations, the solenoids in the liquid lines may be subject to additional control besides the thermostat or humidistat. The power for the solenoid in such cases is not taken directly from a power line. Instead, single-phase power is taken by connecting to any two of the wires leading from the motor starter to the compressor motor. This means that the solenoid can be energized and opened only *when the compressor motor is running.* When the compressor motor stops for any reason, the solenoid closes. While the compressor motor is running, the solenoid is positioned by its controlling thermostat or humidistat.

Unloading Devices. Multiple-cylinder compressors are often equipped with "unloading devices" which cut out one-half, one-quarter, or three-quarters of the cylinders of the compressor. These devices are built into the compressor and act to decrease the compressor capacity when full capacity is not needed, with a corresponding reduction in the power required to drive the compressor. The actual construction and mechanical operation of unloading devices need not concern you because they are built into the compressors and are designed differently by different manufacturers. The basic principle of operation is important and is substantially the same in all cases.

You all know that when the heat input to an evaporator is decreased the rate of evaporation of the refrigerant is decreased. If the compressor capacity remains constant, the low-side pressure will drop. With air-conditioning refrigeration, a reduced need for cooling means a smaller input to the evaporator. Compressor unloading devices are sensitive to evaporator pressure. As evaporator pressure drops below the normal full-load operating condition, cylinders of the compressor are cut out either independently or in pairs.

Cylinder unloading can be arranged to cut out half of the cylinders of the compressor, thus reducing its capacity 50 per cent. This is the cheapest arrangement but gives only two capacities, 100 and 50 per cent. Provided the machine has four or more cylinders, more capacity-reduction steps can be obtained. Usually four or five steps of unloading or capacity reduction are the maximum used, although theoretically with an eight-cylinder compressor there may be eight different steps.

Conclusion. Our purpose in this chapter has been to become familiar with control instruments and devices themselves and with basic control principles. Except for domestic-heating control systems, we have not shown how control instruments are used in complete control systems. The next chapter will consist of sketches of complete control systems for different typical summer, winter, and year-round installations.

Use the two outlines given you in this chapter as a test of your understanding of this subject matter. Remember that a thermostat or a humidistat can be made to *open or close* any damper or valve, or to *stop or start* any motor when the temperature or relative humidity *either rises or falls* beyond the control setting.

In comparing pneumatic and electric control systems, a few general statements can be made. Pneumatic control systems involve a small air compressor and tank similar to that used in gasoline stations to provide compressed air. Since the entire control system depends upon this compressor, it is customary to install two compressors, the second being merely a stand-by in case the first should fall. Because of the expense of these compressors, pneumatic control is limited to large installations. A small (25- to 50-ton) commercial installation on a single floor might involve at most two or three thermostats and one or two humidistats. In such a case pneumatic control would be rejected because of high installation cost, due to the need for two air compressors. On large installations, however, pneumatic controls often are cheaper, because for *modulating* control the instruments themselves are less costly than electric modulating instruments.

Controls may be obtained to any degree of accuracy that the customer desires and is willing to pay for. We are familiar with two air-conditioning installations each of which cost approximately $35,000, *not including* controls. The controls for one cost about $1,300 installed; the controls for the other cost about \$16,000 installed. The very expensive controls were accurate to plus or minus *one-tenth* of one degree Fahrenheit, ***both*** as to *dry bulb* and *wet bulb*. In ordinary field work, much cheaper controls are used, controlling to about plus or minus one-half of one degree dry bulb, and plus or minus approximately 3 per cent relative humidity.

CHAPTER XXVII

APPLICATION OF AUTOMATIC CONTROLS. COMPLETE SYSTEMS

General. From the discussion in Chap. XXVI you have become familiar with different types of control instruments, how they work, what they can do, and how they are used in some simple control systems. Our study of control systems was limited to domestic-heating controls, which have become fairly well standardized. Also, these systems are simple as to wiring, being just series-parallel electric circuits. Therefore, in Chap. XXVI when we studied a control system we studied it as an actual wiring diagram or electric-circuit diagram.

With more complicated control systems, the wiring diagrams get quite complicated, so much so that the operation of the control system would not be clearly evident from a study of the wiring diragam. In addition, pneumatic controls could be used instead of electric devices, and some valves or dampers might be operated by self-contained (or direct-acting) controls. Finally, wiring and piping details for electric and pneumatic controls would vary a little depending on the control manufacturer. This means that we want a new type of diagram to use as a basis for study. We shall choose what is known as a "control schematic diagram"; this shows a sketch of the ducts, pipes, coils, dampers, valves, etc., with the controls in place.

In such a control schematic if a certain thermostat controls a certain damper, we simply draw a dotted line connecting the thermostat and damper. We show the relation between automatic controls and the devices they operate in this manner, and in addition we need a short description to explain exactly the control operation and sequence. As an introduction to this method of representing a control system, let us consider a specific example based on the domestic-heating controls we studied in Chap. XXVI. Let us build up both a schematic and a wiring diagram, and we shall see that the wiring diagram is unnecessary (except for actual wiring of the controls during installation). We shall then learn a few basic principles regarding the use of controls in complete systems, after which we shall examine several systems for winter, several for summer, and one or two for year-round use.

Zone Control. Three-family Residence. If a typical three-story three-family residence were heated with a one-pipe steam system of best quality, using a single boiler, it would be impossible to produce proper winter-comfort conditions in all three apartments with one thermostat. Let us consider such a building, assuming that separate hand-fired boilers have been previously used, now to be replaced by a single large oil-fired boiler. In the first place, let us review Chap. XVI and see what would happen if we used a single thermostat and the typical one-family oil-fired steam control system we studied. We have three possible locations for our thermostat, since there are three apartments in the building.

Suppose we locate the room thermostat at a typical inside wall of the *first-floor* apartment. You know that the radiators in this apartment will heat first, in spite of any adjustable air-vent valves. Hence the first floor would warm up, satisfy the control, and stop the oil burner long before the third-floor radiators would be warm. If the control were located in the third-floor apartment, the oil burner would have to run till this floor was warm enough, driving the occupants of the lower floors to shut off radiators or open the windows. Location of the thermostat on the second floor would be a little better, but this would overheat the ground floor and supply too little heat for the top floor.[1] What we seem to need is a thermostat on *each floor*, each one controlling the steam to its own floor only.

Refer to Fig. 1. Points A, B, and C are points on the old steam mains in which each was disconnected from its old steam boiler. A motorized steam valve of the on-and-off type is mounted at each of these points, all three pipes then being connected to a new steam main or "header" from the new boiler. Valve M_1 controls steam to the first floor and is under command of thermostat T_1, as indicated by the dotted line. Thermostats T_2 and T_3 are located on the second and third floors, and operate valves M_2 and M_3, respectively. Thus we are assured that each floor gets heat as it needs it, independently of the other floors. All objections of a single-thermostat arrangement cited in the preceding paragraph are eliminated.

How can we be sure that steam will be available whenever *any one* of the three thermostats calls for steam? We could have a boiler always having steam at about 2 psi, operating under its pressure control. With this setup we would waste fuel in mild weather (and would have domestic hot water hotter than necessary). We can improve on this by having the boiler simply run *under command of its aquastat* when *all three* thermostats have their *valves closed*. When *any one* of the three valves opens in response to a call for heat on its floor, the oil burner will start. If the burner commences thus with *only one* of the valves open, the boiler is only supplying

[1] The occupants of different floors may like different temperatures; one family may go away for several weeks; the heating load on the third floor is greater than either of the other floors, because of one unheated space above; all this makes the single-thermostat arrangement unsatisfactory even in a two-story two-family house.

about one-third of its capacity in radiation, so the pressure would tend to build up. The pressure control would be set to cut out at about 2 psi, with a differential of about ½ lb. The dotted line running from each of the valves to the burner indicates this control relationship.

The wiring for this control system is shown in Fig. 2, but for simplicity we have shown only two of the three thermostats and valves. The aquastat is in parallel with switches S_1 and S_2, so any one of them can start the burner. Compare Figs. 1 and 2, and you will see that the wiring diagram requires considerable study to understand and is more difficult

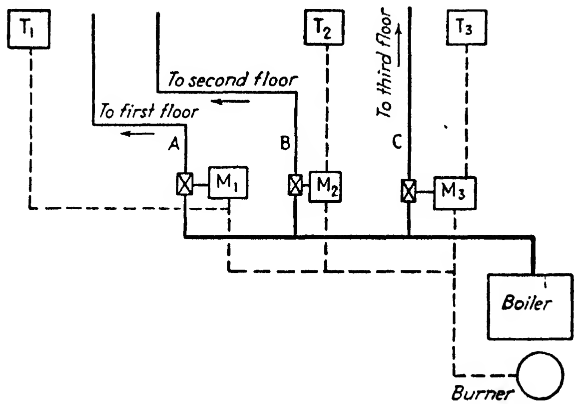


Fig. 1.—Schematic of heating zone control.

to draw than the simple schematic. Ordinarily we use schematics like Fig. 1, with a short written explanation, as the means of illustrating control systems. Study the winter control system on this page, and you will see that it is simpler and clearer than its wiring diagram on the next page. We shall use only the schematic form from now on.

Compensated Control. In the control systems on pages 546 to 554 you will often see the phrase "compensated control," especially in the "alternate" arrangement described at the end of each explanation. With compensated control, the compensated thermostat (or humidistat) does not remain set at a fixed temperature (or relative humidity). Instead, the setting of the instrument is *automatically changed* as some other control instrument is subject to variations in temperature (or relative humidity). Let us consider some examples of compensated controls. In Chap. XXVI we discussed the Friez windowstat. We said that the Friez windowstat consisted of a humidistat, the setting of which *changed, depending upon the temperature* a few inches from exposed window glass. On the coldest days, the windowstat might operate at a relative humidity setting of 20 per cent. In milder weather its relative humidity setting might rise as high as 40 to 50 per cent. In other words, the Friez windowstat is a *compensated humidistat, compensated according to temperature just inside windows.*

In Chaps. XIII and XVIII we learned that is desirable to decrease the indoor temperature in the case of a summer conditioning job as the outdoor temperature becomes lower. In other words, we do *not* leave the thermostat for the conditioned space set at a constant temperature. The conditioned-space thermostat must be set by hand at the proper indoor temperature, unless it is of the compensated type. Using a compensated thermostat, its setting would *automatically* be varied by a compensating thermostat with its bulb outdoors. The compensating thermostat would vary the

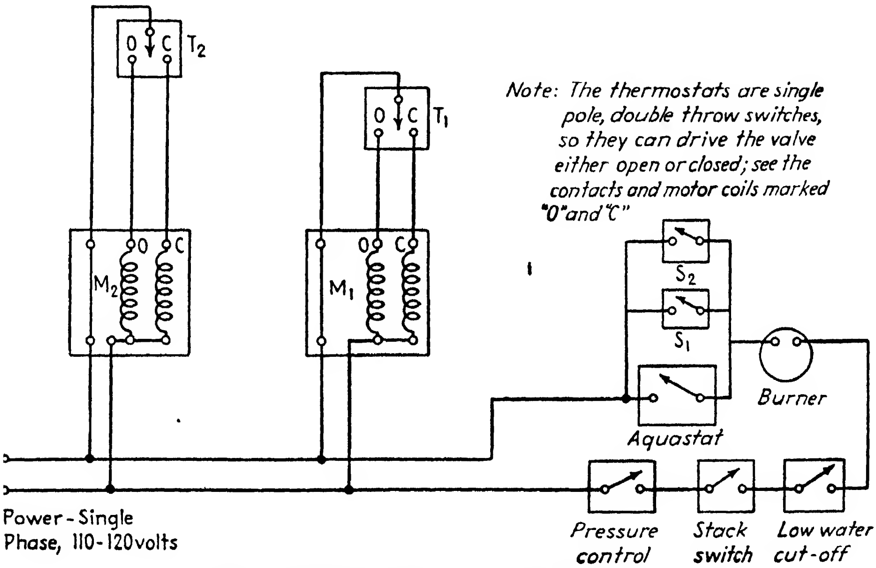


FIG. 2.—Wiring diagram of heating-zone control.

setting of the compensated thermostat from 72 to 80°, depending upon outdoor temperature. It is possible to have any thermostat or humidistat compensated by another thermostat or humidstat whenever the expense is justified by the results.

Location of Conditioned Space Controls. Certain thermostats, such as chilled-water thermostats and domestic aquastats, are located in specific locations with which we are familiar. In residences, room thermostats are ordinarily located on an inside wall such that they are clearly visible and that they measure a fair average temperature of the house.

In larger installations it is not desirable to control temperature by a thermostat in the conditioned space. Instead, the thermostat is better located in the return-air duct. After all, where does return air come from? It is withdrawn *from the conditioned space* and hence represents the best possible average of the temperature and humidity conditions in the conditioned space. The same applies to the location of a humidistat or wet-bulb controller, which is to control the condition within the conditioned space.

Selection of Automatic Controls for Application to Central Fan Air Conditioning Systems[1]

Selection of Automatic Control Equipment. Too much stress cannot be laid upon the selection of the proper automatic controls for all types of air conditioning systems. The total number of controls used on an air conditioning system is not necessarily the deciding factor as to what the results will be because there is almost as much danger in using too many controls as there is in using too few. In general, however, the tendency has been toward the use of too few controls, and the failure of many air conditioning installations to produce the results expected of them can be traced directly to the improper selection and application of the automatic controls.

There are usually several different possible combinations of automatic controls which may be used to produce good results on each individual air conditioning installation, but there is also usually one particular combination of controls which will produce better and more economical results than any of the others.

Application of Automatic Controls to Central Fan Systems. It is obviously impossible to show here control systems for every type of air conditioning system. For this reason this section shows only those control systems which are more commonly used and which have proved to be entirely satisfactory from the standpoint of results obtained, ease of installation, and accuracy of control.

These typical systems are purely schematic. They are not drawn to scale, and only those portions of the air conditioning system which are directly affected by the automatic control system are shown. Therefore, filters, stationary dampers, etc., are omitted from these drawings.

In the following descriptions of various control systems, suggested control settings at the control devices are mentioned, these are given as examples only and may vary considerably depending upon type of installation, load conditions, and geographical location.

These typical systems have been divided as follows:

1. *Ventilating and Warm Air Systems.* Central fan ventilating systems may be subdivided as follows:

a. Split-systems by which air is supplied for ventilating purposes only and heat in winter is supplied by another source such as direct radiation.

b. Warm air systems by which both the ventilation and heating are supplied by the central fan system.

2. *Cooling Systems.* Central fan cooling systems may be divided as follows:

a. Closed coil systems which may use either cold water coils or direct expansion coils.

b. Air washer systems.

[1]Courtesy of Minneapolis-Honeywell Regulator Co.

3. *All-year Conditioning Systems.* An All-Year Central Fan Conditioning System is a combination of a ventilating or heating system and a cooling system. A complete control system for an all-year conditioning system will therefore consist of combinations of the systems of controls already described, with minor changes in some cases. The final system of control in most cases should provide automatic changeover between cooling and heating cycles. This feature is of great importance since the Spring and Fall seasons may require both heating and cooling during the same 24-hour period.

When an air washer is used during the heating cycle, heat must be supplied either to the air entering the sprays or to the water used in the sprays in order that the dew-point temperature of the air leaving the air washer may be maintained at the proper point because air in taking up moisture loses sensible heat which must be added in some way.

Conclusion. After having completed Chaps. XXVI and XXVII you should have a sufficiently complete knowledge of automatic controls so that you can *work intelligently* with a control manufacturer's representative in designing a control system. You should be able to do this before leaving the subject of controls. You should *not* attempt to design control systems and select control instruments without such assistance for any installation except standard "run-of-the-mine" jobs, regarding which we have studied particular layouts. If any special control problem is encountered, such as those involving minimum cost, extreme accuracy, complicated zoning, or other special problems, remember that the control manufacturer offers the specialized service of his representative.

Typical Control Systems. Figure 3 illustrates a heating and ventilating central fan air conditioning system. The control system is designed to use only a minimum amount of outdoor air when heating, although when cooling is required this amount is increased.

Operation. Insertion thermostat T_1 measures the temperature of the return air and positions modulating motorized valve V_1 in accordance with the heat requirements of the conditioned space. Insertion thermostat T_2 measures the temperature of the air being discharged to the room and prevents, by opening motorized valve V_1, the temperature of the air delivered to the room falling so low (such as 65°) that uncomfortable drafts are apt to occur. As the return air temperature rises, T_1 will close the motorized valve V_1 and if the return air temperature continues to rise approximately one degree, indicating a need for cooling in the conditioned area, T_1 will shift the control of damper motor M_1 to insertion thermostat T_4, to take a quantity of outdoor air for cooling. This quantity will vary with the outdoor air temperature to maintain air entering coil at approximately the setting of T_2. Humidity controller H_1 operates solenoid valve V_2 so as to add moisture to the air when required. A control panel contains equipment for varying the minimum position of the outdoor dampers and closing the damper when the fan stops.

Note: If this system is to be used as a straight ventilating system insertion thermostat T_1 would perform the function of aiding in a pickup after a shutdown period. If the pickup feature is not desired, T_1 may be omitted and V_1 will be controlled directly from T_2.

Alternate No. 1. In the compensated control system, T_3 varies the control point of H_1 according to a predetermined schedule as the outdoor temperature changes, in order to prevent frosting of windows.

Figure 4 illustrates a heating and ventilating central fan air conditioner having reheat and preheat coils. The control system is designed to use a minimum amount. of outdoor air when heating but to use more when winter cooling is required

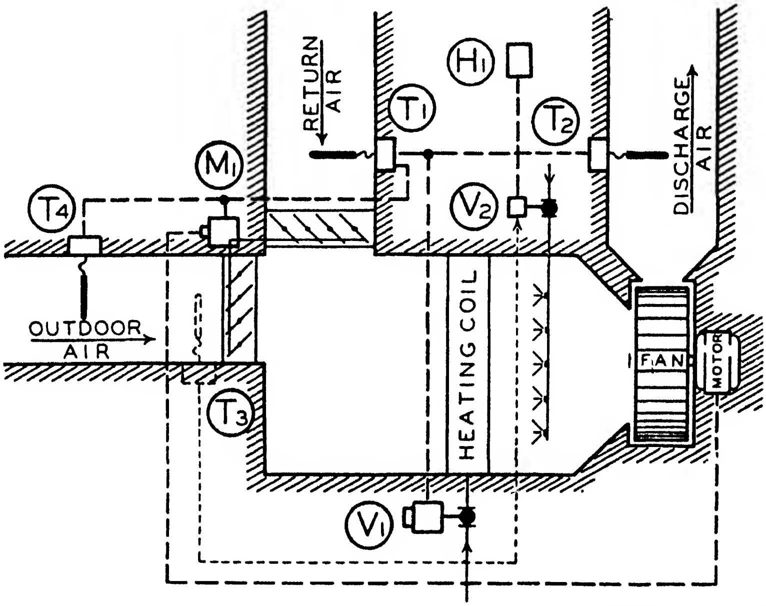


FIG. 3.—Heating and ventilating central fan air conditioning. (*Minneapolis-Honeywell Regulator Co.*)

Operation. When fan is started the face damper on the preheating coil is opened sufficiently to provide a minimum amount of outdoor air at all times. If the outdoor air temperature is below the setting of insertion thermostat T_4 (such as 35°), the two position valve V_2 will be opened, admitting steam to the preheat coil. Insertion thermostat T_1 measures the temperature of the return air and modulates steam valve V_1 according to the heat requirements of the conditioned space. Insertion thermostat T_2 measures the temperature of the discharge air being supplied to the space and opens steam valve V_1 so as to prevent the air from being discharged into the space at a temperature below the setting of T_2 (such as 65°). If the return air temperature is too warm, indicating a need for cooling, a second contact on T_1 permits insertion thermostat T_3 to assume command over damper motor M_2 so as to maintain the air entering the reheat coil at a temperature which will provide cooling (such as 65°). Humidity control H_1 operates solenoid valve V_3 so as to add moisture to the air when required. Damper motor M_1 is controlled by a manually operated switch in the central panel to adjust the minimum amount of outdoor air required for ventilation. A control panel contains the equipment necessary to close the outdoor damper when the fan stops and to adjust the minimum quantity of outdoor air. All control units are wired directly to this control panel.

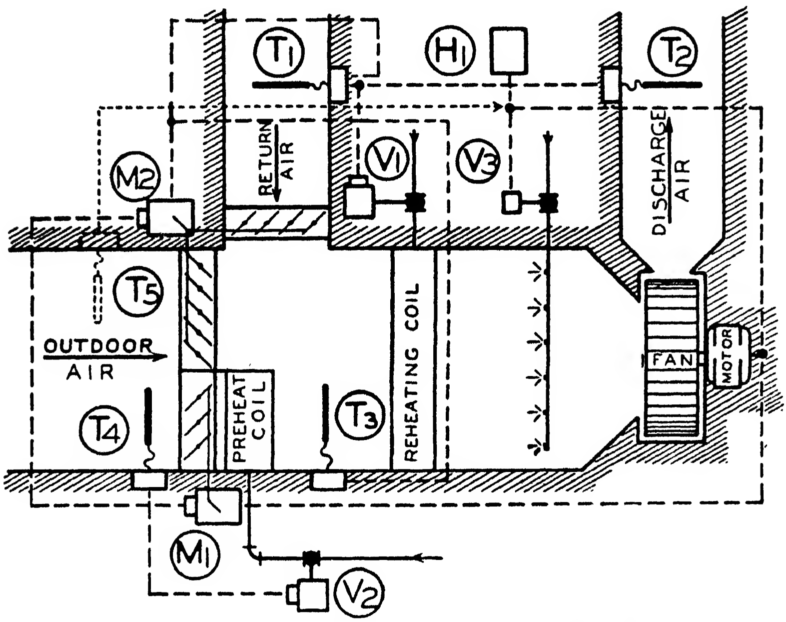


FIG. 4.—System having preheat and reheat coils. (*Minneapolis-Honeywell Regulator Co.*)

Alternate No. 1. In the compensated control system the relative humidity in the space will vary as the outdoor temperature changes. With this system insertion thermostat T_5 will change the control point of H_1 according to a predetermined schedule so as to prevent the frosting of windows.

Figure 5 illustrates a zoned central fan conditioner which supplies constant temperature air to zone booster coils. The booster coils are controlled by modulating type zone thermostats. This control system would also be applicable for a straight ventilating system without zone booster coils where it is desired to discharge air at a constant temperature.

Operation. Zone thermostat T_1, which may be either a return-air insertion thermostat or a room thermostat, is of the modulating type and controls booster coil modulating valves V_1 to proportion the amount of heat added to the air in proportion to the requirements of the conditioned space. Insertion thermostat T_2, located after the main heating coil, modulates motorized valve V_2 to maintain a fixed temperature being supplied to the zone coils (such as 65°). This is usually somewhat lower than the temperature desired in the conditioned space so that when modulating valves V_1 are closed, cooling may be obtained. Insertion thermostat T_3, located after the fan, measures the temperature of the mixed outdoor and return air and positions damper motor M_1 to take varying quantities of outdoor and return air sufficient to maintain a fixed temperature entering the main heating coil (such as 65°).

Humidity controller H_1, located in the return air, operates solenoid valve V_3 to add moisture to the air when required.

Control panel P_1 contains the equipment to adjust the minimum quantity of outdoor air and controls to close valve V_3 and the outdoor damper when the fan stops.

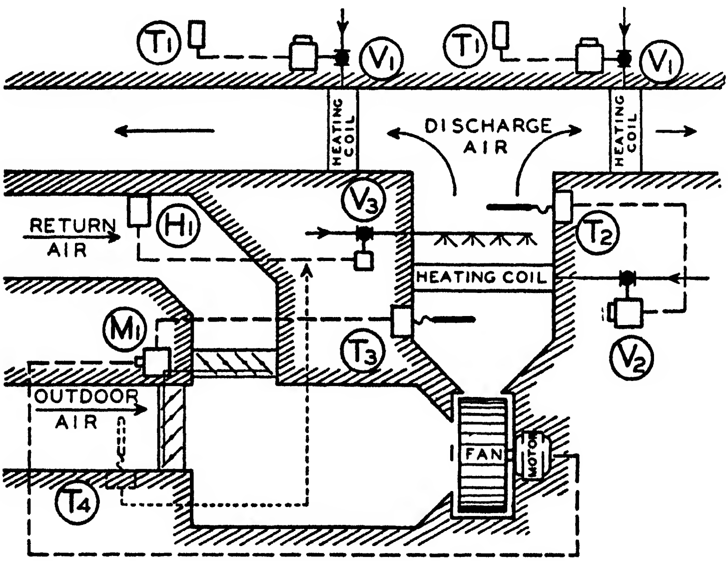


FIG. 5.—Zoned central fan conditioner supplying constant temperature air to zone booster coils. (*Minneapolis-Honeywell Regulator Co.*)

Alternate No. 1. In the compensated control system insertion thermostat T_4 changes the control point of H_1, as the outdoor temperature changes according to a predetermined schedule so as to prevent frosting of windows.

Figure 6 illustrates a central fan cooling system using cold water or brine as the refrigerant. The control system is designed to control the flow of water passing through the coil in proportion to the cooling requirements. Outdoor air is used for cooling whenever possible.

Operation. Insertion thermostat T_1 in the return air controls valve V_1 to control the flow of water to the coil. This control can be made to modulate or control two positions as desired. The outdoor damper is closed to a minimum except when the outdoor temperature as measured by insertion thermostat T_2 is low enough to be used for cooling. If the characteristics of a given installation permit undercooling by outdoor air, a low limit thermostat may be placed in the return air to close the outdoor air damper.

Alternate No. 1. If the compensated control system is used, insertion thermostat T_2 will vary the control point of T_1 as the outdoor temperature changes according to a predetermined schedule.

Alternate No. 2. A second alternate system utilizes the humidity controller H_1, and the insertion thermostat T_1. Humidity controller H_1 positions valve V_1 within the limits of temperature set up by insertion thermostat T_1. If the return air temperature falls too low, the second coil on T_1 will modulate V_1 toward the closed position.

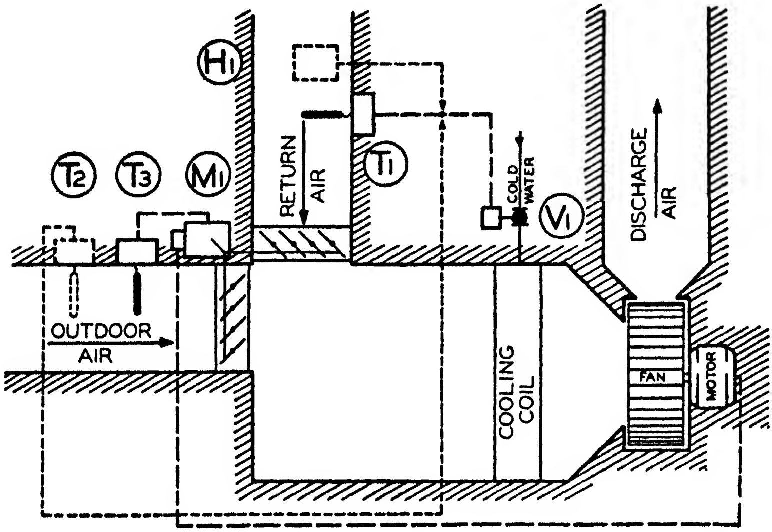


FIG. 6.—Central-fan cooling system using cooling water or brine. (*Minneapolis-Honeywell Regulator Co.*)

Figure 7 illustrates a central fan conditioner using a bank of four direct expansion coils for cooling. The coils are brought on in sequence as the cooling requirements of the space increase. The outdoor damper remains closed to a minimum except when outdoor air can be used for cooling.

This system is also used for the control of multistage compressors, in which event a circuit is provided which protects the compressor motors in event of power failure. This can be furnished with or without sequence changing switches.

Operation. Insertion thermostat T_2 controls damper motor M_1 to take a large quantity of outdoor air on mild days and to close the damper to an adjustable minimum when the outdoor temperature rises. If there is danger of undercooling by outdoor air a low limit thermostat may be located in the return air to close the outdoor air damper on temperature fall. Insertion thermostat T_1 controls a step controller in a control panel to bring on coils in sequence as additional cooling is required. The control panel also contains the equipment for varying the minimum outdoor damper position and closing that damper when the fan stops.

Alternate No. 1. If the compensated control circuit is used, insertion thermostat T_3 varies the indoor temperature control point of T_1 as the outdoor temperature varies according to a predetermined schedule.

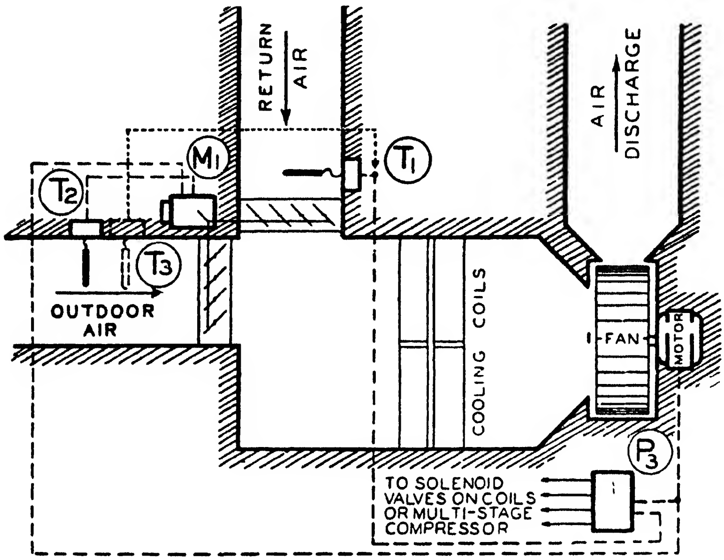


FIG. 7.—Central fan conditioner with bank of four direct-expansion cooling coils. (*Minneapolis-Honeywell Regulator Co.*)

Figure 8 illustrates a central fan cooling system using face and bypass dampers before a direct expansion coil. A minimum amount of outdoor air is used except when it may be used for cooling. If there is danger of undercooling by outdoor air, a low limit thermostat may be used in the return air to close the outdoor air dampers.

Operation. Insertion thermostat T_1 measures the temperature of the return air and modulates damper motor M_2 to position the face and bypass dampers in proportion to the cooling requirements of the room. Insertion thermostat T_3 with its bulb in the outdoor air, positions damper motor M_1 to take a maximum amount of outdoor air on mild days. By the use of face and bypass dampers the minimum coil temperature can be maintained at all times which will give a maximum amount of dehumidification possible with systems of this type.

A control panel contains equipment for varying the minimum outdoor air quantity and closing the outdoor damper on fan shutdown.

Alternate No. 1. If the compensated control system is used, insertion thermostat T_2 will vary the control point of T_1 as the outdoor temperature changes according to a predetermined schedule.

Figure 9 illustrates an air washer which bypasses return air, resulting in a variation of the sensible latent heat ratio.

Operation. Insertion thermostat T_1 measures the temperature of the return

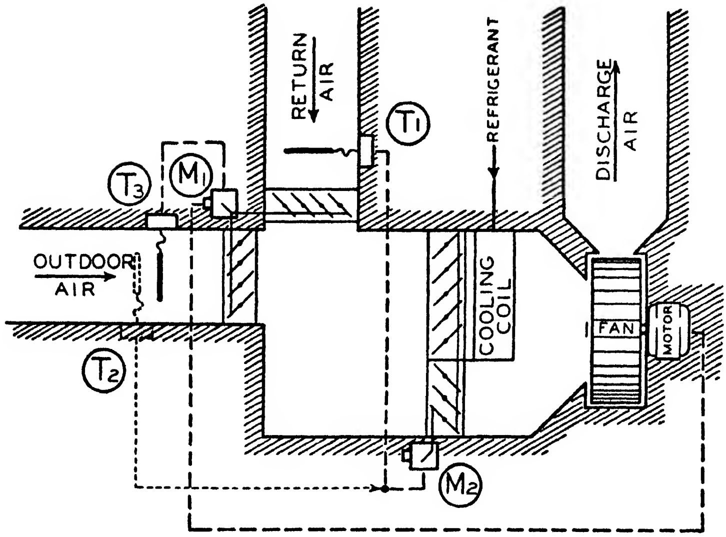


FIG. 8.—Central-fan cooling system with face and bypass dampers and direct expansion cooling coils. (*Minneapolis-Honeywell Regulator Co.*)

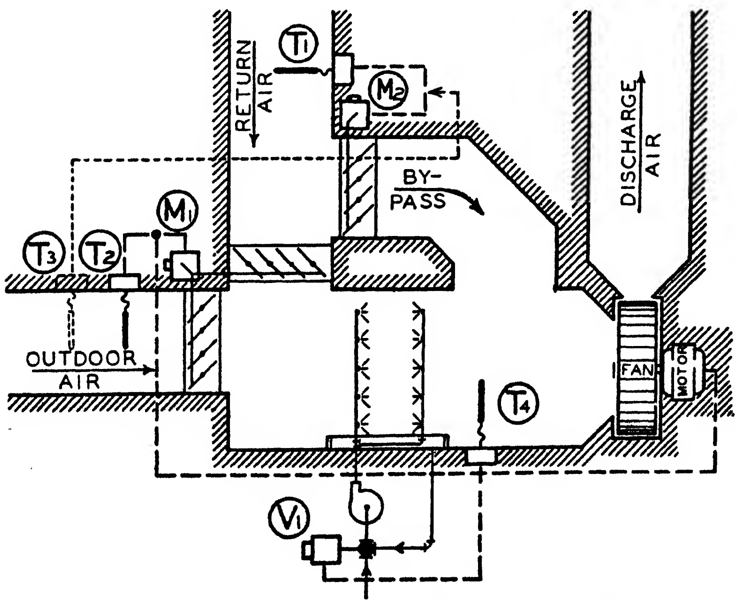


FIG. 9.—Air washer with return-air bypass dampers. (*Minneapolis-Honeywell Regulator Co.*)

air and modulates motor M_2 to send varying quantities of air through and around the washer to maintain the desired temperature in the conditioned space. Insertion thermostat T_2 with its control bulb in the outdoor air modulates damper motor M_1 to increase the amount of outdoor air used when its temperature permits its use for cooling. If there is danger of undercooling by outdoor air a low limit thermostat located in the return air may be used to close the outdoor air dampers. Controls in a panel close the outdoor air damper when the fan stops and provide an adjustable setting of the minimum quantity of outdoor air.

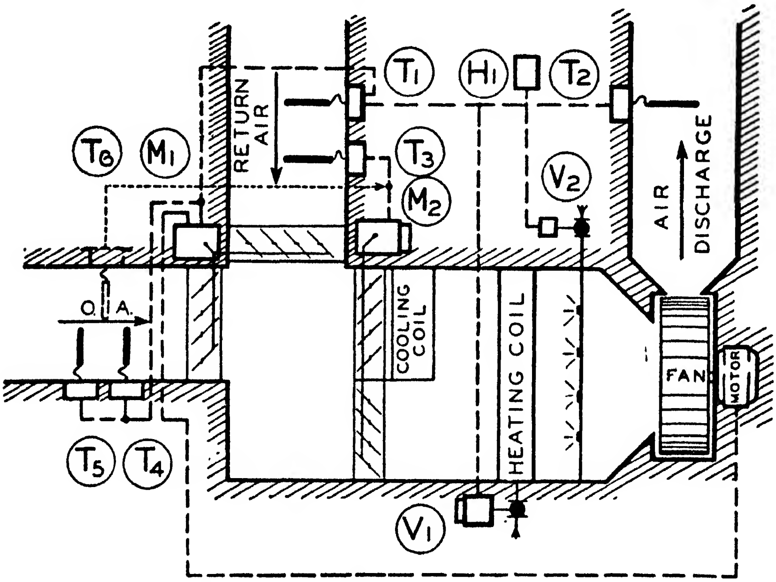


FIG. 10.—All-year air-conditioning system. (*Minneapolis-Honeywell Regulator Co.*)

The dewpoint of the washer is controlled by mixing varying quantities of cold and recirculated water as determined by three-way mixing valve V_1, which is controlled by insertion thermostat T_4.

Alternate No. 1. With the compensated control system insertion thermostat T_3 varies the control point of T_1 according to a predetermined schedule with variations in outdoor temperature.

Figure 10 illustrates an all-year air conditioning system. Mechanical refrigeration is the source of cooling and as face and bypass dampers are used, a low coil temperature can be carried which provides for variation in sensible latent heat ratio and more dehumidification during the cooling cycle. The control system is designed to use outdoor air for winter cooling but at other times only enough for ventilation is taken in. Automatic changeover from cooling to heating cycle is provided.

Operation. *Heating Cycle.* Insertion thermostat T_1 measures the temperature of the return air and positions modulating motorized valve V_1 in accordance with the heat requirements of the conditioned space. Insertion thermostat T_2 measures the temperature of the air being discharged to the room and prevents, by opening motorized valve V_1, the temperature of the air delivered to the room falling so low (such as 65°) that uncomfortable drafts are apt to occur. As the return air temperature rises, T_1 will close the motorized valve V_1 and if the return air temperature continues to rise approximately one degree indicating a need for cooling in the conditioned area, T_1 will shift the control of damper motor M_1 to insertion thermostat T_4, to take a quantity of outdoor air for cooling, which will vary with the outdoor air temperature so as to give a mixed outdoor and return air temperature equal to the setting of T_2. Humidity controller H_1 operates solenoid valve V_2 so as to add moisture to the air when required.

Cooling Cycle. Insertion thermostat T_3 measures the temperature of the return air and modulates damper motor M_2 to position the face and bypass dampers in proportion to the cooling requirements of the room. Insertion thermostat T_5, with its bulb in the outdoor air, positions damper motor M_1 to take a maximum amount of outdoor air on mild days. The use of face and bypass dampers secures low sensible latent ratios at light loads. A control panel contains equipment for varying the minimum outdoor air quantity and closing the outdoor damper on fan shutdown.

Alternate No. 1. If the compensated control system is used, insertion thermostat T_6 will vary the control point of T_3 according to a predetermined schedule, as the outdoor temperature changes.

Figure 11 illustrates a year-round central fan conditioning system.

This system provides temperature control on a cold water coil, and temperature and humidity control on the heating cycle. The changeover between heating and cooling cycles is automatic.

Operation. *Heating Cycle.* Insertion thermostat T_1 measures the temperature of the return air and positions modulating motorized valve V_1 in accordance with the heat requirements of the conditioned space. Insertion thermostat T_2 measures the temperature of the air being discharged to the room and prevents, by opening motorized valve V_1, the temperature of the air delivered to the room falling so low that uncomfortable drafts are apt to occur (such as 65°). As the return air temperature rises, T_1 will close the motorized valve V_1 and if the return air temperature continues to rise approximately 1°, indicating a need for cooling in the conditiond area, T_1 will shift the control of damper motor M_1 to insertion thermostat T_4, to take a quantity of outdoor air for cooling. This quantity will vary with the outdoor air temperature to maintain air entering the heating coil at approximately the setting of T_2. Humidity controller H_2 operates solenoid valve V_3 to add moisture to the air when required.

In the number 1 alternate control system thermostat T_6 varies the control point of H_2 as the outdoor temperature changes according to a predetermined schedule.

Cooling Cycle. Insertion thermostat T_3 located in the return air, controls motorized valve V_2. This control can be modulating or 2-position as desired. Outdoor air damper motor M_1 is controlled by insertion thermostat T_5 to take in a maximum quantity of outdoor air on mild days. In some cases the 2-

position control will provide more dehumidification than will the system using a modulating type valve.

A control panel contains equipment for adjusting the minimum outdoor damper setting and closing the outdoor air damper and valve V_2 on fan shutdown.

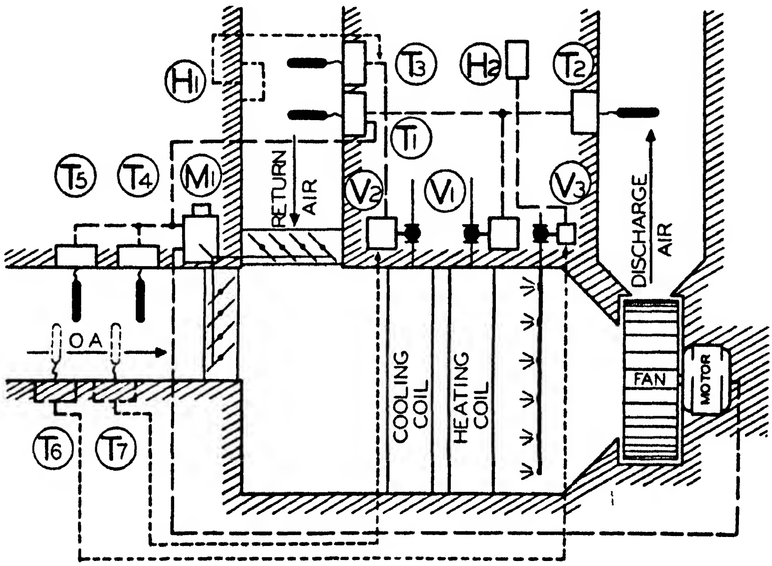


FIG. 11.—A year-round central fan conditioning system. (*Minneapolis-Honeywell Regulator Co.*)

Alternate No. 1. An alternate method is to control by humidity between temperature limits. In this system an insertion thermostat T_3 is used in the return air, together with a modulating type humidity controller H_1. These are so connected that the two controllers will average to maintain desirable conditions. T_3 acts as a low limit and if the temperature of the air in the room falls too low, will modulate valve V_2 toward its closed position.

Alternate No. 2. With the compensated control system insertion thermostat T_1 varies the control point of T_3 as the outdoor temperature changes.

APPENDIX

STEAM TABLES

Table I.—Saturation: Temperatures

Temp. °F, t	Absolute pressure		Specific volume			Enthalpy			Entropy		
	Psi, p	In. Hg, p	Sat liquid, v_f	Evap, v_{fg}	Sat vapor, v_g	Sat liquid, h_f	Evap, h_{fg}	Sat vapor, h_g	Sat liquid, S_f	Evap, S_{fg}	Sat vapor, S_g
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
32	0.08854	0.1803	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877
33	0.09223	0.1878	0.01602	3180	3180	1.01	1075.2	1076.2	0.0020	2.1821	2.1841
34	0.09603	0.1955	0.01602	3061	3061	2.02	1074.7	1076.7	0.0041	2.1764	2.1805
35	0.09995	0.2035	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770
36	0.10401	0.2118	0.01602	2837	2837	4.03	1073.6	1077.6	0.0081	2.1654	2.1735
37	0.10821	0.2203	0.01602	2732	2732	5.04	1073.0	1078.0	0.0102	2.1598	2.1700
38	0.11256	0.2292	0.01602	2632	2632	6.04	1072.4	1078.4	0.0122	2.1544	2.1666
39	0.11705	0.2383	0.01602	2536	2536	7.04	1071.9	1078.9	0.0142	2.1489	2.1631
40	0.12170	0.2478	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597
41	0.12652	0.2576	0.01602	2356	2356	9.05	1070.7	1079.7	0.0182	2.1381	2.1563
42	0.13150	0.2677	0.01602	2271	2271	10.05	1070.1	1080.2	0.0202	2.1327	2.1529
43	0.13665	0.2782	0.01602	2190	2190	11.06	1069.5	1080.6	0.0222	2.1274	2.1496
44	0.14199	0.2891	0.01602	2112	2112	12.06	1068.9	1081.0	0.0242	2.1220	2.1462
45	0.14752	0.3004	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429
46	0.15323	0.3120	0.01602	1964.3	1964.3	14.06	1067.8	1081.9	0.0282	2.1113	2.1395
47	0.15914	0.3240	0.01603	1895.1	1895.1	15.07	1067.3	1082.4	0.0302	2.1060	2.1362
48	0.16525	0.3364	0.01603	1828.6	1828.6	16.07	1066.7	1082.8	0.0321	2.1008	2.1329
49	0.17157	0.3493	0.01603	1764.7	1764.7	17.07	1066.1	1083.2	0.0341	2.0956	2.1297
50	0.17811	0.3626	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264
51	0.18486	0.3764	0.01603	1644.2	1644.2	19.07	1065.0	1084.1	0.0380	2.0852	2.1232
52	0.19182	0.3906	0.01603	1587.6	1587.6	20.07	1064.4	1084.5	0.0400	2.0799	2.1199
53	0.19900	0.4052	0.01603	1533.3	1533.3	21.07	1063.9	1085.0	0.0420	2.0747	2.1167
54	0.20642	0.4203	0.01603	1481.0	1481.0	22.07	1063.3	1085.4	0.0439	2.0697	2.1136
55	0.2141	0.4359	0.01603	1430.7	1430.7	23.07	1062.7	1085.8	0.0459	2.0645	2.1104
56	0.2220	0.4520	0.01603	1382.4	1382.4	24.06	1062.2	1086.3	0.0478	2.0594	2.1072
57	0.2302	0.4686	0.01603	1335.9	1335.9	25.06	1061.6	1086.7	0.0497	2.0544	2.1041
58	0.2386	0.4858	0.01604	1291.1	1291.1	26.06	1061.0	1087.1	0.0517	2.0493	2.1010
59	0.2473	0.5035	0.01604	1248.1	1248.1	27.06	1060.5	1087.6	0.0536	2.0443	2.0979
60	0.2563	0.5218	0.01604	1206.6	1206.7	28.06	1059.9	1088.0	0.0555	2.0393	2.0948
61	0.2655	0.5407	0.01604	1166.8	1166.8	29.06	1059.3	1088.4	0.0574	2.0343	2.0917
62	0.2751	0.5601	0.01604	1128.4	1128.4	30.05	1058.8	1088.9	0.0593	2.0293	2.0886
63	0.2850	0.5802	0.01604	1091.4	1091.4	31.05	1058.2	1089.3	0.0613	2.0243	2.0856
64	0.2951	0.6009	0.01605	1055.7	1055.7	32.05	1057.6	1089.7	0.0632	2.0194	2.0826
65	0.3056	0.6222	0.01605	1021.4	1021.4	33.05	1057.1	1090.2	0.0651	2.0145	0.0796
66	0.3164	0.6442	0.01605	988.4	988.4	34.05	1056.5	1090.6	0.0670	2.0096	2.0766
67	0.3276	0.6669	0.01605	956.6	956.6	35.05	1056.0	1091.0	0.0689	2.0047	2.0736
68	0.3390	0.6903	0.01605	925.9	925.9	36.04	1055.5	1091.5	0.0708	1.9998	2.0706
69	0.3509	0.7144	0.01605	896.3	896.3	37.04	1054.9	1091.9	0.0726	1.9950	2.0676
70	0.3631	0.7392	0.01606	867.8	867.9	38.04	1054.3	1092.3	0.0745	1.9902	2.0647
71	0.3756	0.7648	0.01606	840.4	840.4	39.04	1053.8	1092.8	0.0764	1.9854	2.0618
72	0.3886	0.7912	0.01606	813.9	813.9	40.04	1053.2	1093.2	0.0783	1.9805	2.0588
73	0.4019	0.8183	0.01606	788.3	788.4	41.03	1052.6	1093.6	0.0802	1.9757	2.0559
74	0.4156	0.8462	0.01606	763.7	763.8	42.03	1052.1	1094.1	0.0820	1.9710	2.0530
75	0.4298	0.8750	0.01607	740.0	740.0	43.03	1051.5	1094.5	0.0839	1.9663	2.0502
76	0.4443	0.9046	0.01607	717.1	717.1	44.03	1050.9	1094.9	0.0858	1.9615	2.0473
77	0.4593	0.9352	0.01607	694.9	694.9	45.02	1050.4	1095.4	0.0876	1.9569	2.0445
78	0.4747	0.9666	0.01607	673.6	673.6	46.02	1049.8	1095.8	0.0895	1.9521	2.0416
79	0.4906	0.9989	0.01608	653.0	653.0	47.02	1049.2	1096.2	0.0913	1.9475	2.0388
80	0.5069	1.0321	0.01608	633.1	633.1	48.02	1048.6	1096.6	0.0932	1.9428	2.0360
81	0.5237	1.0664	0.01608	613.9	613.9	49.02	1048.1	1097.1	0.0950	1.9382	2.0332
82	0.5410	1.1016	0.01608	595.3	595.3	50.01	1047.5	1097.5	0.0969	1.9335	2.0304
83	0.5588	1.1378	0.01609	577.4	577.4	51.01	1046.9	1097.9	0.0987	1.9290	2.0277
84	0.5771	1.1750	0.01609	560.1	560.2	52.01	1046.4	1098.4	0.1005	1.9244	2.0249
85	0.5959	1.2133	0.01609	543.4	543.5	53.00	1045.8	1098.8	0.1024	1.9198	2.0222
86	0.6152	1.2527	0.01609	527.3	526.3	54.00	1045.2	1099.2	0.1042	1.9153	2.0195
87	0.6351	1.2931	0.01610	511.7	511.7	55.00	1044.7	1099.7	0.1060	1.9108	2.0168
88	0.6556	1.3347	0.01610	496.6	496.7	56.00	1044.1	1100.1	0.1079	1.9062	2.0141
89	0.6766	1.3775	0.01610	482.1	482.1	56.99	1043.5	1100.5	0.1097	1.9017	2.0114

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TABLE I.—SATURATION: TEMPERATURES (*Cont.*)

Temp, °F, t (1)	Absolute pressure: Psi, p (2)	Absolute pressure: In. Hg, p (3)	Specific volume: Sat liquid, v_f (4)	Specific volume: Evap, v_{fg} (5)	Specific volume: Sat vapor, v_g (6)	Enthalpy: Sat liquid, h_f (7)	Enthalpy: Evap, h_{fg} (8)	Enthalpy: Sat vapor, h_g (9)	Entropy: Sat liquid, S_f (10)	Entropy: Evap, S_{fg} (11)	Entropy: Sat vapor, S_g (12)
90	0.6982	1.4215	0.01610	468.0	468.0	57.99	1042.9	1100.9	0.1115	1.8972	2.0087
91	0.7204	1.4667	0.01611	454.4	454.4	58.99	1042.4	1101.4	0.1133	1.8927	2.0060
92	0.7432	1.5131	0.01611	441.2	441.3	59.99	1041.8	1101.8	0.1151	1.8883	2.0034
93	0.7666	1.5608	0.01611	428.5	428.5	60.98	1041.2	1102.2	0.1169	1.8838	2.0007
94	0.7906	1.6097	0.01612	416.2	416.2	61.98	1040.7	1102.6	0.1187	1.8794	1.9981
95	0.8153	1.6600	0.01612	404.3	404.3	62.98	1040.1	1103.1	0.1205	1.8750	1.9955
96	0.8407	1.7117	0.01612	392.8	392 8	63.98	1039.5	1103.5	0.1223	1.8706	1.9929
97	0.8668	1.7647	0.01612	381.7	381.7	64.97	1038.9	1103.9	0.1241	1.8662	1.9903
98	0.8935	1.8192	0.01613	370.9	370.9	65.97	1038.4	1104.4	0.1259	1.8618	1.9877
99	0.9210	1.8751	0.01613	360.4	360.5	66.97	1037.8	1104.8	0.1277	1.8575	1.9852
100	0.9492	1.9325	0.01613	350.3	350.4	67.97	1037.2	1105.2	0.1295	1.8531	1.9826
101	0.9781	1.9915	0.01614	340.6	340.6	68.96	1036.6	1105.6	0.1313	1.8488	1.9801
102	1.0078	2.0519	0.01614	331.1	331.1	69.96	1036.1	1106.1	0.1330	1.8445	1.9775
103	1.0382	2.1138	0.01614	321.9	321.9	70.96	1035.5	1106.5	0.1348	1.8402	1.9750
104	1.0695	2.1775	0.01615	313.1	313.1	71.96	1034.9	1106.9	0.1366	1.8359	1.9725
105	1.1016	2.2429	0.01615	304.5	304.5	72.95	1034.3	1107.3	0.1383	1.8317	1.9700
106	1.1345	2.3099	0.01615	296.1	296.2	73.95	1033.8	1107.8	0.1401	1.8274	1.9675
107	1.1683	2.3786	0.01616	288.1	288.1	74.95	1033.3	1108.2	0.1419	1.8232	1.9651
108	1.2029	2.4491	0.01616	280.3	280.3	75.95	1032.7	1108.6	0.1436	1.8190	1.9626
109	1.2384	2.5214	0.01616	272.7	272.7	76.94	1032.1	1109.0	0.1454	1.8147	1.9601
110	1.2748	2.5955	0.01617	265.3	265.4	77.94	1031.6	1109.5	0.1471	1.8106	1.9577
111	1.3121	2.6715	0.01617	258.2	258.3	78.94	1031.0	1109.9	0.1489	1.8064	1.9553
112	1.3504	2.7494	0.01617	251.3	251.4	79.94	1030.4	1110.3	0.1506	1.8023	1.9529
113	1.3896	2.8293	0.01618	244.6	244.7	80.94	1029.8	1110.7	0.1524	1.7981	1.9505
114	1.4298	2.9111	0.01618	238.2	238.2	81.93	1029.2	1111.1	0.1541	1.7940	1.9481
115	1.4709	2.9948	0.01618	231.9	231.9	82.93	1028.7	1111.6	0.1559	1.7898	1.9457
116	1.5130	3.0806	0.01619	225.8	225.8	83.93	1028.1	1112.0	0.1576	1.7857	1.9433
117	1.5563	3.1687	0.01619	219.9	219.9	84.93	1027.5	1112.4	0.1593	1.7816	1.9409
118	1.6006	3.2589	0.01620	214.2	214.2	85.92	1026.9	1112.8	0.1610	1.7776	1.9386
119	1.6459	3.3512	0.01620	208.6	208.7	86.92	1026.3	1113.2	0.1628	1.7735	1.9363
120	1.6924	3.4458	0.01620	203.25	203.27	87.92	1025.8	1113.7	0.1645	1.7694	1.9339
121	1.7400	3.5427	0.01621	198.02	198.03	88.92	1025.2	1114.1	0.1662	1.7654	1.9316
122	1.7888	3.6420	0.01621	192.93	192.95	89.92	1024.6	1114.5	0.1679	1.7614	1.9293
123	1.8387	3.7436	0.01622	188.01	188.02	90.91	1024.0	1114.9	0.1696	1.7574	1.9270
124	1.8897	3.8475	0.01622	183.23	183.25	91.91	1023.4	1115.3	0.1714	1.7533	1.9247
125	1.9420	3.9539	0.01622	178.59	178.61	92.91	1022.9	1115.8	0.1731	1.7493	1.9334
126	1.9955	4.0629	0.01623	174.09	174.10	93.91	1022.3	1116.2	0.1748	1.7454	1.9202
127	2.0503	4.1745	0.01623	169.71	169.72	94.91	1021.7	1116.6	0.1765	1.7414	1.9179
128	2.1064	4.2887	0.01624	165.46	165.47	95.91	1021.1	1117.0	0.1782	1.7374	1.9156
129	2.1638	4.4055	0.01624	161.33	161.35	96.90	1020.5	1117.4	0.1799	1.7335	1.9134
130	2.2225	4.5251	0.01625	157.32	157.34	97.90	1020.0	1117.9	0.1816	1.7296	1.9112
131	2.2826	4.6474	0.01625	153.43	153.44	98.90	1019.4	1118.3	0.1833	1.7257	1.9090
132	2.3440	4.7725	0.01626	149.65	149.66	99.90	1018.8	1118.7	0.1849	1.7218	1.9067
133	2.4069	4.9005	0.01626	145.97	145.99	100.90	1018.2	1119.1	0.1866	1.7179	1.9045
134	2.4712	5.0314	0.01626	142.40	142.42	101.90	1017.6	1119.5	0.1883	1.7141	1.9023
135	2.5370	5.1653	0.01627	138.93	138.95	102.90	1017.0	1119.9	0.1900	1.7102	1.9002
136	2.6042	5.3022	0.01627	135.56	135.58	103.90	1016.4	1120.3	0.1917	1.7063	1.8980
137	2.6729	5.4421	0.01628	132.29	132.30	104.89	1015.9	1120.8	0.1934	1.7024	1.8958
138	2.7432	5.5852	0.01628	129.10	129.12	105.89	1015.3	1121.2	0.1950	1.6987	1.8937
139	2.8151	5.7316	0.01629	126.00	126.02	106.89	1014.7	1121.6	0.1967	1.6948	1.8915
140	2.8886	5.8812	0.01629	122.99	123.01	107.89	1014.1	1122.0	0.1984	1.6910	1.8894
141	2.9637	6.0341	0.01630	120.06	120.08	108.89	1013.5	1122.4	0.2000	1.6873	1.8873
142	3.0404	6.1903	0.01630	117.22	117.23	109.89	1012.9	1122.8	0.2016	1.6835	1.8851
143	3.1188	6.3500	0.01631	114.45	114.46	110.89	1012.3	1123.2	0.2033	1.6797	1.8830
144	3.1990	6.5132	0.01631	111.75	111.77	111.89	1011.7	1123.6	0.2049	1.6760	1.8809

TABLE I.—SATURATION: TEMPERATURES (*Cont.*)

Temp, °F, t	Absolute pressure		Specific volume			Enthalpy			Entropy		
	Psi, p	In. Hg, p	Sat liquid, v_f	Evap, v_{fg}	Sat vapor, v_g	Sat liquid, h_f	Evap, h_{fg}	Sat vapor, h_g	Sat liquid, S_f	Evap, S_{fg}	Sat vapor, S_g
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
145	3.281	6.680	0.01632	109.13	109.15	112.89	1011.2	1124.1	0.2066	1.6722	1.8788
146	3.365	6.850	0.01632	106.58	106.60	113.89	1010.6	1124.5	0.2083	1.6685	1.8768
147	3.450	7.024	0.01633	104.10	104.12	114.89	1010.0	1124.9	0.2099	1.6648	1.8747
148	3.537	7.202	0.01633	101.69	101.71	115.89	1009.4	1125.3	0.2116	1.6610	1.8726
149	3.627	7.384	0.01634	99.34	99.36	116.89	1008.8	1125.7	0.2133	1.6573	1.8706
150	3.718	7.569	0.01634	97.06	97.07	117.89	1008.2	1126.1	0.2149	1.6537	1.8685
151	3.811	7.759	0.01635	94.83	94.85	118.89	1007.6	1126.5	0.2165	1.6500	1.8665
152	3.906	7.952	0.01635	92.67	92.68	119.89	1007.0	1126.9	0.2182	1.6463	1.8645
153	4.003	8.150	0.01636	90.56	90.57	120.89	1006.4	1127.3	0.2198	1.6427	1.8624
154	4.102	8.351	0.01636	88.51	88.52	121.89	1005.8	1127.7	0.2214	1.6390	1.8604
155	4.203	8.557	0.01637	86.51	86.52	122.89	1005.2	1128.1	0.2230	1.6354	1.8584
156	4.306	8.767	0.01637	84.56	84.58	123.89	1004.7	1128.6	0.2246	1.6318	1.8564
157	4.411	8.981	0.01638	82.67	82.69	124.89	1004.1	1129.0	0.2263	1.6282	1.8545
158	4.519	9.200	0.01638	80.82	80.84	125.89	1003.5	1129.4	0.2279	1.6246	1.8525
159	4.629	9.424	0.01639	79.03	79.04	126.89	1002.9	1129.8	0.2295	1.6210	1.8505
160	4.741	9.652	0.01639	77.27	77.29	127.89	1002.3	1130.2	0.2311	1.6174	1.8485
161	4.855	9.885	0.01640	75.57	75.58	128.89	1001.7	1130.6	0.2327	1.6138	1.8466
162	4.971	10.122	0.01640	73.91	73.92	129.89	1001.1	1131.0	0.2343	1.6103	1.8446
163	5.090	10.364	0.01641	72.29	72.30	130.89	1000.5	1131.4	0.2360	1.6067	1.8427
164	5.212	10.611	0.01641	70.71	70.73	131.89	999.9	1131.8	0.2376	1.6032	1.8408
165	5.335	10.863	0.01642	69.17	69.19	132.89	999.3	1132.2	0.2392	1.5997	1.8388
166	5.461	11.120	0.01643	67.67	67.69	133.89	998.7	1132.6	0.2408	1.5961	1.8369
167	5.590	11.382	0.01643	66.21	66.23	134.89	998.1	1133.0	0.2424	1.5926	1.8350
168	5.721	11.649	0.01644	64.79	64.80	135.90	997.5	1133.4	0.2440	1.5891	1.8331
169	5.855	11.921	0.02644	63.40	63.41	136.90	996.9	1133.8	0.2455	1.5857	1.8312
170	5.992	12.199	0.01645	62.04	62.06	137.90	996.3	1134.2	0.2472	1.5822	1.8293
171	6.131	12.483	0.01645	60.72	60.74	138.90	995.7	1134.6	0.2488	1.5787	1.8275
172	6.273	12.772	0.01646	59.43	59.45	139.90	995.1	1135.0	0.2503	1.5753	1.8256
173	6.417	13.066	0.01647	58.18	58.20	140.90	994.5	1135.4	0.2519	1.5718	1.8237
174	6.565	13.366	0.01647	56.96	56.97	141.90	993.9	1135.8	0.2535	1.5684	1.8219
175	6.715	13.671	0.02648	55.76	55.78	142.91	993.3	1136.2	0.2551	1.5649	1.8200
176	6.868	13.983	0.01648	54.60	54.61	143.91	992.7	1136.6	0.2567	1.5615	1.8182
177	7.024	14.301	0.01649	53.46	53.48	144.91	992.1	1137.0	0.2583	1.5581	1.8164
178	7.183	14.625	0.01650	52.35	52.37	145.91	991.5	1137.4	0.2599	1.5547	1.8146
179	7.345	14.955	0.01650	51.27	51.29	146.92	990.8	1137.7	0.2614	1.5513	1.8127
180	7.510	15.291	0.01651	50.21	50.23	147.92	990.2	1138.1	0.2630	1.5480	1.8109
181	7.678	15.633	0.01651	49.18	49.20	148.92	989.6	1138.5	0.2645	1.5446	1.8091
182	7.850	15.982	0.01652	48.18	48.19	149.92	989.0	1138.9	0.2661	1.5412	1.8073
183	8.024	16.337	0.01653	47.19	47.21	150.93	988.4	1139.3	0.2676	1.5379	1.8055
184	8.202	16.699	0.01653	46.24	46.25	151.93	987.8	1139.7	0.2692	1.5346	1.8038
185	8.383	17.068	0.01654	45.29	45.31	152.93	987.2	1140.1	0.2708	1.5312	1.8020
186	8.567	17.443	0.01654	44.39	44.40	153.94	986.6	1140.5	0.2723	1.5279	1.8002
187	8.755	17.825	0.01655	43.50	43.51	154.94	986.0	1140.9	0.2739	1.5246	1.7985
188	8.946	18.214	0.01656	42.62	42.64	155.94	985.4	1141.3	0.2754	1.5213	1.7967
189	9.141	18.611	0.01656	41.77	41.79	156.95	984.8	1141.7	0.2770	1.5180	1.7950
190	9.339	19.014	0.01657	40.94	40.96	157.95	984.1	1142.0	0.2785	1.5147	1.7932
191	9.541	19.425	0.01658	40.13	40.15	158.95	983.4	1142.4	0.2801	1.5114	1.7915
192	9.746	19.843	0.01658	39.34	39.36	159.96	982.8	1142.8	0.2816	1.5082	1.7898
193	9.955	20.269	0.01659	38.57	38.58	160.96	982.2	1143.2	0.2831	1.5049	1.7880
194	10.168	20.703	0.01659	37.81	37.83	161.97	981.6	1143.6	0.2846	1.5017	1.7863
195	10.385	21.144	0.01660	37.07	37.09	162.97	981.0	1144.0	0.2862	1.4984	1.7846
196	10.605	21.593	0.01661	36.35	36.37	163.97	980.4	1144.4	0.2877	1.4952	1.7829
197	10.830	22.050	0.01661	35.64	35.66	164.98	979.7	1144.7	0.2892	1.4920	1.7812
198	11.058	22.515	0.01662	34.95	34.97	165.98	979.1	1145.1	0.2907	1.4888	1.7795
199	11.290	22.987	0.01663	34.28	34.30	166.99	978.5	1145.5	0.2923	1.4856	1.7779

TABLE I.—SATURATION: TEMPERATURES (*Concl.*)

Temp, °F, t (1)	Absolute pressure		Specific volume			Enthalpy			Entropy		
	Psi, p (2)	In. Hg, p (3)	Sat liquid, v_f (4)	Evap, v_{fg} (5)	Sat vapor, v_g (6)	Sat liquid, h_f (7)	Evap, h_{fg} (8)	Sat vapor, h_g (9)	Sat liquid, S_f (10)	Evap, S_{fg} (11)	Sat vapor S_g (12)
200	11.526	23.467	0.01663	33.62	33.64	167.99	977.9	1145.9	0.2938	1.4824	1.7762
202	12.011	24.455	0.01665	32.35	32.37	170.00	976.6	1146.6	0.2969	1.4760	1.7729
204	12.512	25.475	0.01666	31.14	31.15	172.02	975.4	1147.4	0.2999	1.4697	1.7696
206	13.031	26.531	0.01667	29.97	29.99	174.03	974.2	1148.2	0.3029	1.4634	1.7663
208	13.568	27.625	0.01669	28.86	28.88	176.04	972.9	1148.9	0.3059	1.4571	1.7630
210	14.123	28.755	0.01670	27.80	27.82	178.05	971.6	1149.7	0.3090	1.4508	1.7598
212	14.696	29.922	0.01672	26.78	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566
214	15.289	31.129	0.01673	25.81	25.83	182.08	969.0	1151.1	0.3149	1.4385	1.7534
216	15.901	32.375	0.01674	24.88	24.90	184.10	967.8	1151.9	0.3179	1.4323	1.7502
218	16.533	33.662	0.01676	23.99	24.01	186.11	966.5	1152.6	0.3209	1.4262	1.7471
220	17.186	34.992	0.01677	23.13	23.15	188.13	965.2	1153.4	0.3239	1.4201	1.7440
222	17.861	36.355	0.01679	22.31	22.33	190.15	963.9	1154.1	0.3268	1.4141	1.7409
224	18.557	37.782	0.01680	21.53	21.55	192.17	962.6	1154.8	0.3298	1.4080	1.7378
226	19.275	39.244	0.01682	20.78	20.79	194.18	961.3	1155.5	0.3328	1.4020	1.7348
228	20.016	40.753	0.01683	20.06	20.07	196.20	960.1	1156.3	0.3357	1.3961	1.7318
230	20.780	42.308	0.01684	19.365	19.382	198.23	958.8	1157.0	0.3387	1.3901	1.7288
240	24.969	50.837	0.01692	16.306	16.323	208.34	952.2	1160.5	0.3531	1.3609	1.7140
250	29.825	60.725	0.01700	13.804	13.821	218.48	945.5	1164.0	0.3675	1.3323	1.6998
260	35.429	72.134	0.01709	11.746	11.763	228.64	938.7	1167.3	0.3817	1.3043	1.6860
270	41.858	85.225	0.01717	10.044	10.061	238.84	931.8	1170.6	0.3958	1.2769	1.6727
280	49.203	100.18	0.01726	8.628	8.645	249.06	924.7	1173.8	0.4096	1.2501	1.6597
290	57.556	117.19	0.01735	7.444	7.461	259.31	917.5	1176.8	0.4234	1.2238	1.6472
300	67.013	136.44	0.01745	6.449	6.466	269.59	910.1	1179.7	0.4369	1.1980	1.6350
310	77.68	.....	0.01755	5.600	5.626	279.92	902.6	1182.5	0.4504	1.1727	1.6231
320	89.66	.....	0.01765	4.896	4.914	290.28	894.9	1185.2	0.4637	1.1478	1.6115
330	103.06	.....	0.01776	4.289	4.307	300.68	887.0	1187.7	0.4769	1.1233	1.6002
340	118.01	.....	0.01787	3.770	3.788	311.13	879.0	1190.1	0.4900	1.0992	1.5891
350	134.63	.....	0.01799	3.324	3.342	321.63	870.7	1192.3	0.5029	1.0754	1.5783
360	153.04	.....	0.01811	2.939	2.957	332.18	862.2	1194.4	0.5158	1.0519	1.5677
370	173.37	.....	0.01823	2.606	2.625	342.79	853.5	1196.3	0.5286	1.0287	1.5573
380	195.77	.....	0.01836	2.317	2.335	353.45	844.6	1198.1	0.5413	1.0059	1.5471
390	220.37	.....	0.01850	2.0651	2.0836	364.17	835.4	1199.6	0.5539	0.9832	1.5371
400	247.31	.....	0.01864	1.8447	1.8633	374.97	826.0	1201.0	0.5664	0.9608	1.5272
425	325.92	.....	0.01902	1.4036	1.4226	402.27	801.2	1203.5	0.5974	0.9056	1.5030
450	422.6	.....	0.0194	1.0799	0.0993	430.1	774.5	1204.6	0.6280	0.8513	1.4793
475	539.9	.....	0.0199	0.8380	0.8579	458.6	745.4	1204.0	0.6584	0.7976	1.4560
500	680.8	.....	0.0204	0.6545	0.6749	487.8	713.9	1201.7	0.6887	0.7438	1.4325
525	848.1	.....	0.0210	0.5130	0.5340	518.0	679.1	1197.1	0.7191	0.6897	1.4088
550	1045.2	.....	0.0218	0.4022	0.4240	549.3	640.8	1190.0	0.7497	0.6346	1.3843
575	1275.4	.....	0.0226	0.3143	0.3369	582.1	597.7	1179.8	0.7809	0.5777	1.3585
600	1542.9	.....	0.0236	0.2432	0.2668	617.0	548.5	1165.5	0.8131	0.5176	1.3307
625	1852.0	.....	0.0250	0.1845	0.2095	654.4	491.4	1145.8	0.8467	0.4530	1.2997
650	2208.2	.....	0.0268	0.1348	0.1616	695.7	422.8	1118.5	0.8828	0.3809	1.2637
675	2618.7	.....	0.0297	0.0899	0.1196	745.4	332.6	1078.0	0.9251	0.2931	1.2183
700	3093.7	.....	0.0369	0.0392	0.0761	823.3	172.1	995.4	0.9905	0.1484	1.1389
702	3134.9	.....	0.0385	0.0325	0.0710	835.4	145.2	980.6	1.0006	0.1249	1.1256
704	3176.7	.....	0.0410	0.0234	0.0645	852.7	106.0	958.7	1.0152	0.0911	1.1063
705	3197.7	.....	0.0438	0.0152	0.0589	869.2	69.1	938.4	1.0293	0.0593	1.0886
705.4	3206.2	.....	0.0503	0	0.0503	902.7	0	902.7	1.0580	0	1.0580

TABLE II.—PROPERTIES OF SATURATED STEAM: PRESSURE TABLE

Abs Pressure In. Hg, p	Temp, °F, t	Specific volume		Enthalpy			Entropy		
		Sat liquid, v_f	Sat vapor, v_g	Sat liquid, h_f	Evap, h_{fg}	Sat vapor, h_g	Sat liquid, S_f	Evap, S_{fg}	Sat vapor, S_g
0.25	40.23	0.01602	2423.7	8.28	1071.1	1079.4	0.0166	2.1423	2.1589
0.50	58.80	0.01604	1256.4	26.86	1060.6	1087.5	0.0532	2.0453	2.0985
0.75	70.43	0.01606	856.1	38.47	1054.0	1092.5	0.0754	1.9881	2.0635
1.00	79.03	0.01608	652.3	47.05	1049.2	1096.3	0.0914	1.9473	2.0387
1.5	91.72	0.01611	444.9	59.71	1042.0	1101.7	0.1147	1.8894	2.0041
2	101.14	0.01614	339.2	69.10	1036.6	1105.7	0.1316	1.8481	1.9797
4	125.43	0.01622	176.7	93.34	1022.7	1116.0	0.1738	1.7476	1.9214
6	140.78	0.01630	120.72	108.67	1013.6	1122.3	0.1996	1.6881	1.8877
8	152.24	0.01635	92.16	120.13	1006.9	1127.0	0.2186	1.6454	1.8640
10	161.49	0.01640	74.76	129.38	1001.4	1130.8	0.2335	1.6121	1.8456
12	169.28	0.01644	63.03	137.18	996.7	1133.9	0.2460	1.5847	1.8307
14	176.05	0.01648	54.55	143.96	992.6	1136.6	0.2568	1.5613	1.8181
16	182.05	0.01652	48.14	149.98	988.9	1138.9	0.2662	1.5410	1.8072
18	187.45	0.01655	43.11	155.39	985.7	1141.1	0.2746	1.5231	1.7977
20	192.37	0.01658	39.07	160.33	982.7	1143.0	0.2822	1.5069	1.7891
22	196.90	0.01661	35.73	164.87	979.8	1144.7	0.2891	1.4923	1.7814
24	201.09	0.01664	32.94	169.09	977.2	1146.3	0.2955	1.4789	1.7744
26	205.00	0.01667	30.56	173.02	974.8	1147.8	0.3014	1.4665	1.7679
28	208.67	0.01669	28.52	176.72	972.5	1149.2	0.3069	1.4550	1.7619
30	212.13	0.01672	26.74	180.19	970.3	1150.5	0.3122	1.4442	1.7564
Psi									
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566
16	216.32	0.01674	24.75	184.42	967.6	1152.0	0.3184	1.4313	1.7497
18	222.41	0.01679	22.17	190.56	963.6	1154.2	0.3275	1.4128	1.7403
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319
22	233.07	0.01687	18.375	201.33	956.8	1158.1	0.3431	1.3811	1.7242
24	237.82	0.01691	16.938	206.14	953.7	1159.8	0.3500	1.3672	1.7172
26	242.25	0.01694	15.715	210.62	950.7	1161.3	0.3564	1.3544	1.7108
28	246.41	0.01698	14.663	214.83	947.9	1162.7	0.3623	1.3425	1.7048
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993
32	254.05	0.01704	12.940	222.59	942.8	1165.4	0.3733	1.3209	1.6941
34	257.58	0.01707	12.226	226.18	940.3	1166.5	0.3783	1.3110	1.6893
36	260.95	0.01709	11.588	229.60	938.0	1167.6	0.3831	1.3017	1.6848
38	264.16	0.01712	11.015	232.89	935.8	1168.7	0.3876	1.2929	1.6805
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763
42	270.21	0.01717	10.029	239.04	931.6	1170.7	0.3960	1.2764	1.6724
44	273.05	0.01720	9.601	241.95	929.6	1171.6	0.4000	1.2687	1.6687
46	275.80	0.01722	9.209	244.75	927.7	1172.4	0.4038	1.2613	1.6652
48	278.45	0.01725	8.848	247.47	925.8	1173.3	0.4075	1.2542	1.6617
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585
52	283.49	0.01729	8.208	252.63	922.2	1174.8	0.4144	1.2409	1.6553
54	285.90	0.01731	7.922	255.09	920.5	1175.6	0.4177	1.2346	1.6523
56	288.23	0.01733	7.656	257.50	918.8	1176.3	0.4209	1.2285	1.6494
58	290.50	0.01736	7.407	259.82	917.1	1176.9	0.4240	1.2226	1.6466
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438
62	294.85	0.01740	6.957	264.30	913.9	1178.2	0.4300	1.2112	1.6412
64	296.94	0.01742	6.752	266.45	912.3	1178.8	0.4328	1.2059	1.6387
66	298.99	0.01744	6.560	268.55	910.8	1179.4	0.4356	1.2006	1.6362
68	300.98	0.01746	6.378	270.60	909.4	1180.0	0.4383	1.1955	1.6338
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315
72	304.83	0.01750	6.044	274.57	906.5	1181.1	0.4435	1.1857	1.6292
74	306.68	0.01752	5.890	276.49	905.1	1181.6	0.4460	1.1810	1.6270
76	308.50	0.01754	5.743	278.37	903.7	1182.1	0.4484	1.1764	1.6248
78	310.29	0.01755	5.604	280.21	902.4	1182.6	0.4508	1.1720	1.6228
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207
82	313.74	0.01759	5.346	283.79	899.7	1183.5	0.4554	1.1633	1.6187
84	315.42	0.01761	5.226	285.53	898.5	1184.0	0.4576	1.1592	1.6168
86	317.07	0.01762	5.111	287.24	897.2	1184.4	0.4598	1.1551	1.6149
88	318.68	0.01764	5.001	288.91	895.9	1184.8	0.4620	1.1510	1.6130
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112
92	321.83	0.01768	4.796	292.18	893.5	1185.7	0.4661	1.1433	1.6094
94	323.36	0.01769	4.699	293.78	892.3	1186.1	0.4682	1.1394	1.6076
96	324.87	0.01771	4.606	295.34	891.1	1186.4	0.4702	1.1358	1.6060
98	326.35	0.01772	4.517	296.89	889.9	1186.8	0.4721	1.1322	1.6043

TABLE II.—PROPERTIES OF SATURATED STEAM: PRESSURE TABLE (*Cont.*)

Abs Pressure, psi, p	Temp, °F, t	Specific volume		Enthalpy			Entropy		
		Sat liquid, v_f	Sat vapor, v_g	Sat liquid, h_f	Evap, h_{fg}	Sat vapor, h_g	Sat liquid, s_f	Evap, s_{fg}	Sat vapor, s_g
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026
102	329.25	0.01775	4.350	299.90	887.6	1187.5	0.4759	1.1251	1.6010
104	330.66	0.01777	4.271	301.37	886.5	1187.9	0.4778	1.1216	1.5994
106	332.05	0.01778	4.194	302.82	885.4	1188.2	0.4796	1.1182	1.5978
108	333.42	0.01780	4.120	304.26	884.3	1188.6	0.4814	1.1149	1.5963
110	334.77	0.01782	4.049	305.66	883.2	1188.9	0.4832	1.1117	1.5948
112	336.11	0.01783	3.981	307.06	882.1	1189.2	0.4849	1.1085	1.5934
114	337.42	0.01784	3.914	308.43	881.1	1189.5	0.4866	1.1053	1.5919
116	338.72	0.01786	3.850	309.79	880.0	1189.8	0.4883	1.1022	1.5905
118	339.99	0.01787	3.788	311.12	879.0	1190.1	0.4900	1.0992	1.5891
120	341.25	0.01789	3.728	312.44	877.9	1190.4	0.4916	1.0962	1.5878
122	342.50	0.01791	3.670	313.75	876.9	1190.7	0.4932	1.0933	1.5865
124	343.72	0.01792	3.614	315.04	875.9	1190.9	0.4948	1.0903	1.5851
126	344.94	0.01793	3.560	316.31	874.9	1191.2	0.4964	1.0874	1.5838
128	346.13	0.01794	3.507	317.57	873.9	1191.5	0.4980	1.0845	1.5825
130	347.32	0.01796	3.455	318.81	872.9	1191.7	0.4995	1.0817	1.5812
132	348.48	0.01797	3.405	320.04	872.0	1192.0	0.5010	1.0790	1.5800
134	349.64	0.01799	3.357	321.25	871.0	1192.2	0.5025	1.0762	1.5787
136	350.78	0.01800	3.310	322.45	870.1	1192.5	0.5040	1.0735	1.5775
138	351.91	0.01801	3.264	323.64	869.1	1192.7	0.5054	1.0709	1.5763
140	353.02	0.01802	3.220	324.82	868.2	1193.0	0.5069	1.0682	1.5751
142	354.12	0.01804	3.177	325.98	867.2	1193.2	0.5083	1.0657	1.5740
144	355.21	0.01805	3.134	327.13	866.3	1193.4	0.5097	1.0631	1.5728
146	356.29	0.01806	3.094	328.27	865.3	1193.6	0.5111	1.0605	1.5716
148	357.36	0.01808	3.054	329.39	864.5	1193.9	0.5124	1.0580	1.5705
150	358.42	0.01809	3.015	330.51	863.6	1194.1	0.5138	1.0556	1.5694
152	359.46	0.01810	2.977	331.61	862.7	1194.3	0.5151	1.0532	1.5683
154	360.49	0.01812	2.940	332.70	861.8	1194.5	0.5165	1.0507	1.5672
156	361.52	0.01813	2.904	333.79	860.9	1194.7	0.5178	1.0483	1.5661
158	362.52	0.01814	2.869	334.86	860.0	1194.9	0.5191	1.0459	1.5650
160	363.53	0.01815	2.834	335.93	859.2	1195.1	0.5204	1.0436	1.5640
162	364.53	0.01817	2.801	336.98	858.3	1195.3	0.5216	1.0414	1.5630
164	365.51	0.01818	2.768	338.02	857.5	1195.5	0.5229	1.0391	1.5620
166	366.48	0.01819	2.736	339.05	856.6	1195.7	0.5241	1.0369	1.5610
168	367.45	0.01820	2.705	340.07	855.7	1195.8	0.5254	1.0346	1.5600
170	368.41	0.01822	2.675	341.09	854.9	1196.0	0.5266	1.0324	1.5590
172	369.35	0.01823	2.645	342.10	854.1	1196.2	0.5278	1.0302	1.5580
174	370.29	0.01824	2.616	343.10	853.3	1196.4	0.5290	1.0280	1.5570
176	371.22	0.01825	2.587	344.09	852.4	1196.5	0.5302	1.0259	1.5561
178	372.14	0.01826	2.559	345.06	851.6	1196.7	0.5313	1.0238	1.5551
180	373.06	0.01827	2.532	346.03	850.8	1196.9	0.5325	1.0217	1.5542
182	373.96	0.01829	2.505	347.00	850.0	1197.0	0.5336	1.0196	1.5532
184	374.86	0.01830	2.479	347.96	849.2	1197.2	0.5348	1.0175	1.5523
186	375.75	0.01831	2.454	348.92	848.4	1197.3	0.5359	1.0155	1.5514
188	376.64	0.01832	2.429	349.83	847.6	1197.5	0.5370	1.0135	1.5506
190	377.51	0.01833	2.404	350.79	846.8	1197.6	0.5381	1.0116	1.5497
192	378.38	0.01834	2.380	351.72	846.1	1197.8	0.5392	1.0096	1.5488
194	379.24	0.02835	2.356	352.64	845.3	1197.9	0.5403	1.0076	1.5479
196	380.10	0.01836	2.333	353.55	844.5	1198.1	0.5414	1.0056	1.5470
198	380.95	0.01838	2.310	354.46	843.7	1198.2	0.5425	1.0037	1.5462
200	381.79	0.02839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453
205	383.86	0.01842	2.234	357.58	841.1	1198.7	0.5461	0.9971	1.5432
210	385.90	0.01844	2.183	359.77	839.2	1199.0	0.5487	0.9925	1.5412
215	387.89	0.01847	2.134	361.91	837.4	1199.3	0.5512	0.9880	1.5392
220	389.86	0.01850	2.087	364.02	835.6	1199.6	0.5537	0.9835	1.5372
225	391.79	0.01852	2.0422	366.09	833.8	1199.9	0.5561	0.9792	1.5353
230	393.68	0.01854	1.9992	368.13	832.0	1200.1	0.5585	0.9750	1.5334
235	395.54	0.01857	1.9579	370.14	830.3	1200.4	0.5608	0.9708	1.5316
240	397.37	0.01860	1.9183	372.12	828.5	1200.6	0.5631	0.9667	1.5298
245	399.18	0.01863	1.8803	374.08	826.8	1200.9	0.5653	0.9627	1.5280

Table II.—Properties of Saturated Steam: Pressure Table (*Concl.*)

Abs Pressure, psi, p	Temp, °F, t	Specific volume		Enthalpy			Entropy		
		Sat liquid, v_f	Sat vapor, v_g	Sat liquid, h_f	Evap, h_{fg}	Sat vapor, h_g	Sat liquid, S_f	Evap, S_{fg}	Sat vapor, S_g
250	400.95	0.01865	1.8438	376.00	825.1	1201.1	0.5675	0.9588	1.5263
260	404.42	0.01870	1.7748	379.76	821.8	1201.5	0.5719	0.9510	1.5229
270	407.78	0.01875	1.7107	383.42	818.5	1201.9	0.5760	0.9436	1.5196
280	411.05	0.01880	1.6511	386.98	815.3	1202.3	0.5801	0.9363	1.5164
290	414.23	0.01885	1.5954	390.46	812.1	1202.6	0.5841	0.9292	1.5133
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104
320	423.29	0.01899	1.4485	400.39	803.0	1203.4	0.5952	0.9094	1.5046
340	428.97	0.01908	1.3645	406.66	797.1	1203.7	0.6022	0.8970	1.4992
360	434.40	0.01917	1.2895	412.67	791.4	1204.1	0.6090	0.8851	1.4941
380	439.60	0.01925	1.2222	418.45	785.8	1204.3	0.6153	0.8738	1.4891
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844
420	449.39	0.0194	1.1061	429.4	775.2	1204.6	0.6272	0.8527	1.4799
440	454.02	0.0195	1.0556	434.6	770.0	1204.6	0.6329	0.8426	1.4755
460	458.50	0.0196	1.0094	439.7	764.9	1204.6	0.6383	0.8330	1.4713
480	462.82	0.0197	0.9670	444.6	759.9	1204.5	0.6436	0.8237	1.4673
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634
520	471.07	0.0198	0.8915	454.1	750.1	1204.2	0.6536	0.8060	1.4596
540	475.01	0.0199	0.8578	458.6	745.4	1204.0	0.6584	0.7976	1.4560
560	478.85	0.0200	0.8265	463.0	740.8	1203.8	0.6631	0.7893	1.4524
580	482.58	0.0201	0.7973	467.4	736.1	1203.5	0.6676	0.7813	1.4489
600	486.21	0.0201	0.7698	471.6	731.6	1203.2	0.6720	0.7734	1.4454
620	489.75	0.0202	0.7440	475.7	727.2	1202.9	0.6763	0.7658	1.4421
640	493.21	0.0203	0.7198	479.8	722.7	1202.5	0.6805	0.7584	1.4389
660	496.58	0.0204	0.6971	483.8	718.3	1202.1	0.6846	0.7512	1.4358
680	499.88	0.0204	0.6757	487.7	714.0	1201.7	0.6886	0.7441	1.4327
700	503.10	0.0205	0.6554	491.5	709.7	1201.2	0.6925	0.7371	1.4296
720	506.25	0.0206	0.6362	495.3	705.4	1200.7	0.6963	0.7303	1.4266
740	509.34	0.0207	0.6180	499.0	701.2	1200.2	0.7001	0.7237	1.4237
760	512.36	0.0207	0.6007	502.6	697.1	1199.7	0.7037	0.7172	1.4209
780	515.33	0.0208	0.5843	506.2	692.9	1199.1	0.7073	0.7108	1.4181
800	518.23	0.0209	0.5687	509.7	688.9	1198.6	0.7108	0.7045	1.4153
820	521.08	0.0209	0.5538	513.2	684.8	1198.0	0.7143	0.6983	1.4126
840	523.88	0.0210	0.5396	516.6	680.8	1197.4	0.7177	0.6922	1.4099
860	526.63	0.0211	0.5260	520.0	676.8	1196.8	0.7210	0.6862	1.4072
880	529.33	0.0212	0.5130	523.3	672.8	1196.1	0.7243	0.6803	1.4046
900	531.98	0.0212	0.5006	526.6	668.8	1195.4	0.7275	0.6744	1.4020
920	534.59	0.0213	0.4886	529.8	664.9	1194.7	0.7307	0.6687	1.3995
940	537.16	0.0214	0.4772	533.0	661.0	1194.0	0.7339	0.6631	1.3970
960	539.68	0.0214	0.4663	536.2	657.1	1193.3	0.7370	0.6576	1.3945
980	542.17	0.0215	0.4557	539.3	653.3	1192.6	0.7400	0.6521	1.3921
1,000	544.61	0.0216	0.4456	542.4	649.4	1191.8	0.7430	0.6467	1.3897
1,050	550.57	0.0218	0.4218	550.0	639.9	1189.9	0.7504	0.6334	1.3838
1,100	556.31	0.0220	0.4001	557.4	630.4	1187.8	0.7575	0.6205	1.3780
1,150	561.86	0.0221	0.3802	564.6	621.0	1185.6	0.7644	0.6079	1.3723
1,200	567.22	0.0223	0.3619	571.7	611.7	1183.4	0.7711	0.5956	1.3667
1,250	572.42	0.0225	0.3450	578.6	602.4	1181.0	0.7776	0.5836	1.3612
1,300	577.46	0.0227	0.3293	585.4	593.2	1178.6	0.7840	0.5719	1.3559
1,350	582.35	0.0229	0.3148	592.1	584.0	1176.1	0.7902	0.5604	1.3506
1,400	587.10	0.0231	0.3012	598.7	574.7	1173.4	0.7963	0.5491	1.3454
1,450	591.73	0.0233	0.2884	605.2	565.5	1170.7	0.8023	0.5379	1.3402
1,500	596.23	0.0235	0.2765	611.6	556.3	1167.9	0.8082	0.5269	1.3351
1,600	604.90	0.0239	0.2548	624.1	538.0	1162.1	0.8196	0.5053	1.3249
1,700	613.15	0.0243	0.2354	636.3	519.6	1155.9	0.8306	0.4843	1.3149
1,800	621.03	0.0247	0.2179	648.3	501.1	1149.4	0.8412	0.4637	1.3047
1,900	628.58	0.0252	0.2021	660.1	482.4	1142.4	0.8516	0.4433	1.2949
2,000	635.82	0.0257	0.1878	671.7	463.4	1135.1	0.8619	0.4230	1.2849
2,200	649.46	0.0268	0.1625	694.8	424.4	1119.2	0.8820	0.3826	1.2646
2,400	662.12	0.0280	0.1407	718.4	382.7	1101.1	0.9023	0.3411	1.2434
2,600	673.94	0.0295	0.1213	743.0	337.2	1080.2	0.9232	0.2973	1.2205
2,800	684.99	0.0315	0.1035	770.1	284.7	1054.8	0.9459	0.2487	1.1946
3,000	695.36	0.0346	0.0858	802.5	217.8	1020.3	0.9731	0.1885	1.1615
3,200	705.11	0.0444	0.0580	872.4	62.0	934.4	1.0320	0.0532	1.0852
3,206.2	705.40	0.0503	0.0503	902.7	0	902.7	1.0580	0	1.0580

Table III.—Superheated Vapor

Abs press, psi (sat temp)		Sat liquid	Sat vapor	Temperature, °F																					
				120	140	160	180	200	220	240	260	280	300	320	340	360	380	400	420	440	460	480	500	600	700
1	v	0.02	333.6	344.6	356.6	368.6	380.6	392.6	404.5	416.5	428.4	440.4	452.3	464.2	476.2	488.1	500.0	512.0	523.9	535.8	547.7	559.7	571.6	631.2	690.8
	h	69.7	1106.0	1114.3	1123.3	1132.4	1141.4	1150.4	1159.5	1168.5	1177.6	1186.7	1195.8	1204.9	1214.1	1223.3	1232.5	1241.7	1251.0	1260.3	1269.6	1278.9	1288.3	1335.7	1383.8
(101.74)	s	0.1326	1.9782	1.9928	2.0081	2.0230	2.0373	2.0512	2.0647	2.0779	2.0907	2.1031	2.1153	2.1271	2.1387	2.1501	2.1612	2.1720	2.1827	2.1931	2.2034	2.2134	2.2233	2.2702	2.3137
2	v	0.02	173.7	.....	177.96	184.01	190.04	196.06	202.1	208.1	214.1	220.0	226.0	232.0	238.0	244.0	249.9	255.9	261.9	267.8	273.8	279.8	285.7	315.5	345.4
	h	94.0	1116.2	.....	1122.6	1131.8	1140.9	1150.0	1159.1	1168.2	1177.3	1186.5	1195.6	1204.8	1213.9	1223.1	1232.3	1241.6	1250.9	1260.2	1269.5	1278.8	1288.2	1335.6	1383.8
(126.08)	s	0.1749	1.9200	.....	1.9308	1.9458	1.9603	1.9743	1.9879	2.0011	2.0140	2.0265	2.0387	2.0506	2.0622	2.0735	2.0846	2.0955	2.1062	2.1166	2.1269	2.1369	2.1468	2.1938	2.2372
3	v	0.02	118.7	.....	.....	122.48	126.53	130.56	134.58	138.59	142.60	146.60	150.60	154.59	158.58	162.57	166.56	170.54	174.52	178.51	182.48	186.47	190.44	210.3	230.2
	h	109.4	1122.6	.....	.....	1131.2	1140.4	1149.6	1158.8	1167.9	1177.1	1186.2	1195.4	1204.6	1213.8	1223.0	1232.2	1241.5	1250.7	1260.0	1269.4	1278.7	1288.1	1335.5	1383.7
(141.48)	s	0.2308	1.8863	.....	.....	1.9003	1.9150	1.9291	1.9428	1.9561	1.9690	1.9815	1.9937	2.0057	2.0173	2.0287	2.0398	2.0507	2.0614	2.0719	2.0821	2.0921	2.1021	2.1490	2.1925
4	v	0.02	90.63	.....	.....	91.71	94.77	97.81	100.84	103.86	106.87	109.88	112.88	115.88	118.88	121.88	124.87	127.86	130.85	133.84	136.83	139.82	142.80	157.73	172.64
	h	120.9	1127.3	.....	.....	1130.6	1139.9	1149.2	1158.5	1167.6	1176.8	1186.0	1195.2	1204.4	1213.6	1222.8	1232.1	1241.3	1250.6	1259.9	1269.3	1278.7	1288.1	1335.5	1383.7
(152.97)	s	0.2198	1.8625	.....	.....	1.8678	1.8827	1.8969	1.9107	1.9240	1.9370	1.9496	1.9618	1.9738	1.9854	1.9968	2.0080	2.0189	2.0296	2.0401	2.0503	2.0603	2.0703	2.1173	2.1608
5	v	0.02	73.52	.....	.....	.....	75.71	78.16	80.59	83.01	85.43	87.85	90.25	92.66	95.06	97.46	99.86	102.26	104.65	107.04	109.44	111.83	114.22	126.16	138.10
	h	130.1	1131.1	.....	.....	.....	1139.4	1148.8	1158.1	1167.3	1176.5	1185.7	1195.0	1204.2	1213.4	1222.7	1231.9	1241.2	1250.5	1259.8	1269.2	1278.6	1288.0	1335.4	1383.6
(162.24)	s	0.2347	1.8441	.....	.....	.....	1.8574	1.8718	1.8857	1.8991	1.9121	1.9247	1.9370	1.9490	1.9607	1.9721	1.9833	1.9942	2.0049	2.0154	2.0256	2.0356	2.0456	2.0927	2.1361
6	v	0.02	61.98	.....	.....	.....	63.00	65.05	67.09	69.12	71.14	73.16	75.17	77.18	79.18	81.18	83.18	85.18	87.18	89.18	91.17	93.17	95.16	105.12	115.07
	h	138.0	1134.2	.....	.....	.....	1138.9	1148.3	1157.7	1167.0	1176.3	1185.5	1194.7	1204.0	1213.2	1222.5	1231.8	1241.1	1250.4	1259.7	1269.1	1278.5	1287.9	1335.3	1383.6
(170.06)	s	0.2472	1.8292	.....	.....	.....	1.8367	1.8512	1.8651	1.8786	1.8917	1.9044	1.9167	1.9287	1.9404	1.9518	1.9630	1.9740	1.9847	1.9952	2.0055	2.0155	2.0255	2.0725	2.1160
7	v	0.02	53.64	.....	.....	.....	53.93	55.69	57.45	59.19	60.93	62.67	64.39	66.12	67.84	69.56	71.27	72.99	74.70	76.42	78.13	79.84	81.55	90.09	98.62
	h	144.8	1136.9	.....	.....	.....	1138.4	1147.9	1157.4	1166.7	1176.0	1185.3	1194.5	1203.8	1213.1	1222.3	1231.6	1240.9	1250.3	1259.6	1269.0	1278.4	1287.8	1335.3	1383.5
(176.85)	s	0.2581	1.8167	.....	.....	.....	1.8190	1.8336	1.8477	1.8613	1.8744	1.8871	1.8995	1.9115	1.9232	1.9347	1.9459	1.9569	1.9676	1.9781	1.9884	1.9984	2.0084	2.0555	2.0990
8	v	0.02	47.34	.....	.....	.....	.....	48.67	50.22	51.75	53.28	54.80	56.31	57.82	59.33	60.84	62.34	63.84	65.35	66.85	68.34	69.84	71.34	78.82	86.28
	h	150.8	1139.3	.....	.....	.....	.....	1147.5	1157.0	1166.3	1175.7	1185.0	1194.3	1203.6	1212.9	1222.2	1231.5	1240.8	1250.1	1259.5	1268.9	1278.3	1287.7	1335.2	1383.5
(182.86)	s	0.2674	1.8057	.....	.....	.....	.....	1.8184	1.8325	1.8462	1.8594	1.8721	1.8845	1.8966	1.9084	1.9199	1.9311	1.9420	1.9528	1.9633	1.9736	1.9836	1.9936	2.0407	2.0842

9	v	0.02	42.40	....	....	....	....	43.21	44.59	45.96	47.32	48.67	50.03	51.37	52.71	54.06	55.39	56.73	58.07	59.40	60.74	62.07	63.40	70.05	76.66
	h	156.2	1141.4	....	....	....	....	1147.0	1156.6	1166.0	1175.4	1184.8	1194.1	1203.4	1212.7	1222.0	1231.4	1240.7	1250.0	1259.4	1268.8	1278.2	1287.6	1335.2	1383.4
(188.28)	s	0.2759	1.7962	....	....	....	....	1.8049	1.8191	1.8328	1.8461	1.8589	1.8713	1.8834	1.8952	1.9067	1.9179	1.9289	1.9397	1.9502	1.9605	1.9706	1.9806	2.0277	2.0712
10	v	0.02	38.42	....	....	....	....	38.85	40.09	41.33	42.56	43.78	45.00	46.21	47.42	48.63	49.84	51.04	52.24	53.45	54.65	55.85	57.05	63.03	69.01
	h	161.2	1143.3	....	....	....	....	1146.6	1156.2	1165.7	1175.1	1184.5	1193.9	1203.2	1212.5	1221.9	1231.2	1240.6	1249.9	1259.3	1268.7	1278.1	1287.5	1335.1	1383.4
(193.21)	s	0.2835	1.7876	....	....	....	....	1.7927	1.8071	1.8208	1.8341	1.8470	1.8595	1.8716	1.8834	1.8950	1.9062	1.9172	1.9280	1.9385	1.9488	1.9589	1.9689	2.0160	2.0596
11	v	0.02	35.14	....	....	....	....	35.27	36.41	37.54	38.66	39.77	40.88	41.99	43.09	44.19	45.29	46.39	47.48	48.58	49.67	50.76	51.85	57.30	62.73
	h	165.7	1145.0	....	....	....	....	1146.1	1155.8	1165.4	1174.9	1184.3	1193.6	1203.0	1212.4	1221.7	1231.1	1240.4	1249.8	1259.2	1268.6	1278.0	1287.4	1335.0	1383.3
(197.75)	s	0.2903	1.7800	....	....	....	....	1.7816	1.7961	1.8100	1.8233	1.8362	1.8487	1.8609	1.8728	1.8843	1.8956	1.9066	1.9173	1.9279	1.9382	1.9484	1.9583	2.0055	2.0490
12	v	0.02	32.40	....	....	....	....	....	33.34	34.38	35.41	36.43	37.45	38.47	39.48	40.49	41.50	42.51	43.51	44.51	45.52	46.52	47.52	52.51	57.50
	h	170.0	1146.6	....	....	....	....	....	1155.4	1165.0	1174.6	1184.0	1193.4	1202.8	1212.2	1221.6	1230.9	1240.3	1249.7	1259.0	1268.5	1277.9	1287.3	1335.0	1383.3
(201.96)	s	0.2967	1.7730	....	....	....	....	....	1.7860	1.8000	1.8134	1.8264	1.8389	1.8511	1.8630	1.8745	1.8858	1.8969	1.9076	1.9182	1.9285	1.9387	1.9486	1.9958	2.0394
13	v	0.02	30.06	....	....	....	....	....	30.74	31.71	32.66	33.61	34.55	35.49	36.43	37.36	38.29	39.22	40.15	41.08	42.00	42.93	43.85	48.47	53.07
	h	173.9	1148.1	....	....	....	....	....	1155.1	1164.7	1174.3	1183.8	1193.2	1202.6	1212.0	1221.4	1230.8	1240.2	1249.5	1258.9	1268.4	1277.8	1287.3	1334.9	1383.2
(205.88)	s	0.3027	1.7665	....	....	....	....	....	1.7768	1.7908	1.8043	1.8173	1.8299	1.8421	1.8540	1.8656	1.8769	1.8879	1.8987	1.9093	1.9196	1.9298	1.9398	1.9870	2.0306
14	v	0.02	28.04	....	....	....	....	....	28.52	29.41	30.30	31.19	32.06	32.94	33.81	34.68	35.54	36.41	37.27	38.13	38.99	39.85	40.71	45.00	49.27
	h	177.6	1149.5	....	....	....	....	....	1154.6	1164.4	1174.0	1183.5	1193.0	1202.4	1211.8	1221.2	1230.6	1240.0	1249.4	1258.8	1268.3	1277.7	1287.2	1334.8	1383.2
(209.56)	s	0.3083	1.7605	....	....	....	....	....	1.7681	1.7822	1.7958	1.8088	1.8215	1.8337	1.8457	1.8573	1.8686	1.8796	1.8904	1.9010	1.9114	1.9216	1.9315	1.9788	2.0224
14.696	v	0.02	26.80	....	....	....	....	....	27.15	28.00	28.85	29.70	30.53	31.37	32.20	33.03	33.85	34.68	35.50	36.32	37.14	37.96	38.78	42.86	46.94
	h	180.1	1150.4	....	....	....	....	....	1154.4	1164.2	1173.8	1183.3	1192.8	1202.3	1211.7	1221.1	1230.5	1239.9	1249.3	1258.8	1268.2	1277.6	1287.1	1334.8	1383.2
(212.00)	s	0.3120	1.7566	....	....	....	....	....	1.7624	1.7766	1.7902	1.8033	1.8160	1.8283	1.8402	1.8518	1.8631	1.8743	1.8850	1.8956	1.9060	1.9162	1.9261	1.9734	2.0170
15	v	0.02	26.29	....	....	....	....	....	26.59	27.43	28.26	29.09	29.91	30.73	31.54	32.35	33.16	33.97	34.78	35.58	36.38	37.19	37.99	41.99	45.98
	h	181.1	1150.8	....	....	....	....	....	1154.3	1164.1	1173.7	1183.2	1192.8	1202.2	1211.7	1221.1	1230.5	1239.9	1249.3	1258.7	1268.2	1277.6	1287.1	1334.8	1383.1
(213.03)	s	0.3135	1.7549	....	....	....	....	....	1.7601	1.7742	1.7879	1.8010	1.8136	1.8259	1.8379	1.8495	1.8609	1.8719	1.8827	1.8933	1.9037	1.9139	1.9238	1.9711	2.0147
16	v	0.02	24.75	....	....	....	....	....	24.90	25.69	26.47	27.25	28.02	28.79	29.56	30.32	31.08	31.84	32.59	33.35	34.10	34.86	35.61	39.36	43.10
	h	184.4	1152.0	....	....	....	....	....	1153.8	1163.7	1173.4	1183.0	1192.5	1202.0	1211.5	1220.9	1230.3	1239.8	1249.2	1258.6	1268.0	1277.5	1287.0	1334.7	1383.1
(216.32)	s	0.3184	1.7497	....	....	....	....	....	1.7524	1.7667	1.7804	1.7936	1.8063	1.8186	1.8306	1.8422	1.8536	1.8647	1.8755	1.8861	1.8965	1.9067	1.9167	1.9639	2.0076

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Table III.—Superheated Vapor (*Cont.*)

Abs press, psi (sat temp)		Sat liquid	Sat vapor	Temperature. °F 220	230	240	250	260	270	280	290	300	320	340	360	380	400	420	440	460	480	500	550	600	650	700
	v	0.02	23.39	23.41	23.78	24.16	24.53	24.90	25.26	25 63	26.00	26.36	27.08	27.80	28.52	29.24	29.95	30.67	31.38	32.09	32.80	33.51	35.28	37.04	38.80	40.56
17	h	187.6	1153.1	1153.4	1158.4	1163.4	1168.3	1173.1	1177.9	1182.7	1187.5	1192.3	1201.8	1211.3	1220.7	1230.2	1239.6	1249.1	1258.5	1267.9	1277.4	1286.9	1310.7	1334.6	1358.7	1383.0
(219.44)	s	0.3231	1.7449	1.7453	1.7526	1.7597	1.7666	1.7734	1.7801	1.7866	1.7931	1.7994	1.8117	1.8238	1.8354	1.8468	1.8579	1.8687	1.8794	1.8897	1.8999	1.9099	1.9341	1.9572	1.9795	2.0009
	v	0.02	22.17	....	22.44	22.79	23.14	23.49	23.84	24.19	24.53	24.88	25.56	26.25	26.93	27.60	28.28	28.95	29.63	30.30	30.97	31.64	33.31	34.98	36.64	38.31
18	h	190.6	1154.2	....	1158.0	1163.0	1167.9	1172.8	1177.7	1182.5	1187.3	1192.1	1201.6	1211.1	1220.6	1230.0	1239.5	1248.9	1258.4	1267.8	1277.3	1286.8	1310.6	1334.6	1358.7	1383.0
(222.41)	s	0.3275	1.7403	....	1.7458	1.7530	1.7599	1.7668	1.7735	1.7800	1.7865	1.7928	1.8052	1.8173	1.8290	1.8404	1.8515	1.8623	1.8730	1.8834	1.8936	1.9035	1.9277	1.9509	1.9731	1.9945
	v	0.02	21.08	....	21.24	21.57	21.91	22.24	22.57	22.90	23.23	23.56	24.20	24.85	25.50	26.14	26.78	27.42	28.06	28.70	29.23	29.97	31.55	33.13	34.71	36.29
19	h	193.4	1155.3	....	1157.7	1162.7	1167.6	1172.5	1177.4	1182.2	1187.1	1191.9	1201.4	1210.9	1220.4	1229.9	1239.4	1248.8	1258.3	1267.7	1277.2	1286.7	1310.5	1334.5	1358.6	1382.9
(225.24)	s	0.3317	1.7360	....	1.7394	1.7466	1.7536	1.7605	1.7672	1.7738	1.7803	1.7867	1.7991	1.8111	1.8229	1.8343	1.8454	1.8563	1.8669	1.8773	1.8875	1.8975	1.9217	1.9449	1.9671	1.9885
	v	0.02	20.09	....	20.15	20.48	20.79	21.11	21.43	21.74	22.05	22.36	22.98	23.60	24.21	24.82	25.43	26.04	26.65	27.25	27.86	28.46	29.97	31.47	32.97	34.47
20	h	196.2	1156.3	....	1157.3	1162.3	1167.3	1172.2	1177.1	1182.0	1186.8	1191.6	1201.2	1210.8	1220.3	1229.7	1239.2	1248.7	1258.2	1267.6	1277.1	1286.6	1310.5	1334.4	1358.6	1382.9
(227.96)	s	0.3356	1.7319	....	1.7333	1.7405	1.7476	1.7545	1.7613	1.7679	1.7744	1.7808	1.7932	1.8053	1.8170	1.8285	1.8396	1.8505	1.8612	1.8716	1.8818	1.8918	1.9160	1.9392	1.9614	1.9829
	v	0.017	19.192	....	....	19.482	19.786	20.09	20.39	20.69	20.99	21.29	21.88	22.46	23.05	23.63	24.21	24.79	25.37	25.95	26.52	27.10	28.54	29.97	31.40	32.82
21	h	193.8	1157.2	....	....	1162.0	1167.0	1171.9	1176.8	1181.7	1186.6	1191.4	1201.0	1210.6	1220.1	1229.6	1239.1	1248.6	1258.1	1267.5	1277.0	1286.5	1310.4	1334.4	1358.5	1382.8
(230.57)	s	0.3395	1.7280	....	....	1.7348	1.7419	1.7488	1.7556	1.7622	1.7687	1.7751	1.7876	1.7997	1.8115	1.8230	1.8341	1.8450	1.8557	1.8661	1.8763	1.8863	1.9106	1.9338	1.9560	1.9774
	v	0.017	18.375	....	....	18.579	18.870	19.160	19.449	19.736	20.02	20.31	20.87	21.43	21.99	22.55	23.11	23.66	24.21	24.76	25.31	25.86	27.23	28.60	29.97	31.33
22	h	201.3	1158.1	....	....	1161.6	1166.6	1171.6	1176.5	1181.4	1186.3	1191.2	1200.8	1210.4	1219.9	1229.5	1239.0	1248.4	1257.9	1267.4	1276.9	1286.4	1310.3	1334.3	1358.5	1382.8
(233.07)	s	0.3431	1.7242	....	....	1.7292	1.7363	1.7433	1.7501	1.7568	1.7633	1.7698	1.7823	1.7944	1.8062	1.8177	1.8289	1.8398	1.8505	1.8609	1.8711	1.8811	1.9054	1.9286	1.9509	1.9723
	v	0.017	17.627	....	....	17.754	18.034	18.312	18.589	18.864	19.138	19.411	19.953	20.49	21.03	21.56	22.09	22.62	23.15	23.68	24.21	24.73	26.04	27.35	28.66	29.96
23	h	203.8	1159.0	....	....	1161.2	1166.3	1171.3	1176.3	1181.2	1186.1	1190.9	1200.6	1210.2	1219.8	1229.3	1238.8	1248.3	1257.8	1267.3	1276.8	1286.4	1310.2	1334.2	1358.4	1382.7
(235.49)	s	0.3466	1.7206	....	....	1.7239	1.7311	1.7381	1.7449	1.7516	1.7582	1.7646	1.7772	1.7894	1.8012	1.8127	1.8238	1.8348	1.8455	1.8559	1.8661	1.8762	1.9004	1.9236	1.9459	1.9674
	v	0.017	16.938	....	....	16.997	17.267	17.535	17.801	18.065	18.329	18.590	19.111	19.628	20.14	20.65	21.17	21.67	22.18	22.69	23.19	23.70	24.96	26.21	27.46	28.71
24	h	206.1	1159.8	....	....	1160.9	1166.0	1171.0	1176.0	1180.9	1185.8	1190.7	1200.4	1210.0	1219.6	1229.2	1238.7	1248.2	1257.7	1267.2	1276.7	1286.3	1310.2	1334.2	1358.4	1382.7
(237.82)	s	0.3500	1.7172	....	....	1.7188	1.7260	1.7330	1.7399	1.7466	1.7532	1.7597	1.7723	1.7845	1.7963	1.8078	1.8190	1.8300	1.8407	1.8511	1.8614	1.8714	1.8957	1.9189	1.9412	1.9626

25 (240.07)	v	0.017	16.303	....	....	....	16.561	16.819	17.076	17.330	17.584	17.836	18.337	18.834	19.329	19.821	20.31	20.80	21.29	21.77	22.26	22.74	23.95	25.16	26.36	27.56
	h	208.4	1160.6	....	....	....	1165.6	1170.7	1175.7	1180.6	1185.6	1190.5	1200.2	1209.8	1219.4	1229.0	1238.5	1248.1	1257.6	1267.1	1276.6	1286.2	1310.1	1334.1	1358.3	1382.6
	s	0.3533	1.7139	....	....	....	1.7212	1.7282	1.7351	1.7418	1.7485	1.7550	1.7676	1.7798	1.7917	1.8032	1.8144	1.8254	1.8361	1.8465	1.8568	1.8668	1.8911	1.9144	1.9367	1.9581
26 (242.25)	v	0.017	15.715	....	....	....	15.910	16.159	16.406	16.652	16.896	17.139	17.622	18.101	18.577	19.051	19.524	19.994	20.46	20.93	21.40	21.86	23.03	24.19	25.34	26.50
	h	210.6	1161.3	....	....	....	1165.3	1170.4	1175.4	1180.4	1185.3	1190.2	1200.0	1209.7	1219.3	1228.9	1238.4	1248.0	1257.5	1267.0	1276.5	1286.1	1310.0	1334.0	1358.2	1382.6
	s	0.3564	1.7108	....	....	....	1.7164	1.7235	1.7305	1.7372	1.7439	1.7504	1.7631	1.7753	1.7872	1.7987	1.8100	1.8210	1.8317	1.8421	1.8524	1.8624	1.8868	1.9100	1.9323	1.9538
27 (244.36)	v	0.017	15.170	....	....	....	15.307	15.548	15.786	16.024	16.259	16.494	16.960	17.422	17.882	18.339	18.794	19.248	19.700	20.15	20.60	21.05	22.17	23.29	24.40	25.51
	h	212.8	1162.0	....	....	....	1165.0	1170.1	1175.1	1180.1	1185.1	1190.0	1199.8	1209.5	1219.1	1228.7	1238.3	1247.8	1257.4	1266.9	1276.4	1286.0	1309.9	1334.0	1358.2	1382.5
	s	0.3594	1.7078	....	....	....	1.7119	1.7190	1.7260	1.7328	1.7395	1.7460	1.7587	1.7710	1.7829	1.7944	1.8057	1.8167	1.8274	1.8379	1.8482	1.8582	1.8825	1.9058	1.9281	1.9496
28 (246.41)	v	0.017	14.663	....	....	....	14.747	14.980	15.211	15.440	15.669	15.895	16.345	16.792	17.235	17.677	18.116	18.554	18.991	19.426	19.860	20.29	21.37	22.45	23.53	24.60
	h	214.8	1162.7	....	....	....	1164.6	1169.7	1174.8	1179.8	1184.8	1189.8	1199.6	1209.3	1218.9	1228.6	1238.1	1247.7	1257.2	1266.8	1276.3	1285.9	1309.9	1333.9	1358.1	1382.5
	s	0.3623	1.7048	....	....	....	1.7075	1.7146	1.7217	1.7285	1.7352	1.7417	1.7545	1.7668	1.7787	1.7903	1.8016	1.8126	1.8233	1.8338	1.8441	1.8541	1.8785	1.9017	1.9241	1.9455
29 (248.40)	v	0.017	14.189	....	....	....	14.225	14.451	14.675	14.897	15.118	15.337	15.773	16.205	16.634	17.061	17.485	17.909	18.331	18.751	19.171	19.590	20.63	21.67	22.71	23.75
	h	216.9	1163.4	....	....	....	1164.3	1169.4	1174.5	1179.6	1184.6	1189.5	1199.3	1209.1	1218.8	1228.4	1238.0	1247.6	1257.1	1266.7	1276.2	1285.8	1309.8	1333.9	1358.1	1382.4
	s	0.3652	1.7020	....	....	....	1.7032	1.7104	1.7175	1.7243	1.7310	1.7376	1.7504	1.7627	1.7747	1.7863	1.7976	1.8086	1.8193	1.8298	1.8401	1.8502	1.8746	1.8978	1.9201	1.9416
30 (250.33)	v	0.017	13.746	....	....	....	....	13.957	14.174	14.390	14.604	14.816	15.238	15.657	16.072	16.485	16.897	17.306	17.714	18.121	18.528	18.933	19.943	20.95	21.95	22.96
	h	218.8	1164.1	....	....	....	....	1169.1	1174.2	1179.3	1184.3	1189.3	1199.1	1208.9	1218.6	1228.3	1237.9	1247.5	1257.0	1266.6	1276.2	1285.7	1309.7	1333.8	1358.0	1382.4
	s	0.3680	1.6993	....	....	....	....	1.7063	1.7134	1.7203	1.7270	1.7336	1.7464	1.7588	1.7708	1.7824	1.7937	1.8047	1.8155	1.8260	1.8363	1.8464	1.8708	1.8940	1.9164	1.9379
31 (252.22)	v	0.017	13.330	....	....	....	....	13.495	13.706	13.915	14.123	14.329	14.738	15.144	15.547	15.947	16.346	16.743	17.138	17.532	17.926	18.318	19.296	20.27	21.24	22.21
	h	220.7	1164.7	....	....	....	....	1168.8	1173.9	1179.0	1184.0	1189.0	1198.9	1208.7	1218.4	1228.1	1237.7	1247.3	1256.9	1266.5	1276.1	1285.6	1309.6	1333.7	1358.0	1382.3
	s	0.3707	1.6967	....	....	....	....	1.7024	1.7095	1.7164	1.7231	1.7298	1.7426	1.7550	1.7670	1.7787	1.7900	1.8010	1.8118	1.8223	1.8326	1.8427	1.8671	1.8904	1.9127	1.9342
32 (254.05)	v	0.017	12.940	....	....	....	....	13.062	13.267	13.470	13.672	13.872	14.270	14.664	15.055	15.443	15.829	16.214	16.598	16.980	17.361	17.741	18.689	19.634	20.58	21.52
	h	222.6	1165.4	....	....	....	....	1168.5	1173.6	1178.7	1183.8	1188.8	1198.7	1208.5	1218.3	1227.9	1237.6	1247.2	1256.8	1266.4	1276.0	1285.5	1309.6	1333.7	1357.9	1382.3
	s	0.3733	1.6941	....	....	....	....	1.6985	1.7056	1.7126	1.7194	1.7260	1.7389	1.7513	1.7634	1.7750	1.7864	1.7974	1.8082	1.8187	1.8290	1.8391	1.8635	1.8868	1.9092	1.9307
33 (255.84)	v	0.017	12.572	....	....	....	....	12.656	12.855	13.052	13.249	13.443	13.830	14.212	14.592	14.969	15.344	15.718	16.090	16.461	16.831	17.200	18.120	19.036	19.950	20.86
	h	224.4	1166.0	....	....	....	....	1168.1	1173.3	1178.4	1183.5	1188.6	1198.5	1208.3	1218.1	1227.8	1237.4	1247.1	1256.7	1266.3	1275.9	1285.4	1309.5	1333.6	1357.8	1382.2
	s	0.3758	1.6917	....	....	....	....	1.6947	1.7019	1.7089	1.7157	1.7224	1.7353	1.7477	1.7598	1.7715	1.7829	1.7939	1.8047	1.8153	1.8256	1.8357	1.8601	1.8834	1.9058	1.9273
34 (257.58)	v	0.017	12.226	....	....	....	....	12.273	12.467	12.659	12.850	13.040	13.415	13.788	14.157	14.523	14.888	15.251	15.612	15.973	16.332	16.690	17.584	18.474	19.361	20.25
	h	226.2	1166.5	....	....	....	....	1167.8	1173.0	1178.2	1183.3	1188.3	1198.3	1208.2	1217.9	1227.6	1237.3	1246.9	1256.6	1266.2	1275.8	1285.4	1309.4	1333.5	1357.8	1382.2
	s	0.3783	1.6893	....	....	....	....	1.6911	1.6983	1.7053	1.7121	1.7188	1.7318	1.7443	1.7564	1.7681	1.7794	1.7905	1.8013	1.8119	1.8222	1.8323	1.8567	1.8800	1.9024	1.9239

Table III.—Superheated Vapor (*Cont.*)

Abs press, psi (sat temp)		Sat liquid	Sat vapor	260	270	280	290	300	320	340	360	380	400	420	440	460	480	500	520	540	560	580	600	650	700
				Temperature, °F																					
	v	0.017	11.898	11.911	12.101	12.288	12.474	12.659	13.025	13.387	13.746	14.103	14.457	14.810	15.162	15.512	15.862	16.210	16.558	16.905	17.251	17.597	17.943	18.805	19.666
35	*h*	227.9	1167.1	1167.[illegible]	1172.7	1177.9	1183.0	1188.1	1198.1	1208.0	1217.8	1227.5	1237.2	1246.8	1256.4	1266.1	1275.7	1285.3	1294.9	1304.5	1314.1	1323.8	1333.5	1357.7	1382.1
(259.28)	*s*	0.3807	1.6870	1.6875	1.6948	1.7018	1.7087	1.7154	1.7284	1.7409	1.7530	1.7647	1.7761	1.7872	1.7980	1.8086	1.8189	1.8290	1.8390	1.8487	1.8582	1.8676	1.8768	1.8992	1.9207
	v	0.017	10.498	.....	10.544	10.711	10.876	11.040	11.364	11.684	12.001	12.315	12.628	12.938	13.247	13.555	13.862	14.168	14.473	14.778	15.082	15.385	15.688	16.444	17.198
40	*h*	236.0	1169.7	.....	1171.2	1176.5	1181.7	1186.8	1197.0	1207.0	1216.9	1226.7	1236.5	1246.2	1255.9	1265.5	1275.2	1284.8	1294.5	1304.1	1313.8	1323.4	1333.1	1357.4	1381.9
(267.25)		0.3919	1.6763	.....	1.6783	1.6855	1.6925	1.6994	1.7126	1.7252	1.7375	1.7493	1.7608	1.7719	1.7828	1.7934	1.8038	1.8140	1.8239	1.8337	1.8432	1.8526	1.8619	1.8843	1.9058
	v	0.017	9.401	.....	.....	9.484	9.633	9.781	10.072	10.359	10.643	10.925	11.204	11.482	11.758	12.033	12.307	12.580	12.852	13.123	13.394	13.665	13.935	14.608	15.279
45	*h*	243.4	1172.0	.....	.....	1175.0	1180.3	1185.6	1195.9	1206.0	1216.0	1225.9	1235.8	1245.6	1255.3	1265.0	1274.7	1284.4	1294.0	1303.7	1313.4	1323.1	1332.8	1357.1	1381.6
(274.44)	*s*	0.4019	1.6669	.....	.....	1.6709	1.6781	1.6850	1.6985	1.7113	1.7237	1.7356	1.7472	1.7584	1.7693	1.7800	1.7904	1.8006	1.8106	1.8204	1.8300	1.8394	1.8487	1.8711	1.8927
	v	0.017	8.515	.....	.....	.....	8.638	8.773	9.038	9.299	9.557	9.812	10.065	10.317	10.567	10.815	11.062	11.309	11.555	11.800	12.044	12.288	12.532	13.139	13.744
50	*h*	250.1	1174.1	.....	.....	.....	1178.9	1184.3	1194.8	1205.0	1215.2	1225.2	1235.1	1244.9	1254.7	1264.5	1274.2	1283.9	1293.6	1303.3	1313.0	1322.7	1332.5	1356.9	1381.4
(281.01)	*s*	0.4110	1.6585	.....	.....	.....	1.6650	1.6721	1.6857	1.6987	1.7112	1.7233	1.7349	1.7462	1.7572	1.7680	1.7784	1.7887	1.7987	1.8085	1.8181	1.8275	1.8368	1.8593	1.8809
	v	0.017	7.787	...	.....	.....	7.823	7.947	8.192	8.432	8.668	8.902	9.134	9.364	9.592	9.819	10.044	10.269	10.493	10.717	10.940	11.162	11.384	11.937	12.488
55	*h*	256.3	1175.9	.....	...	.....	1177.5	1183.0	1193.6	1204.0	1214.3	1224.4	1234.3	1244.3	1254.1	1263.9	1273.7	1283.4	1293.2	1302.9	1312.6	1322.4	1332.1	1356.6	1381.1
(287.07)	*s*	0.4193	1.6509	.....	.....	.....	1.6530	1.6602	1.6740	1.6872	1.6999	1.7120	1.7238	1.7351	1.7462	1.7570	1.7675	1.7778	1.7878	1.7977	1.8073	1.8168	1.8261	1.8486	1.8703
	v	0.017	7.175	.....	.....	.....	.....	7.259	7.486	7.708	7.927	8.143	8.357	8.569	8.779	8.988	9.196	9.403	9.609	9.814	10.019	10.223	10.427	16.935	11.441
60	*h*	262.1	1177.6	.....	.....	.....	.....	1181.6	1192.5	1203.0	1213.4	1223.6	1233.6	1243.6	1253.5	1263.4	1273.2	1283.0	1292.7	1302.5	1312.3	1322.0	1331.8	1356.3	1380.9
(292.71)	*s*	0.4270	1.6438	.....	.....	.....	.....	1.6492	1.6633	1.6766	1.6894	1.7017	1.7135	1.7250	1.7361	1.7470	1.7575	1.7678	1.7779	1.7878	1.7975	1.8069	1.8162	1.8388	1.8605

65 (297.97)	v	0.017	6.655	.....	.....	.....	.....	6.676	6.889	7.096	7.300	7.501	7.700	7.896	8.091	8.285	8.478	8.670	8.861	9.051	9.240	9.429	9.618	10.087	10.555
	h	267.5	1179.1	.....	.....	.....	.....	1180.3	1191.3	1202.0	1212.5	1222.7	1232.9	1242.9	1252.9	1262.8	1272.7	1282.5	1292.3	1302.1	1311.9	1321.7	1331.5	1356.0	1380.6
	s	0.4342	1.6374	.....	.....	.....	.....	1.6389	1.6532	1.6668	1.6797	1.6921	1.7040	1.7156	1.7268	1.7377	1.7483	1.7586	1.7687	1.7786	1.7884	1.7979	1.8072	1.8298	1.8515
70 (302.92)	v	0.017	6.206	.....	.....	.....	.....	.....	6.376	6.571	6.762	6.950	7.136	7.320	7.502	7.683	7.863	8.041	8.219	8.396	8.573	8.749	8.924	9.361	9.796
	h	272.6	1180.6	.....	.....	.....	.....	.....	1190.1	1201.0	1211.5	1221.9	1232.1	1242.3	1252.3	1262.2	1272.2	1282.0	1291.9	1301.7	1311.5	1321.3	1331.1	1355.7	1380.4
	s	0.4409	1.6315	.....	.....	.....	.....	.....	1.6438	1.6576	1.6707	1.6832	1.6952	1.7068	1.7181	1.7291	1.7397	1.7501	1.7603	1.7702	1.7799	1.7894	1.7988	1.8215	1.8432
75 (307.60)	v	0.018	5.816	.....	.....	.....	.....	.....	5.932	6.116	6.296	6.473	6.648	6.820	6.991	7.161	7.329	7.496	7.663	7.829	7.994	8.159	8.323	8.731	9.138
	h	277.4	1181.9	.....	.....	.....	.....	.....	1188.9	1199.9	1210.6	1221.1	1231.4	1241.6	1251.7	1261.7	1271.6	1281.5	1291.4	1301.3	1311.1	1321.0	1330.8	1355.4	1380.1
	s	0.4472	1.6259	.....	.....	.....	.....	.....	1.6350	1.6489	1.6622	1.6748	1.6869	1.6987	1.7100	1.7210	1.7317	1.7421	1.7523	1.7623	1.7720	1.7816	1.7910	1.8137	1.8354
80 (312.03)	v	0.018	5.472	.....	.....	.....	.....	.....	5.543	5.718	5.888	6.055	6.220	6.383	6.544	6.704	6.862	7.020	7.176	7.332	7.488	7.642	7.797	8.180	8.562
	h	282.0	1183.1	.....	.....	.....	.....	.....	1187.6	1198.8	1209.7	1220.3	1230.7	1240.9	1251.1	1261.1	1271.1	1281.1	1291.0	1300.9	1310.7	1320.6	1330.5	1355.1	1379.9
	s	0.4531	1.6207	.....	.....	.....	.....	.....	1.6266	1.6407	1.6541	1.6669	1.6791	1.6909	1.7023	1.7134	1.7242	1.7346	1.7449	1.7549	1.7646	1.7742	1.7836	1.8063	1.8281
85 (316.25)	v	0.018	5.168	.....	.....	.....	.....	.....	5.200	5.366	5.528	5.687	5.843	5.997	6.150	6.301	6.450	6.599	6.747	6.894	7.041	7.187	7.332	7.694	8.054
	h	286.4	1184.2	.....	.....	.....	.....	.....	1186.4	1197.7	1208.7	1219.4	1229.9	1240.2	1250.4	1260.6	1270.6	1280.6	1290.5	1300.5	1310.4	1320.2	1330.1	1354.8	1379.6
	s	0.4587	1.6158	.....	.....	.....	.....	.....	1.6186	1.6330	1.6465	1.6594	1.6718	1.6837	1.6951	1.7062	1.7171	1.7276	1.7378	1.7479	1.7577	1.7673	1.7767	1.7995	1.8213
90 (320.27)	v	0.018	4.896	.....	.....	.....	.....	.....	.....	5.053	5.208	5.359	5.508	5.654	5.799	5.942	6.084	6.225	6.365	6.505	6.644	6.782	6.920	7.262	7.603
	h	290.6	1185.3	.....	.....	.....	.....	.....	.....	1196.6	1207.7	1218.6	1229.1	1239.5	1249.8	1260.0	1270.1	1280.1	1290.1	1300.0	1310.0	1319.9	1329.8	1354.6	1379.4
	s	0.4641	1.6112	.....	.....	.....	.....	.....	.....	1.6256	1.6393	1.6523	1.6648	1.6767	1.6883	1.6995	1.7103	1.7209	1.7312	1.7413	1.7511	1.7607	1.7702	1.7930	1.8149
95 (324.12)	v	0.018	4.652	.....	.....	.....	.....	.....	.....	4.773	4.921	5.065	5.207	5.347	5.485	5.621	5.756	5.891	6.024	6.156	6.288	6.419	6.550	6.875	7.199
	h	294.6	1186.2	.....	.....	.....	.....	.....	.....	1195.5	1206.7	1217.7	1228.3	1238.8	1249.2	1259.4	1269.5	1279.6	1289.6	1299.6	1309.6	1319.5	1329.4	1354.3	1379.1
	s	0.4692	1.6068	.....	.....	.....	.....	.....	.....	1.6185	1.6324	1.6455	1.6581	1.6702	1.6818	1.6930	1.7039	1.7146	1.7249	1.7350	1.7448	1.7545	1.7640	1.7868	1.8087
100 (327.81)	v	0.018	4.432	.....	.....	.....	.....	.....	.....	4.521	4.663	4.801	4.937	5.071	5.202	5.333	5.462	5.589	5.717	5.843	5.968	6.093	6.218	6.527	6.835
	h	298.4	1187.2	.....	.....	.....	.....	.....	.....	1194.3	1205.7	1216.8	1227.6	1238.1	1248.6	1258.8	1269.0	1279.1	1289.2	1299.2	1309.2	1319.2	1329.1	1354.0	1378.9
	s	0.4740	1.6026	.....	.....	.....	.....	.....	.....	1.6117	1.6258	1.6391	1.6518	1.6639	1.6756	1.6869	1.6979	1.7085	1.7189	1.7290	1.7389	1.7486	1.7581	1.7810	1.8029

TABLE III.—SUPERHEATED VAPOR (*Concl.*)

Abs press, psi (sat temp)		Sat liquid	Sat vapor	340	350	360	370	380	390	400	420	440	460	480	500	520	540	560	580	600	620	640	660	680	700
				Temperature, °F																					
110	v	0.018	4.049	4.085	4.151	4.216	4.281	4.345	4.408	4.470	4.593	4.714	4.834	4.952	5.069	5.186	5.301	5.416	5.530	5.644	5.757	5.870	5.983	6.095	6.207
(334.77)	h	305.7	1188.9	1192.0	1197.9	1203.7	1209.4	1215.0	1220.5	1226.0	1236.7	1247.3	1257.7	1268.0	1278.2	1288.3	1298.4	1308.4	1318.4	1328.4	1338.4	1348.4	1358.4	1368.4	1378.3
	s	0.4832	1.5948	1.5988	1.6062	1.6133	1.6202	1.6269	1.6334	1.6398	1.6521	1.6640	1.6754	1.6865	1.6972	1.7077	1.7179	1.7278	1.7376	1.7471	1.7564	1.7656	1.7746	1.7834	1.7921
120	v	0.018	3.728	....	3.783	3.844	3.904	3.964	4.023	4.081	4.195	4.307	4.418	4.527	4.636	4.743	4.849	4.955	5.061	5.165	5.270	5.374	5.477	5.580	5.683
(341.25)	h	312.4	1190.4	....	1195.7	1201.6	1207.5	1213.2	1218.8	1224.4	1235.3	1246.0	1256.5	1266.9	1277.2	1287.4	1297.5	1307.6	1317.7	1327.7	1337.8	1347.8	1357.8	1367.8	1377.8
	s	0.4916	1.5878	....	1.5944	1.6017	1.6087	1.6156	1.6222	1.6287	1.6413	1.6533	1.6649	1.6760	1.6869	1.6974	1.7077	1.7177	1.7274	1.7370	1.7464	1.7556	1.7646	1.7734	1.7822
130	v	0.018	3.455	....	3.471	3.529	3.585	3.641	3.696	3.751	3.858	3.963	4.066	4.168	4.269	4.369	4.468	4.566	4.663	4.761	4.857	4.953	5.049	5.145	5.240
(347.32)	h	318.8	1191.7	....	1193.4	1199.5	1205.5	1211.3	1217.1	1222.7	1233.8	1244.6	1255.3	1265.8	1276.2	1286.5	1296.7	1306.8	1317.0	1327.0	1337.1	1347.2	1357.2	1367.3	1377.3
	s	0.4995	1.5812	....	1.5833	1.5908	1.5980	1.6050	1.6118	1.6184	1.6312	1.6433	1.6550	1.6663	1.6773	1.6879	1.6982	1.7083	1.7181	1.7277	1.7371	1.7463	1.7554	1.7643	1.7730
140	v	0.018	3.220	....	....	3.258	3.312	3.365	3.417	3.468	3.569	3.667	3.764	3.860	3.954	4.047	4.140	4.232	4.323	4.413	4.503	4.593	4.683	4.772	4.861
(353.02)	h	324.8	1193.0	....	....	1197.3	1203.5	1209.4	1215.3	1221.1	1232.3	1243.3	1254.1	1264.7	1275.2	1285.5	1295.8	1306.0	1316.2	1326.4	1336.5	1346.6	1356.6	1366.7	1376.8
	s	0.5069	1.5751	....	....	1.5804	1.5879	1.5950	1.6020	1.6087	1.6217	1.6340	1.6458	1.6573	1.6683	1.6790	1.6894	1.6995	1.7094	1.7190	1.7285	1.7378	1.7468	1.7558	1.7645
150	v	0.018	3.015	....	....	3.023	3.074	3.124	3.174	3.223	3.318	3.411	3.502	3.592	3.681	3.769	3.856	3.942	4.028	4.113	4.197	4.281	4.365	4.449	4.532
(358.42)	h	330.5	1194.1	....	....	1195.1	1201.4	1207.5	1213.5	1219.4	1230.8	1242.0	1252.9	1263.6	1274.1	1284.6	1295.0	1305.2	1315.5	1325.7	1335.8	1346.0	1356.1	1366.2	1376.3
	s	0.5138	1.5694	....	....	1.5706	1.5782	1.5856	1.5927	1.5995	1.6127	1.6252	1.6372	1.6487	1.6599	1.6707	1.6811	1.6913	1.7012	1.7109	1.7204	1.7297	1.7389	1.7478	1.7566
160	v	0.018	2.834	....	....	....	2.866	2.914	2.961	3.008	3.098	3.187	3.273	3.359	3.443	3.525	3.607	3.689	3.769	3.849	3.929	4.008	4.087	4.166	4.244
(363.53)	h	335.9	1195.1	....	....	....	1199.2	1205.5	1211.6	1217.6	1229.3	1240.6	1251.6	1262.4	1273.1	1283.7	1294.1	1304.4	1314.7	1325.0	1335.2	1345.3	1355.5	1365.6	1375.7
	s	0.5204	1.5640	....	....	....	1.5690	1.5766	1.5838	1.5908	1.6042	1.6169	1.6291	1.6407	1.6519	1.6628	1.6733	1.6836	1.6936	1.7033	1.7129	1.7222	1.7313	1.7403	1.7491
170	v	0.018	2.675	....	....	....	2.682	2.728	2.773	2.818	2.904	2.989	3.071	3.152	3.232	3.310	3.388	3.465	3.541	3.617	3.692	3.767	3.842	3.916	3.990
(368.41)	h	341.1	1196.0	....	....	....	1197.1	1203.5	1209.7	1215.8	1227.7	1239.2	1250.4	1261.3	1272.1	1282.7	1293.2	1303.6	1314.0	1324.2	1334.5	1344.7	1354.9	1365.1	1375.2
	s	0.5266	1.5590	....	....	....	1.5603	1.5679	1.5753	1.5825	1.5961	1.6090	1.6213	1.6331	1.6444	1.6554	1.6660	1.6763	1.6864	1.6962	1.7057	1.7151	1.7243	1.7333	1.7421
180	v	0.018	2.532	....	....	....	....	2.563	2.606	2.649	2.732	2.813	2.891	2.969	3.044	3.119	3.193	3.266	3.339	3.411	3.482	3.553	3.623	3.694	3.764
(373.06)	h	346.0	1196.9	....	....	....	....	1201.4	1207.8	1214.0	1226.1	1237.8	1249.1	1260.2	1271.0	1281.7	1292.3	1302.8	1313.2	1323.5	1333.8	1344.1	1354.3	1364.5	1374.7
	s	0.5325	1.5542	....	....	....	....	1.5596	1.5672	1.5745	1.5884	1.6015	1.6139	1.6258	1.6373	1.6483	1.6590	1.6694	1.6795	1.6894	1.6990	1.7084	1.7176	1.7266	1.7355
190	v	0.018	2.404	....	....	....	....	2.414	2.456	2.498	2.577	2.655	2.730	2.804	2.877	2.948	3.019	3.089	3.157	3.226	3.294	3.361	3.429	3.495	3.562
(377.51)	h	350.8	1197.6	....	....	....	....	1199.3	1205.8	1212.2	1224.5	1236.3	1247.8	1259.0	1270.0	1280.8	1291.4	1302.0	1312.4	1322.8	1333.2	1343.5	1353.7	1363.9	1374.2
	s	0.5381	1.5497	....	....	....	....	1.5517	1.5594	1.5668	1.5810	1.5943	1.6069	1.6189	1.6305	1.6416	1.6524	1.6628	1.6730	1.6829	1.6925	1.7020	1.7112	1.7203	1.7297

200 (381.79)	v	0.018	2.288	...	...	...	...	...	2.321	2.361	2.438	2.513	2.585	2.656	2.726	2.794	2.862	2.928	2.994	3.060	3.124	3.189	3.253	3.316	3.380
	h	355.4	1198.4	...	...	...	...	...	1203.8	1210.3	1222.9	1234.9	1246.5	1257.8	1268.9	1279.8	1290.5	1301.1	1311.7	1322.1	1332.5	1342.8	1353.1	1363.4	1373.6
	s	0.5435	1.5453	...	...	...	...	...	1.5518	1.5594	1.5738	1.5873	1.6001	1.6123	1.6240	1.6352	1.6460	1.6566	1.6668	1.6767	1.6864	1.6959	1.7052	1.7143	1.7232
210 (385.90)	v	0.018	2.183	...	...	...	...	...	2.199	2.238	2.312	2.384	2.454	2.522	2.589	2.655	2.720	2.783	2.847	2.909	2.971	3.033	3.094	3.155	3.215
	h	359.8	1199.0	...	...	...	...	...	1201.8	1208.4	1221.2	1233.4	1245.2	1256.6	1267.8	1278.8	1289.6	1300.3	1310.9	1321.4	1331.8	1342.2	1352.5	1362.8	1373.1
	s	0.5487	1.5412	...	...	...	...	...	1.5445	1.5523	1.5670	1.5807	1.5936	1.6059	1.6177	1.6291	1.6400	1.6506	1.6608	1.6709	1.6806	1.6901	1.6994	1.7085	1.7175
220 (389.86)	v	0.019	2.087	...	...	...	...	...	2.087	2.125	2.198	2.267	2.335	2.400	2.465	2.528	2.590	2.652	2.712	2.772	2.831	2.891	2.949	3.008	3.066
	h	364.0	1199.6	...	...	...	...	...	1199.7	1206.5	1219.5	1231.9	1243.8	1255.4	1266.7	1277.8	1288.7	1299.5	1310.1	1320.7	1331.1	1341.6	1351.9	1362.3	1372.6
	s	0.5537	1.5372	...	...	...	...	...	1.5373	1.5453	1.5603	1.5742	1.5874	1.5998	1.6117	1.6231	1.6342	1.6448	1.6551	1.6652	1.6750	1.6845	1.6939	1.7030	1.7120
230 (393.68)	v	0.0185	1.9992	...	...	...	...	...	...	2.022	2.093	2.160	2.226	2.289	2.351	2.412	2.472	2.531	2.589	2.647	2.704	2.761	2.817	2.873	2.929
	h	368.1	1200.1	...	...	...	...	...	...	1204.5	1217.8	1230.4	1242.5	1254.2	1265.6	1276.8	1287.8	1298.6	1309.3	1319.9	1330.5	1340.9	1351.3	1361.7	1372.0
	s	0.5585	1.5334	...	...	...	...	...	...	1.5385	1.5538	1.5680	1.5813	1.5939	1.6059	1.6174	1.6286	1.6393	1.6497	1.6598	1.6696	1.6792	1.6886	1.6978	1.7068
240 (397.37)	v	0.0186	1.9183	...	...	...	...	...	...	1.9276	1.9964	2.062	2.126	2.187	2.247	2.306	2.364	2.421	2.477	2.533	2.587	2.642	2.696	2.750	2.804
	h	372.1	1200.6	...	...	...	...	...	...	1202.5	1216.0	1228.8	1241.1	1253.0	1264.5	1275.8	1286.9	1297.8	1308.5	1319.2	1329.8	1340.3	1350.7	1361.1	1371.5
	s	0.5631	1.5298	...	...	...	...	...	...	1.5319	1.5475	1.5619	1.5754	1.5882	1.6003	1.6120	1.6232	1.6339	1.6444	1.6546	1.6644	1.6741	1.6835	1.6927	1.7017
250 (400.95)	v	0.0187	1.8438	...	...	...	...	...	...	...	1.9077	1.9717	2.033	2.093	2.151	2.208	2.264	2.319	2.374	2.427	2.480	2.533	2.585	2.637	2.688
	h	376.0	1201.1	...	...	...	...	...	...	...	1214.2	1227.3	1239.7	1251.7	1263.4	1274.8	1285.9	1296.9	1307.7	1318.5	1329.1	1339.6	1350.1	1360.6	1371.0
	s	0.5675	1.5263	...	...	...	...	...	...	...	1.5414	1.5560	1.5697	1.5826	1.5949	1.6067	1.6179	1.6288	1.6393	1.6495	1.6595	1.6691	1.6786	1.6878	1.6969
260 (404.42)	v	0.0187	1.7748	...	...	...	...	...	...	...	1.8257	1.8882	1.9483	2.006	2.063	2.118	2.172	2.226	2.278	2.330	2.381	2.432	2.482	2.532	2.582
	h	379.8	1201.5	...	...	...	...	...	...	...	1212.4	1225.7	1238.3	1250.5	1262.3	1273.8	1285.0	1296.0	1306.9	1317.7	1328.4	1339.0	1349.5	1360.0	1370.4
	s	0.5719	1.5229	...	...	...	...	...	...	...	1.5354	1.5503	1.5642	1.5773	1.5897	1.6015	1.6129	1.6238	1.6344	1.6447	1.6547	1.6644	1.6739	1.6832	1.6922
270 (407.78)	v	0.0188	1.7107	...	...	...	...	...	...	...	1.7497	1.8108	1.8694	1.9260	1.9809	2.03	2.087	2.139	2.190	2.240	2.289	2.338	2.387	2.436	2.484
	h	383.4	1201.9	...	...	...	...	...	...	...	1210.6	1224.1	1236.9	1249.2	1261.1	1272.7	1284.0	1295.2	1306.1	1317.0	1327.7	1338.3	1348.9	1359.4	1369.9
	s	0.5760	1.5196	...	...	...	...	...	...	...	1.5295	1.5447	1.5588	1.5721	1.5846	1.5965	1.6080	1.6190	1.6297	1.6400	1.6500	1.6598	1.6693	1.6786	1.6877
280 (411.05)	v	0.0188	1.6511	...	...	...	...	...	...	...	1.6789	1.7388	1.7960	1.8512	1.9047	1.9569	2.008	2.058	2.107	2.156	2.204	2.252	2.299	2.346	2.392
	h	387.0	1202.3	...	...	...	...	...	...	...	1208.7	1222.4	1235.4	1247.9	1260.0	1271.7	1283.1	1294.3	1305.3	1316.2	1327.0	1337.7	1348.3	1358.9	1369.4
	s	0.5801	1.5164	...	...	...	...	...	...	...	1.5238	1.5392	1.5536	1.5670	1.5796	1.5917	1.6032	1.6143	1.6251	1.6354	1.6455	1.6553	1.6649	1.6742	1.6834
290 (414.23)	v	0.0188	1.5954	...	...	...	...	...	...	...	1.6130	1.6717	1.7277	1.7816	1.833	1.8846	1.9[illegible]42	1.[illegible]2	2.031	2.078	2.125	2.[illegible]71	2.21	2.262	2.307
	h	390.5	1202.6	...	...	...	...	...	...	...	1206.8	1220.8	1234.0	1246.6	1258.8	1270.6	1282.1	1293.4	1304.5	1315.5	1326.3	1337.0	1347.7	1358.3	1368.8
	s	0.5841	1.5133	...	...	...	...	...	...	...	1.5181	1.5339	1.5484	1.5620	1.5748	1.5870	1.5986	1.6098	1.6206	1.6310	1.6411	1.6510	1.6606	1.6700	1.6791
300 (417.33)	v	0.0189	1.5433	...	...	...	...	...	...	...	1.5513	1.6090	1.6638	1.7165	1.7675	1.8170	1.8654	1.9128	1.9594	2.005	2.051	2.095	2.140	2.184	2.227
	h	393.8	1202.8	...	...	...	...	...	...	...	1204.8	1219.1	1232.5	1245.3	1257.6	1269.5	1281.1	1292.5	1303.7	1314.7	1325.6	1336.4	1347.1	1357.7	1368.3
	s	0.5879	1.5104	...	...	...	...	...	...	...	1.5126	1.5286	1.5434	1.5572	1.5701	1.5824	1.5941	1.6054	1.6163	1.6268	1.6369	1.6468	1.6565	1.6659	1.6751

PSYCHROMETRIC TABLES

Table IV.—Properties of Saturated Water Vapor with Air at Low Temperatures

Temp, °F	Pressure of saturated vapor × 10^5		Weight of saturated vapor				Volume, cu ft barometer, 29.92 in. Hg		Enthalpy per pound		
	In. Hg	Psi	Per cubic feet		Per pound of dry air		Of 1 lb of dry air	Of 1 lb of dry air + vapor to saturate it	Dry air 0°F datum	Vapor 32°F datum	Dry air with vapor to saturate it
			Pounds × 10^5	Grains	Pounds × 10^5	Grains					
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
−25	946.4	464.87	1.8016	0.12611	19.68	1.3776	10.95	10.95	−6.011	1048.0	−5.805
−24	1,003.	492.67	1.9049	0.13334	20.86	1.4602	10.97	10.97	−5.770	1048.4	−5.551
−23	1,064.	522.64	2.0162	0.14113	22.13	1.5491	11.00	11.00	−5.529	1048.9	−5.297
−22	1,126.	553.09	2.1287	0.14901	23.42	1.6394	11.02	11.02	−5.288	1049.3	−5.042
−21	1,192.	585.51	2.2484	0.15739	24.79	1.7353	11.05	11.05	−5.047	1049.8	−4.787
−20	1,262.0	619.89	2.3750	0.16625	26.25	1.8375	11.07	11.07	−4.807	1050.2	−4.531
−19	1,337.	656.73	2.5105	0.17574	27.81	1.9467	11.10	11.10	−4.566	1050.7	−4.274
−18	1,416.	695.54	2.6527	0.18569	29.45	2.0615	11.13	11.13	−4.325	1051.1	−4.015
−17	1,496.	734.84	2.7963	0.19574	31.12	2.1784	11.15	11.1	−4.085	1051.6	−3.758
−16	1,584.	778.06	2.9542	0.20679	32.95	2.3065	11.18	11.18	−3.844	1052.0	−3.497
−15	1,675.0	822.76	3.1168	0.21818	34.84	2.4388	11.20	11.21	−3.604	1052.5	−3.237
−14	1,772.	870.41	3.2899	0.23029	36.86	2.5802	11.23	11.24	−3.363	1052.9	−2.975
−13	1,874.	920.51	3.4714	0.24300	38.98	2.7286	11.25	11.26	−3.123	1053.4	−2.712
−12	1,980.	972.58	3.6596	0.25617	41.19	2.8833	11.28	11.29	−2.883	1053.8	−2.449
−11	2,093.	1,028.1	3.8599	0.27019	43.54	3.0478	11.30	11.31	−2.642	1054.3	−2.183
−10	2,210.0	1,085.6	4.0666	0.28466	45.98	3.2186	11.33	11.34	−2.402	1054.7	−1.917
−9	2,335.	1,147.0	4.2871	0.30009	48.58	3.4006	11.35	11.36	−2.162	1055.2	−1.649
−8	2,463.	1,209.8	4.5120	0.31584	51.25	3.5875	11.38	11.39	−1.921	1055.6	−1.380
−7	2,502.	1,229.0	4.5734	0.32014	52.06	3.6442	11.40	11.41	−1.681	1056.1	−1.131
−6	2,745.	1,348.3	5.0066	0.35046	57.12	3.9984	11.43	11.44	−1.441	1056.5	−0.8375
−5	2,898.0	1,423.5	5.2738	0.36917	60.30	4.2210	11.45	11.46	−1.201	1057.0	−0.5636
−4	3,055.	1,500.6	5.5473	0.38831	63.57	4.4499	11.48	11.49	−0.9604	1057.4	−0.2882
−3	3,222.	1,582.6	5.8379	0.40865	67.05	4.6935	11.50	11.51	−0.7203	1057.9	−0.01098
−2	3,397.	1,668.6	6.1414	0.42990	70.69	4.9483	11.53	11.54	−0.4802	1058.3	+0.2679
−1	3,580.	1,758.5	6.4583	0.45208	74.50	5.2150	11.55	11.57	−0.2401	1058.8	+0.5487
0	3,773.0	1,853.3	6.7914	0.47500	78.52	5.5000	11.58	11.59	0	1059.2	+0.8317

Source: Copyright, A.S.H.V.E.—from "Heating, Ventilating and Air Conditioning Guide," Chap. 1, 1939; compiled by W. M. Sawdon, vapor pressures converted from International Critical Tables.

TABLE V.—PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 TO 200°F

Temp, °F	Pressure of saturated vapor		Weight of saturated vapor				Volume, cu ft barometer, 29.92 in. Hg		Enthalpy per pound		
	In. Hg	Psi	Per cubic feet, Pounds	Per cubic feet, Grains	Per pound of dry air, Pounds	Per pound of dry air, Grains	Of 1 lb of dry air	Of 1 lb of dry air + vapor to saturate it	Dry air 0°F datum	Vapor 32°F datum	Dry air with vapor to saturate it
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
0	0.03773	0.01853	0.000067914	0.475	0.0007852	5.50	11.58	11.59	0.0000	1059.2	0.8317
1	0.03975	0.01963	0.000071395	0.500	0.0008275	5.79	11.60	11.62	0.2401	1059.7	1.117
2	0.04186	0.02056	0.000075021	0.525	0.0008714	6.10	11.63	11.64	0.4801	1060.1	1.404
3	0.04409	0.02166	0 000078851	0.552	0.0009179	6.43	11.65	11.67	0.7201	1060.6	1.694
4	0.04645	0.02282	0.000082890	0.580	0.0009671	6.77	11.68	11.70	0.9601	1061.0	1.986
5	0.04886	0.02400	0.000087005	0.609	0.001017	7.12	11.70	11.72	1.200	1061.5	2.280
6	0.05144	0.02527	0.000091399	0.640	0.001071	7.50	11.73	11.75	1.440	1061.9	2.577
7	0.05412	0.02658	0.000095955	0.672	0.001127	7.89	11.75	11.77	1.680	1062.4	2.877
8	0.05692	0.02796	0.00010070	0.705	0.001186	8.30	11.78	11.80	1.920	1062.8	3.180
9	0.05988	0.02941	0.00010572	0.740	0.001247	8.73	11.80	11.83	2.160	1063.3	3.486
10	0.06295	0.03092	0.00011090	0.776	0.001311	9.18	11.83	11.85	2.400	1063.7	3.795
11	0.06618	0.03251	0.00011634	0.814	0.001379	9.65	11.86	11.88	2.640	1064.2	4.108
12	0.06958	0.03418	0.00012206	0.854	0.001450	10.15	11.88	11.91	2.880	1064.6	4.424
13	0.07309	0.03590	0.00012794	0.896	0.001523	10.66	11.91	11.93	3.120	1065.1	4.742
14	0.07677	0.03771	0.00013410	0.939	0.001600	11.20	11.93	11.96	3.359	1065.5	5.064
15	0.08067	0.03963	0.00014062	0.984	0.001682	11.77	11.96	11.99	3.599	1066.0	5.392
16	0.08469	0.04160	0.00014732	1.031	0.001766	12.36	11.98	12.01	3.839	1066.4	5.722
17	0.08895	0.04369	0.00015440	1.081	0.001855	12.99	12.00	12.04	4.079	1066.9	6.058
18	0.09337	0.04586	0.00016174	1.132	0.001947	13.63	12.03	12.07	4.319	1067.3	6.397
19	0.09797	0.04812	0.00016935	1.185	0.002043	14.30	12.06	12.09	4.559	1067 8	6.741

SOURCE: Copyright, A.S.H.V.E., from "Heating, Ventilating and Air Conditioning Guide," Chap. 1, 1939; compiled by W. M. Sawdon, vapor pressures converted from International Critical Tables.

TABLE V.—PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 TO 200°F (*Cont.*)

Temp, °F	Pressure of saturated vapor		Weight of saturated vapor				Volume, cu ft barometer, 29.92 in. Hg		Enthalpy per pound		
			Per cubic feet		Per pound of dry air						
	In. Hg.	Psi	Pounds	Grains	Pounds	Grains	Of 1 lb of dry air	Of 1 lb of dry air + vapor to saturate it	Dry air 0°F datum	Vapor 32°F datum	Dry air with vapor to saturate it
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
20	0.1028	0.05050	0.00017747	1.242	0.002144	15.01	12.08	12.12	4.798	1068.2	7.088
21	0.1078	0.05295	0.00018564	1.299	0.002250	15.75	12.11	12.15	5.038	1068.7	7.443
22	0.1132	0.05560	0.00019439	1.361	0.002361	16.53	12.13	12.18	5.278	1069.1	7.802
23	0.1186	0.05826	0.00020335	1.423	0.002476	17.33	12.16	12.20	5.518	1069.6	8.166
24	0.1244	0.06111	0.00021276	1.489	0.002596	18.17	12.18	12.23	5.758	1070.0	8.536
25	0.1304	0.06405	0.00022255	1.558	0.002722	19.05	12.21	12.26	5.998	1070.5	8.912
26	0.1366	0.06710	0.00023278	1.629	0.002853	19.97	12.23	12.29	6.237	1070.9	9.292
27	0.1432	0.07034	0.00024342	1.704	0.002991	20.94	12.26	12.32	6.477	1071.4	9.682
28	0.1500	0.07368	0.00025445	1.781	0.003133	21.93	12.28	12.34	6.717	1071.8	10.075
29	0.1571	0.07717	0.00026597	1.862	0.003283	22.99	12.31	12.37	6.957	1072.3	10.477
30	0.1645	0.08080	0.00027797	1.946	0.003439	24.07	12.33	12.40	7.197	1072.7	10.886
31	0.1722	0.08458	0.00029043	2.033	0.003601	25.21	12.36	12.43	7.437	1073.2	11.302
32	0.1803	0.08856	0.00030343	2.124	0.003771	26.40	12.38	12.46	7.677	1073.6	11.726
33	0.1879	0.09230	0.00031471	2.203	0.003931	27.52	12.41	12.49	7.917	1074.1	12.139
34	0.1957	0.09610	0.00032690	2.288	0.004094	28.66	12.43	12.51	8.157	1074.5	12.556
35	0.20360	0.1000	0.0003394	2.376	0.004262	29.83	12.46	12.54	8.397	1075.0	12.979
36	0.21195	0.1041	0.0003527	2.469	0.004438	31.07	12.48	12.57	8.636	1075.4	13.409
37	0.22050	0.1083	0.0003662	2.563	0.004618	32.33	12.51	12.60	8.876	1075.9	13.845
38	0.22925	0.1126	0.0003799	2.660	0.004803	33.62	12.53	12.63	9.116	1076.3	14.285
39	0.23842	0.1171	0.0003943	2.760	0.004996	34.97	12.56	12.66	9.356	1076.8	14.736
40	0.24778	0.1217	0.0004090	2.863	0.005194	36.36	12.59	12.69	9.596	1077.2	15.191
41	0.25755	0.1265	0.0004243	2.970	0.005401	37.80	12.61	12.72	9.836	1077.7	15.657
42	0.26773	0.1315	0.0004401	3.081	0.005616	39.31	12.64	12.75	10.08	1078.1	16.13
43	0.27832	0.1367	0.0004566	3.196	0.005840	40.88	12.66	12.78	10.32	1078.6	16.62
44	0.28911	0.1420	0.0004735	3.315	0.006069	42.48	12.69	12.81	10.56	1079.0	17.11

45	0.30031	0.1475	0.0004909	3.436	0.006306	44.14	12.71	12.84	10.80	1079.5	17.61
46	0.31191	0.1532	0.0005088	3.562	0.006553	45.87	12.74	12.87	11.04	1079.9	18.12
47	0.32393	0.1591	0.0005274	3.692	0.006808	47.66	12.76	12.90	11.28	1080.4	18.64
48	0.33635	0.1652	0.0005465	3.826	0.007072	49.50	12.79	12.93	11.52	1080.8	19.16
49	0.34917	0.1715	0.0005663	3.964	0.007345	51.42	12.81	12.96	11.76	1081.3	19.70
50	0.36241	0.1780	0.0005866	4.106	0.007626	53.38	12.84	12.99	12.00	1081.7	20.25
51	0.37625	0.1848	0.0006078	4.255	0.007921	55.45	12.86	13.02	12.23	1082.2	20.80
52	0.39051	0.1918	0.0006296	4.407	0.008226	57.58	12.89	13.06	12.47	1082.6	21.38
53	0.40496	0.1989	0.0006516	4.561	0.008534	59.74	12.91	13.09	12.71	1083.1	21.95
54	0.42003	0.2063	0.0006746	4.722	0.008856	61.99	12.94	13.12	12.95	1083.5	22.55
55	0.43570	0.2140	0.0006984	4.889	0.009192	64.34	12.96	13.15	13.19	1084.0	23.15
56	0.45179	0.2219	0.0007228	5.060	0.009536	66.75	12.99	13.19	13.43	1084.4	23.77
57	0.46828	0.2300	0.0007477	5.234	0.009890	69.23	13.01	13.22	13.67	1084.9	24.40
58	0.48538	0.2384	0.0007735	5.415	0.01026	71.82	13.04	13.25	13.91	1085.3	25.05
59	0.50310	0.2471	0.0008003	5.602	0.01064	74.48	13.06	13.29	14.15	1085.8	25.70
60	0.52142	0.2561	0.0008278	5.795	0.01103	77.21	13.09	13.32	14.39	1086.2	26.37
61	0.54035	0.2654	0.0008562	5.993	0.01144	80.08	13.11	13.35	14.63	1086.7	27.06
62	0.55970	0.2749	0.0008852	6.196	0.01186	83.02	12.14	13.39	14.87	1087.1	27.76
63	0.57985	0.2848	0.0009153	6.407	0.01229	86.03	13.16	13.42	15.11	1087.6	28.48
64	0.60042	0.2949	0.0009460	6.622	0.01274	89.18	13.19	13.46	15.35	1088.0	29.21
65	0.62179	0.3054	0.0009778	6.845	0.01320	92.40	13.21	13.49	15.59	1088.5	29.96
66	0.64378	0.3162	0.0010105	7.074	0.01368	95.76	13.24	13.53	15.83	1088.9	30.73
67	0.66638	0.3273	0.0010440	7.308	0.01417	99.19	13.26	13.57	16.07	1089.4	31.51
68	0.68980	0.3388	0.0010816	7.571	0.01468	102.8	13.29	13.60	16.31	1089.8	32.31
69	0.71382	0.3506	0.0011140	7.798	0.01520	106.4	13.31	13.64	16.55	1090.3	33.12
70	0.73866	0.3628	0.0011507	8.055	0.01574	110.2	13.34	13.68	16.79	1090.7	33.96
71	0.76431	0.3754	0.0011884	8.319	0.01631	114.2	13.37	13.71	17.03	1091.2	34.83
72	0.79058	0.3883	0.0012269	8.588	0.01688	118.2	13.40	13.75	17.27	1091.6	35.70
73	0.81766	0.4016	0.0012667	8.867	0.01748	122.4	13.42	13.79	17.51	1092.1	36.60
74	0.84555	0.4153	0.0013075	9.153	0.01809	126.6	13.44	13.83	17.75	1092.5	37.51
75	0.87448	0.4295	0.0013497	9.448	0.01873	131.1	13.47	13.87	17.99	1093.0	38.46
76	0.90398	0.4440	0.0013927	9.749	0.01938	135.7	13.49	13.91	18.23	1093.4	39.42
77	0.93452	0.4590	0.0014371	10.06	0.02005	140.4	13.52	13.95	18.47	1093.9	40.40
78	0.96588	0.4744	0.0014825	10.38	0.02075	145.3	13.54	13.99	18.71	1094.3	41.42
79	0.99825	0.4903	0.0015295	10.71	0.02147	150.3	13.57	14.03	18.95	1094.8	42.46

TABLE V.—PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 TO 200°F (*Cont.*)

Temp, °F	Pressure of saturated vapor		Weight of saturated vapor				Volume, cu ft barometer, 29.92 in. Hg		Enthalpy per pound		
	In. Hg	Psi	Per cubic feet		Per pound of dry air		Of 1 lb of dry air	Of 1 lb of dry air + vapor to saturate it	Dry air 0°F datum	Vapor 32°F datum	Dry air with vapor to saturate it
			Pounds (4)	Grains (5)	Pounds (6)	Grains (7)					
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
80	1.0316	0.5067	0.0015777	11.04	0.02221	155.5	13.59	14.08	19.19	1095.2	43.51
81	1.0661	0.5236	0.0016273	11.39	0.02298	160.9	13.62	14.12	19.43	1095.7	44.61
82	1.1013	0.5409	0.0016781	11.75	0.02377	166.4	13.64	14.16	19.67	1096.1	45.72
83	1.1377	0.5588	0.0017304	12.11	0.02459	172.1	13.67	14.21	19.91	1096.6	46.88
84	1.1752	0.5772	0.0017841	12.49	0.02543	178.0	13.69	14.26	20.15	1097.0	48.05
85	1.2135	0.5960	0.0018389	12.87	0.02629	184.0	13.72	14.30	20.39	1097.5	49.24
86	1.2527	0.6153	0.0018950	13.27	0.02718	190.3	13.74	14.34	20.63	1097.9	50.47
87	1.2933	0.6352	0.0019531	13.67	0.02810	196.7	13.77	14.39	20.87	1098.4	51.74
88	1.3346	0.6555	0.0020116	14.08	0.02904	203.3	13.79	14.44	21.11	1098.8	53.02
89	1.3774	0.6765	0.0020725	14.51	0.03002	210.1	13.82	14.48	21.35	1099.3	54.35
90	1.4211	0.6980	0.0021344	14.94	0.03102	217.1	13.84	14.53	21.59	1099.7	55.70
91	1.4661	0.7201	0.0021982	15.39	0.03205	224.4	13.87	14.58	21.83	1100.2	57.09
92	1.5125	0.7429	0.0022634	15.84	0.03312	231.8	13.89	14.63	22.07	1100.6	58.52
93	1.5600	0.7662	0.0023304	16.31	0.03421	239.5	13.92	14.69	22.32	1101.1	59.99
94	1.6088	0.7902	0.0023992	16.79	0.03535	247.5	13.94	14.73	22.56	1101.5	61.50
95	1.6591	0.8149	0.0024697	17.28	0.03652	255.6	13.97	14.79	22.80	1102.0	63.05
96	1.7108	0.8403	0.0025425	17.80	0.03772	264.0	13.99	14.84	23.04	1102.4	64.62
97	1.7638	0.8663	0.0026164	18.31	0.03896	272.7	14.02	14.90	23.28	1102.9	66.25
98	1.8181	0.8930	0.0026925	18.85	0.04024	281.7	14.04	14.95	23.52	1103.3	67.92
99	1.8741	0.9205	0.0027700	19.39	0.04156	290.9	14.07	15.01	23.76	1103.8	69.63
100	1.9316	0.9487	0.0028506	19.95	0.04293	300.5	14.10	15.07	24.00	1104.2	71.40
101	1.9904	0.9776	0.0029316	20.52	0.04433	310.3	14.12	15.12	24.24	1104.7	73.21
102	2.0507	1.0072	0.0030156	21.11	0.04577	320.4	14.15	15.18	24.48	1105.1	75.06
103	2.1128	1.0377	0.0031017	21.71	0.04726	330.8	14.17	15.25	24.72	1105.6	76.97
104	2.1763	1.0689	0.0031887	22.32	0.04879	341.5	14.20	15.31	24.96	1106.0	78.92

105	2.2414	1.1009	0.0032786	22.95	0.05037	352.6	14.22	15.37	25.20	1106.5	80.93
106	2.3084	1.1338	0.0033715	23.60	0.05200	364.0	14.25	15.44	25.44	1106.9	83.00
107	2.3770	1.1675	0.0034650	24.26	0.05368	375.8	14.27	15.50	25.68	1107.4	85.13
108	2.4473	1.2020	0.0035612	24.93	0.05541	387.9	14.30	15.57	25.92	1107.8	87.30
109	2.5196	1.2375	0.0036603	25.62	0.05719	400.3	14.32	15.64	26.16	1108.3	89.54
110	2.5939	1.274	0.0037622	26.34	0.05904	413.3	14.35	15.71	26.40	1108.7	91.86
111	2.6692	1.311	0.0038669	27.07	0.06092	426.4	14.37	15.78	26.64	1109.2	94.21
112	2.7486	1.350	0.0039729	27.81	0.06292	440.4	14.39	15.85	26.88	1109.6	96.70
113	2.8280	1.389	0.0040816	28.57	0.06493	454.5	14.42	15.93	27.12	1110.1	99.20
114	2.9094	1.429	0.0041911	29.34	0.06700	469.0	14.45	16.00	27.36	1110.5	101.76
115	2.9929	1.470	0.0043047	30.13	0.06913	483.9	14.47	16.08	27.60	1111.0	104.40
116	3.0784	1.512	0.0044208	30.95	0.07134	499.4	14.50	16.16	27.84	1111.4	107.13
117	3.1660	1.555	0.0045372	31.76	0.07361	515.3	14.52	16.24	28.08	1111.9	109.92
118	3.2576	1.600	0.0046620	32.63	0.07600	532.0	14.55	16.32	28.32	1112.3	112.85
119	3.3492	1.645	0.0047846	33.49	0.07840	548.8	14.57	16.41	28.56	1112.8	115.80
120	3.4449	1.692	0.0049115	34.38	0.08093	566.5	14.60	16.50	28.80	1113.2	118.89
121	3.5406	1.739	0.005040	35.28	0.08348	584.4	14.62	16.58	29.04	1113.7	122.01
122	3.6404	1.788	0.005173	36.21	0.08616	603.1	14.65	16.68	29.28	1114.1	125.27
123	3.7422	1.838	0.005311	37.18	0.08892	622.4	14.67	16.77	29.52	1114.6	128.63
124	3.8460	1.889	0.005450	38.15	0.09175	642.3	14.70	16.87	29.76	1115.0	132.06
125	3.9519	1.941	0.005590	39.13	0.09466	662.6	14.72	16.96	30.00	1115.5	135.59
126	4.0618	1.995	0.005734	40.14	0.09770	683.9	14.75	17.06	30.24	1115.9	139.26
127	4.1718	2.049	0.005882	41.17	0.1008	705.6	14.77	17.17	30.48	1116.4	143.01
128	4.2858	2.105	0.006031	42.22	0.1040	728.0	14.80	17.27	30.72	1116.8	146.87
129	4.4039	2.163	0.006188	43.32	0.1074	751.8	14.83	17.38	30.96	1117.3	150.96
130	4.5220	2.221	0.006344	44.41	0.1107	774.9	14.85	17.49	31.20	1117.7	154.93
131	4.6441	2.281	0.006504	45.53	0.1143	800.1	14.88	17.61	31.45	1118.2	159.26
132	4.7703	2.343	0.006671	46.70	0.1180	826.0	14.90	17.73	31.69	1118.6	163.68
133	4.8986	2.406	0.006839	47.87	0.1218	852.6	14.93	17.85	31.93	1119.1	168.24
134	5.0289	2.470	0.007010	49.07	0.1257	879.9	14.95	17.97	32.17	1119.5	172.89
135	5.1633	2.536	0.007185	50.30	0.1297	907.9	14.98	18.10	32.41	1120.0	177.67
136	5.2997	2.603	0.007364	51.55	0.1339	937.3	15.00	18.23	32.65	1120.4	182.67
137	5.4402	2.672	2.007547	52.83	0.1382	967.4	15.03	18.36	32.89	1120.9	187.80
138	5.5827	2.742	0.007732	54.12	0.1427	998.9	15.05	18.50	33.13	1121.3	193.14
139	5.7293	2.814	0.007923	55.46	0.1473	1,031.1	15.08	18.65	33.37	1121.8	198.61

TABLE V.—PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 TO 200F° (*Cont.*)

Temp, °F	Pressure of saturated vapor		Weight of saturated vapor				Volume, cu ft barometer, 29.92 in. Hg		Enthalpy per pound		
	In. Hg	Psi	Per cubic feet		Per pound of dry air		Of 1 lb of dry air	Of 1 lb of dry air + vapor to saturate it	Dry air 0°F datum	Vapor 32°F datum	Dry air with vapor to saturate it
			Pounds	Grains	Pounds	Grains					
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
140	5.8779	2.887	0.008116	56.81	0.1521	1,064.7	15.10	18.79	33.61	1122.2	204.30
141	6.0306	2.962	0.008313	58.19	0.1570	1,099.0	15.13	18.94	33.85	1122.7	210.11
142	6.1874	3.039	0.008516	59.61	0.1622	1,135.4	15.15	19.10	34.09	1123.1	216.26
143	6.3482	3.118	0.008724	61.07	0.1675	1,172.5	15.18	19.26	34.33	1123.6	222.53
144	6.5111	3.198	0.008933	62.53	0.1730	1,211.0	15.20	19.43	34.57	1124.0	229.02
145	6.6781	3.280	0.009148	64.04	0.1787	1,250.9	15.23	19.60	34.81	1124.5	235.76
146	6.8471	3.363	0.009366	65.56	0.1846	1,292.2	15.25	19.78	35.05	1124.9	242.71
147	7.0222	3.449	0.009590	67.13	0.1908	1,335.6	15.28	19.96	35.29	1125.4	250.02
148	7.1993	3.536	0.009817	68.72	0.1971	1,379.7	15.30	20.15	35.53	1125.8	257.43
149	7.3805	3.625	0.010040	70.28	0.2037	1,425.9	15.33	20.35	35.77	1126.3	265.20
150	7.5658	3.716	0.010284	71.99	0.2105	1,473.5	15.35	20.55	36.02	1126.7	273.19
151	7.7551	3.809	0.010526	73.68	0.2176	1,523.2	15.38	20.76	36.26	1127.2	281.54
152	7.9485	3.904	0.010772	75.40	0.2250	1,575.0	15.40	20.97	36.50	1127.6	290.21
153	8.1460	4.001	0.011022	77.15	0.2327	1,628.9	15.43	21.20	36.74	1128.1	299.25
154	8.3476	4.100	0.011279	78.95	0.2407	1,684.9	15.45	21.43	36.98	1128.5	308.61
155	8.5532	4.201	0.011539	80.77	0.2490	1,743.0	15.48	21.67	37.22	1129.0	318.34
156	8.7650	4.305	0.011807	82.65	0.2577	1,803.9	15.50	21.93	37.46	1129.4	328.51
157	8.9788	4.410	0.012077	84.54	0.2667	1,866.9	15.53	22.19	37.70	1129.9	339.04
158	9.1986	4.518	0.012354	86.48	0.2761	1,932.7	15.56	22.46	37.94	1130.3	350.02
159	9.4206	4.627	0.012634	88.44	0.2858	2,000.6	15.58	22.74	38.18	1130.8	361.36
160	9.6486	4.739	0.012919	90.43	0.2961	2,072.7	15.61	23.03	38.43	1131.2	373.38
161	9.8807	4.853	0.013211	92.48	0.3067	2,146.9	15.63	23.33	38.67	1131.7	385.76
162	10.119	4.970	0.013509	94.56	0.3179	2,225.3	15.66	23.65	38.91	1132.1	398.80
163	10.361	5.089	0.013812	96.68	0.3295	2,306.5	15.68	23.98	39.15	1132.5	412.34
164	10.608	5.210	0.014120	98.84	0.3416	2,391.2	15.71	24.33	39.39	1133.0	426.42

165	10.860	5.334	0.014434	101.0	0.3544	2,480.8	15.73	24.69	39.63	1133.5	441.34
166	11.177	5.460	0.014753	103.3	0.3677	2,573.9	15.76	25.07	39.87	1133.9	456.81
167	11.379	5.589	0.015080	105.6	0.3817	2,671.9	15.78	25.46	40.11	1134.4	473.11
168	11.646	5.720	0.015410	107.9	0.3964	2,774.8	15.81	25.88	40.35	1134.8	490.18
169	11.919	5.854	0.015750	110.3	0.4118	2,882.6	15.83	26.31	40.59	1135.3	508.11
170	12.196	5.990	0.016092	112.6	0.4280	2,996.0	15.86	26 77	40.83	1135.7	526.91
171	12.480	6.130	0.016444	115.1	0.4451	3,115.7	15.88	27.24	41.07	1136.2	546.79
172	12.770	6.272	0.016801	117.6	0.4631	3,241.7	15.91	27.74	41.32	1136.6	567.68
173	13.065	6.417	0.017164	120.1	0.4821	3,374.7	15.93	28.28	41.56	1137.1	589.76
174	13.366	6.565	0.017534	122.7	0.5022	3,515.4	15.96	28.84	41.80	1137.5	613.05
175	13.674	6.716	0.017914	125.4	0.5235	3,664.5	15.98	29.43	42.04	1138.0	637.78
176	13.985	6.869	0.018294	128.1	0.5459	3,821.3	16.01	30.05	42.28	1138.4	663.73
177	14.303	7.025	0.018684	130.8	0.5697	3,987.9	16.03	30.71	42.52	1138.9	691.35
178	14.627	7.184	0.019080	133.6	0.5949	4,164.3	16.06	31.41	42.76	1139.3	720.53
179	14.954	7.345	0.019477	136.3	0.6215	4,350.5	16.08	32.15	43.00	1139.8	751.39
180	15.290	7.510	0.019888	139.2	0.6501	4,550.7	16.11	32.94	43.24	1140.2	784.48
181	15.632	7.678	0.020304	142.1	0.6805	4,763.5	16.13	33.78	43.49	1140.7	819.74
182	15.981	7.849	0.020729	145.1	0.7131	4,991.7	16.16	34.68	43.73	1141.1	857.45
183	16.337	8.024	0.021159	148.1	0.7481	5,236.7	16.18	35.65	43.97	1141.6	898.00
184	16.697	8.201	0.021598	151.2	0.7854	5,497.8	16.21	36.67	44.21	1142.0	941.14
185	17.066	8.382	0.022045	154.3	0.8258	5,780.6	16.23	37.78	44.45	1142.5	987.93
186	17.440	8.566	0.022497	157.5	0.8693	6,085.1	16.26	38.98	44.69	1142.9	1038.21
187	17.821	8.753	0.022956	160.7	0.9162	6,413.4	16.28	40.27	44.93	1143.4	1092.51
188	18.210	8.944	0.023424	164.0	0.9673	6,771.1	16.31	41.67	45.18	1143.8	1151.58
189	18.605	9.138	0.023900	167.3	1.0227	7,158.9	16.34	43.04	45.42	1144.3	1216.04
190	19.008	9.336	0.024384	170.7	1.083	7,581.0	16.36	44.85	45.66	1144.7	1285.37
191	19.419	9.538	0.025881	174.2	1.150	8,050.0	16.39	46.68	45.90	1145.2	1362.88
192	19.839	9.744	0.025380	177.7	1.224	8,568.0	16.41	48.70	46.14	1145.6	1448.35
193	20.266	9.954	0.025893	181.3	1.306	9,142.0	16.44	50.93	46.38	1146.1	1543.19
194	20.702	10.168	0.026413	184.9	1.397	9,779.0	16.46	53.42	46.62	1146.5	1648.28
195	21.144	10.385	0.026939	188.6	1.499	10,493 0	16.49	56.20	46.86	1147.0	1766.21
196	21.592	10.605	0.027472	192.3	1.613	11,291 0	16.51	59.31	47.10	1147 4	1897.86
197	22.048	10.829	0.028019	196.1	1.742	12,194 0	16.54	62.85	47.34	1147.9	2046.98
198	22.512	11.057	0.028571	200.0	1.890	13,230 0	16.56	66.88	47.59	1148.3	2217.88
199	22.984	11.289	0.029129	203.9	2.061	14,427.0	16.59	71.54	47.83	1148.8	2415.51
200	23.465	11.525	0.029700	207.9	2.261	15,827.0	16.61	76.99	48.07	1149.2	2646.41

AMMONIA

Table VI.—Toxic Properties of Gases

(Underwriters' Laboratory, M. H. 2375.)

Refrigerant	Kills or seriously injures guinea pigs, % by volume	Time, min	Poisonous decomposition products	Underwriters laboratory class
Ammonia	0.5 - 0.6	1- 30	—	2
Butane	—	—	—	5
Carbon dioxide	30	30- 60	—	5
Carbon tetrachloride	0.2 - 2.3	60	Yes	3
Dichloroethylene	2.0 - 2.5	120	Yes	4
Ethyl bromide	2.0 - 2.5	120	Yes	4
Ethyl chloride	6.0 -10.0	30- 60	Yes	5
Freon 11	10	120	Yes	5
Freon 12	30	120	Yes	6
Freon 113	....	....	Yes	5
Freon 114	30	120	Yes	6
Methyl bromide	0.7 - 1.0	30	Yes	2
Methyl chloride	2.0 - 2.5	120	Yes	4
Methyl formate	2.0 - 2.5	60	—	3
Sulphur dioxide	0.7	5	—	1

Source: Reproduced from the "Refrigerating Data Book," by permission of the American Society of Refrigerating Engineers.

Table VII.—Refrigerant Combustion Data

Refrigerant	Heat of combustion with air, gross Btu at 64°F per cu ft	Heat of formation	Volume, cu ft per lb at 64 °F	Combustion volumes, cu ft of gas				
				Air required, cu ft	Products of combustion			
					CO_2	H_2O	Cl_2	N_2
Ammonia	432	57	22.15	3.58	...	1.5	...	3.33
Butane	3335	208	6.38	31.04	4.0	5.0	...	24.54
Ethane	1764	136	12.59	16.71	2.0	3.0	...	13.21
Ethyl chloride	1790	160	5.36	15.52	2.0	2.5	0.50	12 27
Methyl chloride	851	109	7.41	8.36	1.0	1.5	0.50	6.61
Propane	2557	170	8.45	23.88	3.0	4.0	...	18.88

Source: Reproduced from the "Refrigerating Data Book," by permission of the American Society of Refrigerating Engineers.

TABLE VIII.—REFRIGERANT EXPLOSION DATA, WITH COMPARATIVE GASES

Refrigerant	Relative rates of diffusion (air = 1)	Gas required for complete combustion, %	Apparent ignition temperature, °F	Explosion limits with air, %		Maximum explosion pressures with air, psi,	Time required to develop maximum pressure, sec
				Low	High		
Ammonia....	1.301	21.83	....	13.1	26.8	54	0.252
Butane......	0.705	3.12	806	1.7	5.7	102	0.027
Dichlorethylene.......	.....	....	....	5.6	11.4	76	0.095
Ethane......	0.977	5.65	950	3.1	10.7	108	0.018
Ethyl chloride	0.658	6.05	....	4.3	14.0	98	0.049
Gasoline.....	0.577	Varies	536	1.4	6.0	100	0.026
Illuminating gas........	1.240	Varies	1094	7.0	21.0	95	0.017
Methyl chloride.......	0.750	10.69	....	8.9	15.5	81	0.099
Methyl formate.... .	.....	.....	....	45.	20.	96	0.026
Propane	0.814	4.02	871	2.4	8.4	104	0.020

SOURCE: Reproduced from the "Refrigerating Data Book," by permission of the American Society of Refrigerating Engineers.

TABLE IX.—DESIGN CHARACTERISTICS OF REFRIGERANTS

Refrigerant	(1) Displacement per standard ton, cfm	(2) Weight circulated per standard ton, lb per min	(3) Refrigerating effect at 5 to 86°F, Btu per lb	(4) Differential heat of liquid, 86 to 5°F	(5) Latent heat at 5°	(6) Ratio, col. (5) to col (4)	(7) Standard cycle, hp per ton	(8) Coefficient of performance	(9) Efficiency, % (Carnot)
1. Ammonia	3.44	0.422	474.5	90.6	565.0	6.23	0.989	4.77	84.5
2. Carbon dioxide	0.943	3.528	56.69	62.0	117.0	1.89	1.843	2.56	44.6
3. Dichloroethylene	.....	.....	.....	21.7	136.1	6.27	0.918	5.14	89.4
4. Ethyl chloride	.....	.....	.....	34.7	177.6	5.12	.....	.....	.....
5. Freon 11	36.33	2.961	67.54	16.46	84.0	5.10	.....	.....	.....
6. Freon 12	5.82	3.916	51.07	18.4	69.5	3.77	.....	.....	.....
7. Freon 21	20.43	2.237	89.41	19.93	109.34	5.48	.....	.....	.....
8. Freon 113	100.75	3.672	53.67	16.95	70.62	4.11	.....	.....	.....
9. Freon 114	19.37	4.589	43.58	18.88	61.98	3.29	.....	.....	.....
10. Isobutane	11.50	1.794	111.5	48.0	159.5	3.32	.....	.....	.....
11. Methyl chloride	6.09	1.345	148.7	30.3	179.0	5.91	0.997	4.72	82.2
12. Methyl formate	49.9	1.056	189.2	41.7	231.0	5.54	.....	.....	.....
13. Methylene chloride	74.45	1.492	134.1	27.5	162.1	5.89	0.965	4.9	85.3
14. Sulphur dioxide	9.08	1.414	141.4	25.9	169.0	6.53	0.935	5.05	88.0
15. Water	.....	.....	.....	.....	.....	.....	1.15	4.10	71.5
16. Carnot cycle	.....	.....	.....	.....	.....	.....	0.8214	5.74	100.0

TABLE X.—COMPRESSION RATIO COMPARISONS

Refrigerant	5°F suction		86°F condenser		40°F suction		100°F condenser	
	Suction, psi	Head, psi	Difference, psi	Ratio, head/suction	Suction, psi	Head, psi	Difference, psi	Ratio, head/suction
Freon 113	0.98	7.86	6.88	8.02	2.66	10.48	7.82	3.93
Methylene chloride	1.17	10.60	9.43	9.07	3.38	13.25	9.87	3.92
Methyl formate	1.75	13.81	12.06	7.90	7.032	23.76	16.728	3.37
Freon 11	2.93	18.28	15.35	6.24	7.02	23.60	16.58	3.36
Ethyl chloride	4.65	27.10	22.45	5.83	10.79	34.79	24.00	3.22
Freon 21	5.24	30.5	24.26	5.83	12.32	40.04	27.72	3.25
Butane	8.2	41.6	33.4	5.07	17.7	52.2	34.5	2.95
Isobutane	13.1	59.5	46.4	4.54	26.9	73.7	46.8	2.74
Sulphur dioxide	11.81	66.45	54.64	5.63	27.10	84.52	57.42	3.12
Methyl chloride	20.8	95.53	74.73	4.57	42.61	119.04	76.43	2.79
Freon 12	26.51	107.9	81.39	4.07	51.68	131.6	79.92	2.54
Ammonia	34.27	169.2	134.93	4.94	73.32	211.9	138.58	2.89
Propane	42.1	155.3	113.2	3.69	78.0	187.0	109.0	2.40
CO_2	331.9	1043.	711.1	3.16	.....	.....	.....	.....

Table XI.—Saturated Ammonia (NH_3)

Temp, °F	Pressure, psia, p	Volume: Liquid, cu ft per lb, v_f	Volume: Vapor, cu ft per lb, v_g	Total heat above —40°: Liquid, Btu per lb, h_f	Total heat above —40°: Latent, Btu per lb, h	Total heat above —40°: Vapor, Btu per lb, h_g	Entropy from —40°: Liquid, s_f	Entropy from —40°: Vapor, s_g
—60	5.55	0.02278	44.73	—21.2	610.8	589.6	—0.0517	1.4769
—59	5.74	0.02280	43.37	—20.1	610.1	590.0	—0.0490	1.4741
—58	5.93	0.02282	42.05	—19.1	609.5	590.4	—0.0464	1.4713
—57	6.13	0.02284	40.79	—18.0	608.8	590.8	—0.0438	1.4686
—56	6.33	0.02286	39.56	—17.0	608.2	591.2	—0.0412	1.4658
—55	6.54	0.02288	38.38	—15.9	607.5	591.6	—0.0386	1.4631
—54	6.75	0.02291	37.24	—14.8	606.9	592.1	—0.0360	1.4604
—53	6.97	0.02293	36.15	—13.8	606.2	592.4	—0.0334	1.4577
—52	7.20	0.02295	35.09	—12.7	605.6	592.9	—0.0307	1.4551
—51	7.43	0.02297	34.06	—11.7	604.9	593.2	—0.0281	1.4524
—50	7.67	0.02299	33.08	—10.6	604.3	593.7	—0.0256	1.4497
—49	7.91	0.02301	32.12	—9.6	603.6	594.0	—0.0230	1.4471
—48	8.16	0.02304	31.20	—8.5	602.9	594.4	—0.0204	1.4445
—47	8.42	0.02306	30.31	—7.4	602.3	594.9	—0.0179	1.4419
—46	8.68	0.02308	29.45	—6.4	601.6	595.2	—0.0153	1.4393
—45	8.95	0.02310	28.62	—5.3	600.9	595.6	—0.0127	1.4368
—44	9.23	0.02313	27.82	—4.3	600.3	596.0	—0.0102	1.4342
—43	9.51	0.02315	27.04	—3.2	599.6	596.4	—0.0076	1.4317
—42	9.81	0.02317	26.29	—2.1	598.9	596.8	—0.0051	1.4292
—41	10.10	0.02319	25.56	—1.1	598.3	597.2	—0.0025	1.4267
—40	10.41	0.02322	24.86	0.0	597.6	597.6	0.0000	1.4242
—39	10.72	0.02324	24.18	1.1	596.9	598.0	0.0025	1.4217
—38	11.04	0.02326	23.53	2.1	596.2	598.3	0.0051	1.4193
—37	11.37	0.02328	22.89	3.2	595.5	598.7	0.0076	1.4169
—36	11.71	0.02331	22.27	4.3	594.8	599.1	0.0101	1.4144
—35	12.05	0.02333	21.68	5.3	594.2	599.5	0.0126	1.4120
—34	12.41	0.02335	21.10	6.4	593.5	599.9	0.0151	1.4096
—33	12.77	0.02338	20.54	7.4	592.8	600.2	0.0176	1.4072
—32	13.14	0.02340	20.00	8.5	592.1	600.6	0.0201	1.4048
—31	13.52	0.02342	19.48	9.6	591.4	601.0	0.0226	1.4025
—30	13.90	0.02345	18.97	10.7	590.7	601.4	0.0250	1.4001
—29	14.30	0.02347	18.48	11.7	590.0	601.7	0.0275	1.3978
—28	14.71	0.02349	18.00	12.8	589.3	602.1	0.0300	1.3955
—27	15.12	0.02352	17.54	13.9	588.6	602.5	0.0325	1.3932
—26	15.55	0.02354	17.09	14.9	587.9	602.8	0.0350	1.3909
—25	15.98	0.02357	16.66	16.0	587.2	603.2	0.0374	1.3886
—24	16.42	0.02359	16.24	17.1	586.5	603.6	0.0399	1.3863
—23	16.88	0.02361	15.83	18.1	585.8	603.9	0.0423	1.3840
—22	17.34	0.02364	15.43	19.2	585.1	604.3	0.0448	1.3818
—21	17.81	0.02366	15.05	20.3	584.3	604.6	0.0472	1.3796
—20	18.30	0.02369	14.68	21.4	583.6	605.0	0.0497	1.3774
—19	18.79	0.02371	14.32	22.4	582.9	605.3	0.0521	1.3752
—18	19.30	0.02374	13.97	23.5	582.2	605.7	0.0545	1.3729
—17	19.81	0.02376	13.62	24.6	581.5	606.1	0.0570	1.3708
—16	20.34	0.02378	13.29	25.6	580.8	606.4	0.0594	1.3686
—15	20.88	0.02381	12.97	26.7	580.0	606.7	0.0618	1.3664
—14	21.43	0.02383	12.66	27.8	579.3	607.1	0.0642	1.3643
—13	21.99	0.02386	12.36	28.9	578.6	607.5	0.0666	1.3621
—12	22.56	0.02388	12.06	30.0	577.8	607.8	0.0690	1.3600
—11	23.15	0.02391	11.78	31.0	577.1	608.1	0.0714	1.3579
—10	23.74	0.02393	11.50	32.1	576.4	608.5	0.0738	1.3558
—9	24.35	0.02396	11.23	33.2	575.6	608.8	0.0762	1.3537
—8	24.97	0.02399	10.97	34.3	574.9	609.2	0.0786	1.3516
—7	25.61	0.02401	10.71	35.4	574.1	609.5	0.0809	1.3495
—6	26.26	0.02404	10.47	36.4	573.4	609.8	0.0833	1.3474
—5	26.92	0.02406	10.23	37.5	572.6	610.1	0.0857	1.3454
—4	27.59	0.02409	9.991	38.6	571.9	610.5	0.0880	1.3433
—3	28.28	0.02411	9.763	39.7	571.1	610.8	0.0904	1.3413
—2	28.98	0.02414	9.541	40.7	570.4	611.1	0.0928	1.3393
—1	29.69	0.02417	9.326	41.8	569.6	611.4	0.0951	1.3372

TABLE XI.—SATURATED AMMONIA (NH_3) (*Cont.*)

Temp, °F, t	Pressure, psia, p	Volume		Total heat above −40°			Entropy from −40°	
		Liquid, cu ft per lb, v_f	Vapor, cu ft per lb, v_g	Liquid, Btu per lb, h_f	Latent, Btu per lb, h	Vapor, Btu per lb, h_g	Liquid, s_f	Vapor, s_g
0	30.42	0.02419	9.116	42.9	568.9	611.8	0.0975	1.3352
1	31.16	0.02422	8.912	44.0	568.1	612.1	0.0998	1.3332
2	31.92	0.02424	8.714	45.1	567.3	612.4	0.1022	1.3312
3	32.69	0.02427	8.521	46.2	566.5	612.7	0.1045	1.3292
4	33.47	0.02430	8.333	47.2	565.8	613.0	0.1069	1.3273
5	34.27	0.02432	8.150	48.3	565.0	613.3	0.1092	1.3253
6	35.09	0.02435	7.971	49.4	564.2	613.6	0.1115	1.3234
7	35.92	0.02438	7.798	50.5	563.4	613.9	0.1138	1.3214
8	36.77	0.02440	7.629	51.6	562.7	614.3	0.1162	1.3195
9	37.63	0.02443	7.464	52.7	561.9	614.6	0.1185	1.3176
10	38.51	0.02446	7.304	53.8	561.1	614.9	0.1208	1.3157
11	39.40	0.02449	7.148	54.9	560.3	615.2	0.1231	1.3137
12	40.31	0.02451	6.996	56.0	559.5	615.5	0.1254	1.3118
13	41.24	0.02454	6.847	57.1	558.7	615.8	0.1277	1.3099
14	42.18	0.02457	6.703	58.2	557.9	616.1	0.1300	1.3081
15	43.14	0.02460	6.562	59.2	557.1	616.3	0.1323	1.3062
16	44.12	0.02462	6.425	60.3	556.3	616.6	0.1346	1.3043
17	45.12	0.02465	6.291	61.4	555.5	616.9	0.1369	1.3025
18	46.13	0.02468	6.161	62.5	554.7	617.2	0.1392	1.3006
19	47.16	0.02471	6.034	63.6	553.9	617.5	0.1415	1.2988
20	48.21	0.02474	5.910	64.7	553.1	617.8	0.1437	1.2969
21	49.28	0.02476	5.789	65.8	552.2	618.0	0.1460	1.2951
22	50.36	0.02479	5.671	66.9	551.4	618.3	0.1483	1.2933
23	51.47	0.02482	5.556	68.0	550.6	618.6	0.1505	1.2915
24	52.59	0.02485	5.443	69.1	549.8	618.9	0.1528	1.2897
25	53.73	0.02488	5.334	70.2	548.9	619.1	0.1551	1.2879
26	54.90	0.02491	5.227	71.3	548.1	619.4	0.1573	1.2861
27	56.08	0.02494	5.123	72.4	547.3	619.7	0.1596	1.2843
28	57.28	0.02497	5.021	73.5	546.4	619.9	0.1618	1.2825
29	58.50	0.02500	4.922	74.6	545.6	620.2	0.1641	1.2808
30	59.74	0.02503	4.825	75.7	544.8	620.5	0.1663	1.2790
31	61.00	0.02505	4.730	76.8	543.9	620.7	0.1686	1.2773
32	62.29	0.02508	4.637	77.9	543.1	621.0	0.1708	1.2755
33	63.59	0.02511	4.547	79.0	542.2	621.2	0.1730	1.2738
34	64.91	0.02514	4.459	80.1	541.4	621.5	0.1753	1.2721
35	66.26	0.02518	4.373	81.2	540.5	621.7	0.1775	1.2704
36	67.63	0.02521	4.289	82.3	539.7	622.0	0.1797	1.2686
37	69.02	0.02524	4.207	83.4	538.8	622.2	0.1819	1.2669
38	70.43	0.02527	4.126	84.6	537.9	622.5	0.1841	1.2652
39	71.87	0.02530	4.048	85.7	537.0	622.7	0.1863	1.2635
40	73.32	0.02533	3.971	86.8	536.2	623.0	0.1885	1.2618
41	74.80	0.02536	3.897	87.9	535.3	623.2	0.1908	1.2602
42	76.31	0.02539	3.823	89.0	534.4	623.4	0.1930	1.2585
43	77.83	0.02542	3.752	90.1	533.6	623.7	0.1952	1.2568
44	79.38	0.02545	3.682	91.2	532.7	623.9	0.1974	1.2552
45	80.96	0.02548	3.614	92.3	531.8	624.1	0.1996	1.2535
46	82.55	0.02551	3.547	93.5	530.9	624.4	0.2018	1.2519
47	84.18	0.02555	3.481	94.6	530.0	624.6	0.2040	1.2502
48	85.82	0.02558	3.418	95.7	529.1	624.8	0.2062	1.2486
49	87.49	0.02561	3.355	96.8	528.2	625.0	0.2083	1.2469
50	89.19	0.02564	3.294	97.9	527.3	625.2	0.2105	1.2453
51	90.91	0.02568	3.234	99.1	526.4	625.5	0.2127	1.2437
52	92.66	0.02571	3.176	100.2	525.5	625.7	0.2149	1.2421
53	94.43	0.02574	3.119	101.3	524.6	625.9	0.2171	1.2405
54	96.23	0.02577	3.063	102.4	523.7	626.1	0.2192	1.2389
55	98.06	0.02581	3.008	103.5	522.8	626.3	0.2214	1.2373
56	99.91	0.02584	2.954	104.7	521.8	626.5	0.2236	1.2357
57	101.8	0.02587	2.902	105.8	520.9	626.7	0.2257	1.2341
58	103.7	0.02590	2.851	106.9	520.0	626.9	0.2279	1.2325
59	105.6	0.02594	2.800	108.1	519.0	627.1	0.2301	1.2310

TABLE XI.—SATURATED AMMONIA (NH_3) (*Concl.*)

Temp, °F, t	Pressure, psia, p	Volume		Total heat above —40°			Entropy from —40°	
		Liquid, cu ft per lb, v_f	Vapor, cu ft per lb, v_g	Liquid, Btu per lb, h_f	Latent, Btu per lb, h	Vapor, Btu per lb, h_g	Liquid, s_f	Vapor, s_g
60	107.6	0.02587	2.751	109.2	518.1	627.3	0.2322	1.2294
61	109.6	0.02601	2.703	110.3	517.2	627.5	0.2344	1.2278
62	111.6	0.02604	2.656	111.5	516.2	627.7	0.2365	1.2262
63	113.6	0.02608	2.610	112.6	515.3	627.9	0.2387	1.2247
64	115.7	0.02611	2.565	113.7	514.3	628.0	0.2408	1.2231
65	117.8	0.02614	2.520	114.8	513.4	628.2	0.2430	1.2216
66	120.0	0.02618	2.477	116.0	512.4	628.4	0.2451	1.2201
67	122.1	0.02621	2.435	117.1	511.5	628.6	0.2473	1.2186
68	124.3	0.02625	2.393	118.3	510.5	628.8	0.2494	1.2170
69	126.5	0.02628	2.352	119.4	509.5	628.9	0.2515	1.2155
70	128.8	0.02632	2.312	120.5	508.6	629.1	0.2537	1.2140
71	131.1	0.02635	2.273	121.7	507.6	629.3	0.2558	1.2125
72	133.4	0.02639	2.235	122.8	506.6	629.4	0.2579	1.2110
73	135.7	0.02643	2.197	124.0	505.6	629.6	0.2601	1.2095
74	138.1	0.02646	2.161	125.1	504.7	629.8	0.2622	1.2080
75	140.5	0.02650	2.125	126.2	503.7	629.9	0.2643	1.2065
76	143.0	0.02653	2.089	127.4	502.7	630.1	0.2664	1.2050
77	145.4	0.02657	2.055	128.5	501.7	630.2	0.2685	1.2035
78	147.9	0.02661	2.021	129.7	500.7	630.4	0.2706	1.2020
79	150.5	0.02664	1.988	130.8	499.7	630.5	0.2728	1.2006
80	153.0	0.02668	1.955	132.0	498.7	630.7	0.2749	1.2991
81	155.6	0.02672	1.923	133.1	497.7	630.8	0.2769	1.1976
82	158.3	0.02675	1.892	134.3	496.7	631.0	0.2791	1.1962
83	161.0	0.02679	1.861	135.4	495.7	631.1	0.2812	1.1947
84	163.7	0.02684	1.831	136.6	494.7	631.3	0.2833	1.1933
85	166.4	0.02687	1.801	137.8	493.6	631.4	0.2854	1.1918
86	169.2	0.02691	1.772	138.9	492.6	631.5	0.2875	1.1904
87	172.0	0.02695	1.744	140.1	491.6	631.7	0.2895	1.1889
88	174.8	0.02699	1.716	141.2	490.6	631.8	0.2917	1.1875
89	177.7	0.02703	1.688	142.4	489.5	631.9	0.2937	1.1860
90	180.6	0.02707	1.661	143.5	488.5	632.0	0.2958	1.1846
91	183.6	0.02711	1.635	144.7	487.4	632.1	0.2979	1.1832
92	186.6	0.02715	1.609	145.8	486.4	632.2	0.3000	1.1818
93	189.6	0.02719	1.584	147.0	485.3	632.3	0.3021	1.1804
94	192.7	0.02723	1.559	148.2	484.3	632.5	0.3041	1.1789
95	195.8	0.02727	1.534	149.4	483.2	632.6	0.3062	1.1775
96	198.9	0.02731	1.510	150.5	482.1	632.6	0.3083	1.1761
97	202.1	0.02735	1.487	151.7	481.1	632.8	0.3104	1.1747
98	205.3	0.02739	1.464	152.9	480.0	632.9	0.3125	1.1733
99	208.6	0.02743	1.441	154.0	478.9	632.9	0.3145	1.1719
100	211.9	0.02747	1.419	155.2	477.8	633.0	0.3166	1.1705
101	215.2	0.02752	1.397	156.4	476.7	633.1	0.3187	1.1691
102	218.6	0.02756	1.375	157.6	475.6	633.2	0.3207	1.1677
103	222.0	0.02760	1.354	158.7	474.6	633.3	0.3228	1.1663
104	225.4	0.02764	1.334	159.9	473.5	633.4	0.3248	1.1649
105	228.9	0.02769	1.313	161.1	472.3	633.4	0.3269	1.1635
106	232.5	0.02773	1.293	162.3	471.2	633.5	0.3289	1.1621
107	236.0	0.02777	1.274	163.5	470.1	633.6	0.3310	1.1607
108	239.7	0.02782	1.254	164.6	469.0	633.6	0.3330	1.1593
109	243.3	0.02786	1.235	165.8	467.9	633.7	0.3351	1.1580
110	247.0	0.02790	1.217	167.0	466.7	633.7	0.3372	1.1566
112	254.5	0.02799	1.180	169.4	464.4	633.8	0.3413	1.1538
114	262.2	0.02808	1.145	171.8	642.1	633.9	0.3453	1.1510
116	270.1	0.02817	1.112	174.2	459.8	634.0	0.3495	1.1483
118	278.2	0.02827	1.079	176.6	457.4	634.0	0.3535	1. 455
120	286.4	0.02836	1.047	179.0	455.0	634.0	0.3576	1.1427
122	294.8	0.02846	1.017	181.4	452.6	634.0	0.3618	1.1400
124	303.4	0.02855	0.987	183.9	450.1	634.0	0.2659	1.1372
126	312.2	0.02865	0.958	186.3	447.6	633.9	0.3700	1.1344
128	321.2	0.02875	0.931	188.8	445.1	633.9	0.3741	1.1316

TABLE XII.—SUPERHEATED AMMONIA

Temp., °F, t	5 psi (−63.11°F)			10 psi (−41.34°F)			15 psi (−27.29°F)			20 psi (−16.64°F)		
	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s
(At saturation)	*(49.31)*	*(588.3)*	*(1.4857)*	*(25.81)*	*(597.1)*	*(1.4276)*	*(17.67)*	*(602.4)*	*(1.3938)*	*(13.50)*	*(606.2)*	*(1.3700)*
−50	51.05	595.2	1.5025									
−40	52.36	600.3	1.5149	25.90	597.8	1.4293						
−30	53.67	605.4	1.5269	26.58	603.2	1.4420						
−20	54.97	610.4	1.5385	27.26	608.5	1.4542	18.01	606.4	1.4031			
−10	56.26	615.4	1.5498	27.92	613.7	1.4659	18.47	611.9	1.4154	13.74	610.0	1.3784
0	57.55	620.4	1.5608	28.58	618.9	1.4773	18.92	617.2	1.4272	14.09	615.5	1.3907
10	58.84	625.4	1.5716	29.24	624.0	1.4884	19.37	622.5	1.4386	14.44	621.0	1.4025
20	60.12	630.4	1.5821	29.90	629.1	1.4992	19.82	627.8	1.4497	14.78	626.4	1.4138
30	61.41	635.4	1.5925	30.55	634.2	1.5097	20.26	633.0	1.4604	15.11	631.7	1.4248
40	62.69	640.4	1.6026	31.20	639.3	1.5200	20.70	638.2	1.4709	15.45	637.0	1.4356
50	63.96	645.5	1.6125	31.85	644.4	1.5301	21.14	643.4	1.4812	15.78	642.3	1.4460
60	65.24	650.5	1.6223	32.49	649.5	1.5400	21.58	648.5	1.4912	16.12	647.5	1.4562
70	66.51	655.5	1.6319	33.14	654.6	1.5497	22.01	653.7	1.5011	16.45	652.8	1.4662
80	67.79	660.6	1.6413	33.78	659.7	1.5593	22.44	658.9	1.5108	16.78	658.0	1.4760
90	69.06	665.6	1.6506	34.42	664.8	1.5687	22.88	664.0	1.5203	17.10	663.2	1.4856
100	70.33	670.7	1.6598	35.07	670.0	1.5779	23.31	669.2	1.5296	17.43	668.5	1.4950
110	71.60	675.8	1.6689	35.71	675.1	1.5870	23.74	674.4	1.5388	17.76	673.7	1.5042
120	72.87	680.9	1.6778	36.35	680.3	1.5960	24.17	679.6	1.5478	18.08	678.9	1.5133
130	74.14	686.1	1.6865	36.99	685.4	1.6049	24.60	684.8	1.5567	18.41	684.2	1.5223
140	75.41	691.2	1.6952	37.62	690.6	1.6136	25.03	690.0	1.5655	18.73	689.4	1.5312
150	76.68	696.4	1.7038	38.26	695.8	1.6222	25.46	695.3	1.5742	19.05	694.7	1.5399
160	77.95	701.6	1.7122	38.90	701.1	1.6307	25.88	700.5	1.5827	19.37	700.0	1.5485
170	79.21	706.8	1.7206	39.54	706.3	1.6391	26.31	705.8	1.5911	19.70	705.3	1.5569
180	80.48	712.1	1.7289	40.17	711.6	1.6474	26.74	711.1	1.5995	20.02	710.6	1.5653
190	.....	.....	.....	40.81	716.9	1.6556	27.16	716.4	1.6077	20.34	715.9	1.5736
200	.....	.....	.....	.....	.....	.....	27.59	721.7	1.6158	20.66	721.2	1.5817
210	.....	.....	.....	.....	.....	.....	28.02	727.0	1.6239	20.98	726.6	1.5898
220	.....	.....	.....	.....	.....	.....	28.44	732.4	1.6318	21.30	732.0	1.5978
230	.....	.....	.....	.....	.....	.....	28.88	737.8	1.6397	21.62	737.4	1.6057
240	.....	.....	.....	.....	.....	.....	29.29	743.2	1.6475	21.94	742.8	1.6135
250	.....	.....	.....	.....	.....	.....	29.71	748.6	1.6552	22.26	748.3	1.6212

TABLE XII.—SUPERHEATED AMMONIA (Cont.)

Temp, °F, t	25 psi (−7.96°F)			30 psi (−0.57°F)			35 psi (5.89°F)			40 psi (11.66°F)		
	Volume, cu ft per lb, v	Total heat Btu per lb, h	Entropy, Btu per lb, °F, s	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s
(At saturation)	(10.96)	(609.1)	(1.3515)	(9.236)	(611.6)	(1.3364)	(7.991)	(613.6)	(1.3236)	(7.047)	(615.4)	(1.3125)
0	11.19	613.8	1.3616	9.250	611.9	1.3371						
10	11.47	619.4	1.3738	9.492	617.8	1.3497	8.078	616.1	1.3289			
20	11.75	625.0	1.3855	9.731	623.5	1.3618	8.287	622.0	1.3413	7.203	620.4	1.3231
30	12.03	630.4	1.3967	9.966	629.1	1.3733	8.493	627.7	1.3532	7.387	626.3	1.3353
40	12.30	635.8	1.4077	10.20	634.6	1.3845	8.695	633.4	1.3646	7.568	632.1	1.3470
50	12.57	641.2	1.4183	10.43	640.1	1.3953	8.895	638.9	1.3756	7.746	637.8	1.3583
60	12.84	646.5	1.4287	10.65	645.5	1.4059	9.093	644.4	1.3863	7.922	643.4	1.3692
70	13.11	651.8	1.4388	10.88	650.9	1.4161	9.289	649.9	1.3967	8.096	648.9	1.3797
80	13.37	657.1	1.4487	11.10	656.2	1.4261	9.484	655.3	1.4069	8.268	654.4	1.3900
90	13.64	662.4	1.4584	11.33	661.6	1.4359	9.677	660.7	1.4168	8.439	659.9	1.4000
100	13.90	667.7	1.4679	11.55	666.9	1.4456	9.869	666.1	1.4265	8.609	665.3	1.4098
110	14.17	673.0	1.4772	11.77	672.2	1.4550	10.06	671.5	1.4360	8.777	670.7	1.4194
120	14.43	678.2	1.4864	11.99	677.5	1.4642	10.25	676.8	1.4453	8.945	676.1	1.4288
130	14.69	683.5	1.4954	12.21	682.9	1.4733	10.44	682.2	1.4545	9.112	681.5	1.4381
140	14.95	688.8	1.5043	12.43	688.2	1.4823	10.63	687.6	1.4635	9.278	686.9	1.4471
150	15.21	694.1	1.5131	12.65	693.5	1.4911	10.82	692.9	1.4724	9.444	692.3	1.4561
160	15.47	699.4	1.5217	12.87	698.8	1.4998	11.00	698.3	1.4811	9.609	697.7	1.4648
170	15.73	704.7	1.5303	13.08	704.2	1.5083	11.19	703.7	1.4897	9.774	703.1	1.4735
180	15.99	710.1	1.5387	13.30	709.6	1.5168	11.38	709.1	1.4982	9.938	708.5	1.4820
190	16.25	715.4	1.5470	13.52	714.9	1.5251	11.56	714.5	1.5066	10.10	714.0	1.4904
200	16.50	720.8	1.5552	13.73	720.3	1.5334	11.75	719.9	1.5148	10.27	719.4	1.4987
210	16.76	726.2	1.5633	13.95	725.7	1.5415	11.94	725.3	1.5230	10.43	724.9	1.5069
220	17.02	731.6	1.5713	14.16	731.3	1.5495	12.12	730.7	1.5311	10.59	730.3	1.5150
230	17.27	737.0	1.5792	14.38	736.6	1.5575	12.31	736.2	1.5390	10.75	735.8	1.5230
240	17.53	742.5	1.5870	14.59	742.0	1.5653	12.49	741.7	1.5469	10.92	741.3	1.5309
250	17.79	747.9	1.5948	14.81	747.5	1.5732	12.68	747.2	1.5547	11.08	746.8	1.5387
260	18.04	753.4	1.6025	15.02	753.0	1.5808	12.86	752.7	1.5624	11.24	752.3	1.5465
270	18.30	758.9	1.6101	15.23	758.5	1.5884	13.04	758.2	1.5701	11.40	757.8	1.5541

Temp, °F	50 psi (21.67°F)			60 psi (30.21°F)			70 psi (37.70°F)			80 psi (44.40°F)		
(*At saturation*)	(*5.710*)	(*618.2*)	(*1.2939*)	(*4.805*)	(*620.5*)	(*1.2787*)	(*4.151*)	(*622.4*)	(*1.2658*)	(*3.655*)	(*624.0*)	(*1.2345*)
30	5.838	623.4	1.3046									
40	5.988	629.5	1.3169	4.933	626.8	1.2913	4.177	623.9	1.2688			
50	6.135	635.4	1.3286	5.060	632.9	1.3035	4.290	630.4	1.2816	3.712	627.7	1.2619
60	6.280	641.2	1.3399	5.184	639.0	1.3152	4.401	636.6	1.2937	3.812	634.3	1.2745
70	6.423	646.9	1.3508	5.307	644.9	1.3265	4.509	642.7	1.3054	3.909	640.6	1.2866
80	6.564	652.6	1.3613	5.428	650.7	1.3373	4.615	648.7	1.3166	4.005	646.7	1.2981
90	6.704	658.2	1.3716	5.547	656.4	1.3479	4.719	654.6	1.3274	4.098	652.8	1.3092
100	6.843	663.7	1.3816	5.665	662.1	1.3581	4.822	660.4	1.3378	4.190	658.7	1.3199
110	6.980	669.2	1.3914	5.781	667.7	1.3681	4.924	666.1	1.3480	4.281	664.6	1.3303
120	7.117	674.7	1.4009	5.897	673.3	1.3778	5.025	671.8	1.3579	4.371	670.4	1.3404
130	7.252	680.2	1.4103	6.012	678.9	1.3873	5.125	677.5	1.3676	4.460	676.1	1.3502
140	7.387	685.7	1.4195	6.126	684.4	1.3966	5.224	683.1	1.3770	4.548	681.8	1.3598
150	7.521	691.1	1.4286	6.239	689.9	1.4058	5.323	688.7	1.3863	4.635	687.5	1.3692
160	7.655	696.6	1.4374	6.352	695.5	1.4148	5.420	694.3	1.3954	4.722	693.2	1.3784
170	7.788	702.1	1.4462	6.464	701.0	1.4236	5.518	699.9	1.4043	4.808	698.8	1.3874
180	7.921	707.5	1.4548	6.576	706.5	1.4323	5.615	705.5	1.4131	4.893	704.4	1.3963
190	8.053	713.0	1.4633	6.687	712.0	1.4409	5.711	711.0	1.4210	4.978	710.0	1.4050
200	8.185	718.5	1.4716	6.798	717.5	1.4493	5.807	716.6	1.4302	5.063	715.6	1.4136
210	8.317	724.0	1.4799	6.909	723.1	1.4576	5.902	722.2	1.4386	5.147	721.3	1.4220
220	8.448	729.4	1.4880	7.019	728.6	1.4658	5.998	727.7	1.4469	5.231	726.9	1.4304
230	8.579	735.0	1.4961	7.129	734.1	1.4739	6.093	733.3	1.4550	5.315	732.5	1.4386
240	8.710	740.5	1.5040	7.238	739.7	1.4819	6.187	738.9	1.4631	5.398	738.1	1.4467
250	8.840	746.0	1.5119	7.348	745.3	1.4898	6.281	744.5	1.4711	5.482	743.8	1.4547
260	8.970	751.6	1.5197	7.457	750.9	1.4976	6.376	750.1	1.4789	5.565	749.4	1.4626
270	9.100	757.2	1.5274	7.566	756.5	1.5053	6.470	755.8	1.4866	5.647	755.1	1.4704
280	9.230	762.7	1.5350	7.675	762.1	1.5130	6.563	761.4	1.4943	5.730	760.7	1.4781
290	9.360	768.4	1.5425	7.783	767.7	1.5206	6.657	767.1	1.5019	5.812	766.4	1.4857
300	9.489	774.0	1.5500	7.892	773.3	1.5281	6.750	772.7	1.5095	5.894	772.1	1.4933
310	9.618	779.6	1.5574	8.000	779.0	1.5355	6.844	778.4	1.5169	5.976	777.8	1.5008

TABLE XII.—SUPERHEATED AMMONIA (*Cont.*)

Temp, °F, *t*	90 psi (50.47°F)			100 psi (56.05°F)			110 psi (61.21°F)			120 psi (66.02°F)		
	Volume, cu ft per lb, *v*	Total heat, Btu per lb, *h*	Entropy, Btu per lb, °F, *s*	Volume, cu ft per lb, *v*	Total heat, Btu per lb, *h*	Entropy, Btu per lb, °F, *s*	Volume, cu ft per lb, *v*	Total heat, Btu per lb, *h*	Entropy, Btu per lb, °F, *s*	Volume, cu ft per lb, *v*	Total heat, Btu per lb, *h*	Entropy, Btu per lb, °F, *s*
(*At saturation*)	(*3.366*)	(*625.3*)	(*1.2445*)	(*2.952*)	(*626.5*)	(*1.2356*)	(*2.693*)	(*627.5*)	(*1.2275*)	(*2.476*)	(*628.4*)	(*1.2201*)
70	3.442	638.3	1.2695	3.068	636.0	1.2539	2.761	633.7	1.2392	2.505	631.3	1.2255
80	3.529	644.7	1.2814	3.149	642.6	1.2661	2.837	640.5	1.2519	2.576	638.3	1.2386
90	3.614	650.9	1.2928	3.227	649.0	1.2778	2.910	647.0	1.2640	2.645	645.0	1.2510
100	3.698	657.0	1.3088	3.304	655.2	1.2891	2.981	653.4	1.2755	2.712	651.6	1.2628
110	3.780	663.0	1.3144	3.380	661.3	1.2999	3.051	659.7	1.2866	2.778	658.0	1.2741
120	3.862	668.9	1.3247	3.454	667.3	1.3104	3.120	665.8	1.2972	2.842	664.2	1.2850
130	3.942	674.7	1.3347	3.527	673.3	1.3206	3.188	671.9	1.3076	2.905	670.4	1.2956
140	4.021	680.5	1.3444	3.600	679.2	1.3305	3.255	677.8	1.3176	2.967	676.5	1.3058
150	4.100	686.3	1.3539	3.672	685.0	1.3401	3.321	683.7	1.3274	3.029	682.5	1.3157
160	4.178	692.0	1.3633	3.743	690.8	1.3495	3.386	689.6	1.3370	3.089	688.4	1.3254
170	4.255	697.7	1.3724	3.813	696.6	1.3588	3.451	695.4	1.3463	3.149	694.3	1.3348
180	4.332	703.4	1.3813	3.883	702.3	1.3678	3.515	701.2	1.3555	3.209	700.2	1.3441
190	4.408	709.0	1.3901	3.952	708.0	1.3767	3.579	707.0	1.3644	3.268	706.0	1.3531
200	4.484	714.7	1.3988	4.021	713.7	1.3854	3.642	712.8	1.3732	3.326	711.8	1.3620
210	4.560	720.4	1.4073	4.090	719.4	1.3940	3.705	718.5	1.3819	3.385	717.6	1.3707
220	4.635	726.0	1.4157	4.158	725.1	1.4024	3.768	724.3	1.3904	3.442	723.4	1.3793
230	4.710	731.7	1.4239	4.226	730.8	1.4108	3.830	730.0	1.3988	3.500	729.2	1.3877
240	4.785	737.3	1.4321	4.294	736.5	1.4190	3.892	735.7	1.4070	3.557	734.9	1.3960
250	4.859	743.0	1.4401	4.361	742.2	1.4271	3.954	741.5	1.4151	3.614	740.7	1.4042
260	4.933	748.7	1.4481	4.428	747.9	1.4350	4.015	747.2	1.4232	3.671	746.5	1.4123
270	5.007	754.4	1.4559	4.495	753.6	1.4429	4.076	752.9	1.4311	3.727	752.2	1.4202
280	5.081	760.0	1.4637	4.562	759.4	1.4507	4.137	758.7	1.4389	3.783	758.0	1.4281
290	5.155	765.8	1.4713	4.629	765.1	1.4584	4.198	764.5	1.4466	3.839	763.8	1.4359
300	5.228	771.5	1.4789	4.695	770.8	1.4660	4.259	770.2	1.4543	3.895	769.6	1.4435
310	5.301	777.2	1.4864	4.761	776.6	1.4736	4.319	776.0	1.4619	3.951	775.4	1.4511
320	5.374	783.0	1.4938	4.827	782.4	1.4810	4.379	781.8	1.4693	4.006	781.2	1.4586
330	....	....	....	4.893	788.2	1.4884	4.439	787.6	1.4767	4.061	787.0	1.4660
340	....	....	....	4.959	794.0	1.4957	4.500	793.4	1.4841	4.117	792.9	1.4734
350	....	....	....	5.024	799.8	1.5029	4.559	799.3	1.4859	4.172	798.7	1.4807

Temp, °F	130 psi (70.53°F)			140 psi (74.79°F)			150 psi (78.81°F)			160 psi (82.64°F)		
(At saturation)	*(2.291)*	*(629.2)*	*(1.2132)*	*(2.132)*	*(629.9)*	*(1.2068)*	*(1.994)*	*(630.5)*	*(1.2009)*	*(1.872)*	*(631.1)*	*(1.1962)*
90	2.421	643.0	1.2388	2.228	640.9	1.2272	2.061	638.8	1.2161	1.914	636.6	1.2055
100	2.484	649.7	1.2509	2.288	647.8	1.2396	2.118	645.9	1.2289	1.969	643.9	1.2186
110	2.546	656.3	1.2625	2.347	654.5	1.2515	2.174	652.8	1.2410	2.023	651.0	1.2311
120	2.606	662.7	1.2736	2.404	661.1	1.2628	2.228	659.4	1.2526	2.075	657.8	1.2429
130	2.665	668.9	1.2843	2.460	667.4	1.2738	2.281	665.9	1.2638	2.125	664.4	1.2542
140	2.724	675.1	1.2947	2.515	673.7	1.2843	2.334	672.3	1.2745	2.175	670.9	1.2652
150	2.781	681.2	1.3048	2.569	679.9	1.2945	2.385	678.6	1.2849	2.224	677.2	1.2757
160	2.838	687.2	1.3146	2.622	686.0	1.3045	2.435	684.8	1.2949	2.272	683.5	1.2859
170	2.894	693.2	1.3241	2.675	692.0	1.3141	2.485	690.9	1.3047	2.319	689.7	1.2958
180	2.949	699.1	1.3335	2.727	698.0	1.3236	2.534	696.9	1.3142	2.365	695.8	1.3054
190	3.004	705.0	1.3426	2.779	704.0	1.3328	2.583	702.9	1.3236	2.411	701.9	1.3148
200	3.059	710.9	1.3516	2.830	709.9	1.3418	2.631	708.9	1.3327	2.457	707.9	1.3240
210	3.113	716.7	1.3604	2.880	715.8	1.3507	2.679	714.8	1.3416	2.502	713.9	1.3331
220	3.167	722.5	1.3690	2.931	721.6	1.3594	2.726	720.7	1.3504	2.547	719.9	1.3419
230	3.220	728.3	1.3775	2.981	727.5	1.3679	2.773	726.6	1.3590	2.591	725.8	1.3506
240	3.273	734.1	1.3858	3.030	733.3	1.3763	2.820	732.5	1.3675	2.635	731.7	1.3591
250	3.326	739.9	1.3941	3.080	739.2	1.3846	2.866	738.4	1.3758	2.679	737.6	1.3675
260	3.379	745.7	1.4022	3.129	745.0	1.3928	2.912	744.3	1.3840	2.723	743.5	1.3757
270	3.431	751.5	1.4102	3.179	750.8	1.4008	2.958	750.1	1.3921	2.766	749.4	1.3838
280	3.483	757.3	1.4181	3.227	756.7	1.4088	3.004	756.0	1.4001	2.809	755.3	1.3919
290	3.535	763.1	1.4259	3.275	762.5	1.4166	3.049	761.8	1.4079	2.852	761.2	1.3998
300	3.587	769.0	1.4336	3.323	768.3	1.4243	3.095	767.7	1.4157	2.895	767.1	1.4076
310	3.639	774.8	1.4412	3.371	774.2	1.4320	3.140	773.6	1.4234	2.937	773.0	1.4153
320	3.690	780.6	1.4487	3.420	780.0	1.4395	3.185	779.4	1.4310	2.980	778.9	1.4229
330	3.742	786.5	1.4562	3.467	785.9	1.4470	3.230	785.3	1.4385	3.022	734.8	1.4304
340	3.793	792.3	1.4636	3.515	791.8	1.4544	3.274	791.2	1.4459	3.064	790.7	1.4379
350	3.844	798.2	1.4709	3.563	797.7	1.4617	3.319	797.1	1.4532	3.106	796.6	1.4452
360	....	....	....	3.610	803.6	1.4690	3.364	803.0	1.4605	3.148	802.5	1.4525
370	....	....	....	....	....	....	....	....	....	3.189	808.5	1.4597
380	....	....	....	....	....	....	....	....	....	3.231	814.5	1.4669
390	....	....	....	....	....	....	....	....	....	3.273	820.4	1.4740
400	...	....	....	....	....	....	....	....	....	3.314	826.4	1.4810

TABLE XII.—SUPERHEATED AMMONIA (*Concl.*)

Temp, °F	170 psi (86.29°F)			180 psi (89.78°F)			190 psi (93.13°F)			200 psi (96.34°F)		
	Volume, cu ft per lb, *v*	Total heat, Btu per lb, *h*	Entropy Btu per lb. °F, *s*	Volume cu ft per lb, *v*	Total heat, Btu per lb, *h*	Entropy Btu per lb. °F, *s*	Volume cu ft per lb, *v*	Total heat, Btu per lb, *h*	Entropy Btu per lb, °F, *s*	Volume, cu ft per lb, *v*	Total heat, Btu per lb, *h*	Entropy Btu per lb, °F, *s*
(At saturation)	(*1.764*)	(*631.6*)	(*1.1900*)	(*1.667*)	(*632.0*)	(*1.1850*)	(*1.581*)	(*632.4*)	(*1.1802*)	(*1.502*)	(*632.7*)	(*1.1756*)
100	1.837	641.9	1.2087	1.720	639.9	1.1992	1.615	637.8	1.1899	1.520	635.6	1.1809
110	1.889	649.1	1.2215	1.770	647.3	1.2123	1.663	645.4	1.2034	1.567	643.4	1.1947
120	1.939	656.1	1.2336	1.818	654.4	1.2247	1.710	652.6	1.2160	1.612	650.9	1.2077
130	1.988	662.8	1.2452	1.865	661.3	1.2364	1.755	659.7	1.2281	1.656	658.1	1.2200
140	2.035	669.4	1.2563	1.910	668.0	1.2477	1.799	666.5	1.2396	1.698	665.0	1.2317
150	2.081	675.9	1.2669	1.955	674.6	1.2586	1.842	673.2	1.2506	1.740	671.8	1.2429
160	2.127	682.3	1.2773	1.999	681.0	1.2691	1.884	679.7	1.2612	1.780	678.4	1.2537
170	2.172	688.5	1.2873	2.042	687.3	1.2792	1.925	686.1	1.2715	1.820	684.9	1.2641
180	2.216	694.7	1.2971	2.084	693.6	1.2891	1.966	692.5	1.2815	1.859	691.3	1.2742
190	2.260	700.8	1.3066	2.126	699.8	1.2987	2.005	698.7	1.2912	1.897	697.7	1.2840
200	2.303	706.9	1.3159	2.167	705.9	1.3081	2.045	704.9	1.3007	1.935	703.9	1.2935
210	2.346	713.0	1.3249	2.208	712.0	1.3172	2.084	711.1	1.3099	1.972	710.1	1.3029
220	2.389	719.0	1.3338	2.248	718.1	1.3262	2.123	717.2	1.3189	2.009	716.3	1.3120
230	2.431	724.9	1.3426	2.288	724.1	1.3350	2.161	723.2	1.3278	2.046	722.4	1.3209
240	2.473	730.9	1.3512	2.328	730.1	1.3436	2.199	729.3	1.3365	2.082	728.4	1.3296
250	2.514	736.8	1.3596	2.367	736.1	1.3521	2.236	735.3	1.3450	2.118	734.5	1.3382
260	2.555	742.8	1.3679	2.407	742.0	1.3605	2.274	741.3	1.3534	2.154	740.5	1.3467
270	2.596	748.7	1.3761	2.446	748.0	1.3687	2.311	747.3	1.3617	2.189	746.5	1.3550
280	2.637	754.6	1.3841	2.484	753.9	1.3768	2.348	753.2	1.3098	2.225	752.5	1.3631
290	2.678	760.5	1.3921	2.523	759.9	1.3847	2.384	759.2	1.3778	2.260	758.5	1.3712
300	2.718	766.4	1.3999	2.561	765.8	1.3926	2.421	765.2	1.3857	2.295	764.5	1.3791
310	2.758	772.3	1.4076	2.599	771.7	1.4004	2.457	771.1	1.3935	2.329	770.5	1.3869
320	2.798	778.3	1.4153	2.637	777.7	1.4081	2.493	777.1	1.4012	2.364	776.5	1.3947
330	2.838	784.2	1.4228	2.675	783.6	1.4156	2.529	783.1	1.4088	2.398	782.5	1.4023
340	2.878	790.1	1.4303	2.713	789.6	1.4231	2.565	789.0	1.4163	2.432	788.5	1.4099
350	2.918	796.2	1.4377	2.750	795.6	1.4305	2.601	795.1	1.4238	2.466	794.5	1.4173
360	2.957	802.0	1.4450	2.788	801.5	1.4379	2.637	801.0	1.4311	2.500	800.5	1.4241
370	2.997	808.0	1.4522	2.825	807.5	1.4451	2 672	807.0	1.4384	2.534	806.5	1.4320
380	3.036	814.0	1.4594	2.863	813.5	1.4523	2 707	813.0	1.4456	2.568	812.5	1.4392
390	3.075	820.0	1.4665	2.900	819.5	1.4594	2.743	819.0	1.4527	2.601	818.6	1.4464
400	3.114	826.0	1.4735	2.937	825.5	1.4665	2.778	825.1	1.4598	2.635	824.6	1.4534

TABLE XIII.—SOLID AND SATURATED CARBON DIOXIDE (CO_2)

Temp, °F, t	Pressure, psia p	Volume		Total heat from —40°			Entropy from 32° plus 1.0	
		Solid, cu ft per lb, v_f	Vapor, cu ft per lb, v_g	Solid, Btu per lb, h_f	Latent, Btu per lb, h	Vapor, Btu per lb, h_g	Solid or liquid, Btu per lb, °F, s_f	Vapor, Btu per lb, °F, s_g
—140	3.18	0.01008	24.320	—121.5	250.7	129.2	0.6065	1.3908
—130	5.39	0.01012	14 740	—118.8	249.4	130.6	0.6150	1.3718
—120	8.90	0.01018	9.179	—116.0	248.0	132.0	0.6232	1.3636
—110	14.22	0.01024	5.848	—113.2	246.4	133.2	0.6314	1.3563
—109.3	14.70	0.01025	5.665	—113.0	246.3	133.3	0.6318	1.3373
—105	17.82	0.01028	4.703	—111.6	245.4	133.8	0.6359	1.3281
—100	22.22	0.01032	3.804	—110.1	244.4	134.3	0.6403	1.3199
—95	27.54	0.01035	3.093	—108.5	243.2	134.7	0.6449	1.3119
—90	33.98	0.01040	2.525	—106.7	241.8	135.1	0.6499	1.3033
—85	41.67	0.01044	2.070	—104.6	240.1	135.5	0.6551	1.2960
—80	50.85	0.01048	1.700	—102.5	238.2	135.7	0.6607	1.2881
—79	52.89	0 01049	1.636	—102.0	237.7	135.7	0.6618	1.2866
—78	55.00	0.01050	1.575	—101.5	237.3	135.8	0.6630	1.2850
—77	57.19	0.01051	1.516	—101.1	236.9	135.8	0.6642	1.2836
—76	59.45	0.01052	1.460	—100.5	236.3	135.8	0.6655	1.2819
—75	61.79	0.01053	1.407	—100.2	236.0	135.8	0.6665	1.2802
—74	64.21	0.01054	1.356	—99.8	235.6	135.8	0.6677	1.2788
—73	66.72	0.01055	1.306	—99.3	235.1	135.8	0.6688	1.2772
—72	69.33	0.01057	1.257	—98.9	234.8	135.9	0.6699	1.2757
—71	72.03	0.01058	1.209	—98.4	234.3	135.9	0.6712	1.2741
—70	74.82	0.01059	1.162	—98.0	233.9	135.9	0.6724	1.2726
—69.9	75.10	0.01059	1.157	—97.9	233.8	135.9	0.6725	1.2724
—69.9	75.10	0.01360	1.1570	—13.7	149.6	135.9	0.8885	1.2724
—62	90.5	0.01379	0.9686	—10.1	146.6	136.5	0.8975	1.2662
—60	94.7	0.01384	0.9270	—9.2	145.8	136.6	0.8997	1.2647
—58	99.1	0.01389	0.8875	—8.3	145.0	136.7	0.9020	1.2632
—56	103.7	0.01393	0.8502	—7.4	144.2	136.8	0.9042	1.2617
—52	113.2	0.01404	0.7812	—5.6	142.7	137.1	0.9087	1.2587
—50	118.2	0.01409	0.7492	—4.7	141.9	137.2	0.9110	1.2572
—48	123.4	0.01414	0.7188	—3.8	141.1	137.3	0.9131	1.2558
—46	128.7	0.01420	0.6899	—2.9	140.3	137.4	0.9153	1.2544
—42	139.9	0.01431	0.6362	—1.0	138.7	137.7	0.9196	1.2516
—40	145.8	0.01437	0.6113	0.00	137.8	137.8	0.9218	1.2503
—38	151.8	0.01442	0.5876	+0.9	136.9	137.8	0.9239	1.2489
—36	158.0	0.01448	0.5649	1.8	136.2	138.0	0.9261	1.2475
—32	171.0	0.01460	0.5227	3.6	134.5	138.1	0.9303	1.2449
—30	177.8	0.01466	0.5029	4.5	133.7	138.2	0.9325	1.2436
—28	184.8	0.01472	0.4841	5.4	132.9	138.3	0.9345	1.2424
—26	192.0	0.01479	0.4661	6.3	132.0	138.3	0.9366	1.2411
—24	199.4	0.01485	0.4480	7.2	131.2	138.4	0.9387	1.2398
—22	207.0	0.01491	0.4325	8.1	130.3	138.4	0.9408	1.2385
—20	214.9	0.01498	0.4168	9.1	129.4	138.5	0.9430	1.2372
—18	223.0	0.01504	0.4092	10.1	128.5	138.6	0.9449	1.2358
—16	231.2	0.01511	0.3872	11.1	127.6	138.7	0.9470	1.2345
—14	239.6	0.01518	0.3666	12.0	126.7	138.7	0.9491	1.2333
—12	248.3	0.01525	0.3600	12.9	125.8	138.7	0.9511	1.2321

TABLE XIII.—SATURATED CARBON DIOXIDE (CO_2) (*Concl.*)

Temp, °F, t	Pressure, psia, p	Volume		Total heat from −40°			Entropy from 32° plus 1	
		Liquid, cu ft per lb, v_f	Vapor, cu ft per lb, v_g	Liquid, Btu per lb, h_f	Latent, Btu per lb, h	Vapor, Btu per lb, h_g	Liquid, Btu per lb, °F, s_f	Vapor, Btu per lb, °F, s_g
−10	257.3	0.01532	0.3472	13.9	124.8	138.7	0.9532	1.2303
−8	266.5	0.01540	0.3349	14.9	123.9	138.8	0.9552	1.2297
−6	275.9	0.01547	0.3231	15.9	122.9	138.8	0.9573	1.2284
−4	285.4	0.01555	0.3118	16.9	122.0	138.9	0.9594	1.2272
−2	295.3	0.01563	0.3009	17.9	121.0	138.9	0.9614	1.2259
0	305.5	0.01570	0.2904	18.8	120.1	138.9	0.9636	1.2247
2	315.9	0.01579	0.2803	19.8	119.0	138.8	0.9657	1.2235
4	326.5	0.01588	0.2707	20.8	118.0	138.8	0.9679	1.2224
6	337.4	0.01596	0.2614	21.8	116.9	138.7	0.9701	1.2212
8	348.7	0.01605	0.2526	22.9	115.8	138.7	0.9722	1.2199
10	360.2	0.01614	0.2437	24.0	114.7	138.7	0.9744	1.2188
12	371.9	0.01623	0.2354	25.0	113.6	138.6	0.9765	1.2175
14	383.9	0 01632	0.2274	26.1	112.5	138.6	0.9787	1.2163
16	396.2	0.01642	0.2197	27.2	111.3	138.5	0.9810	1.2151
18	408.9	0.01652	0.2121	28.3	110.1	138.4	0.9833	1.2139
20	421.8	0.01663	0.2049	29.4	108.9	138.3	0.9856	1.2127
22	434.0	0.01673	0.1979	30.5	107.7	138.2	0.9879	1.2115
24	448.4	0.01684	0.1912	31.7	106.4	138.1	0.9902	1.2103
26	462.2	0.01695	0.1846	32.9	105.1	138.0	0.9927	1.2091
28	476.3	0.01707	0.1783	34.1	103.8	137.9	0.9951	1.2079
30	490.8	0.01719	0.1722	35.4	102.4	137.8	0.9976	1.2067
32	505.5	0.01731	0.1663	36.7	101.0	137.7	1.0000	1.2055
34	522.6	0.01744	0.1603	37.9	99.5	137.4	1.0023	1.2038
36	536.0	0.01759	0.1550	39.1	98.1	137.2	1.0046	1.2024
38	551.7	0.01773	0.1496	40.4	96.5	136.9	1.0071	1.2009
40	567.8	0.01787	0.1444	41.7	95.0	136.7	1.0092	1.1994
42	584.3	0.01801	0.1393	42.9	93.4	136.3	1.0115	1.1979
44	601.1	0.01817	0.1344	44.3	91.8	136.1	1.0140	1.1964
46	618.2	0.01834	0.1297	45.6	90.1	135.7	1.0166	1.1949
48	635.7	0.01851	0.1250	47.0	88.4	135.4	1.0194	1.1933
50	653.6	0.01868	0.1205	48.4	86.6	135.0	1.0218	1.1917
52	671.9	0.01887	0.1161	49.8	84.7	134.5	1.0254	1.1901
54	690.6	0.01906	0.1117	51.2	82.7	133.9	1.0272	1.1882
56	709.5	0.01927	0.1075	52.6	80.8	133.4	1.0283	1.1865
58	728.8	0.01948	0.1034	54.0	78.7	132.7	1.0312	1.1846
60	748.6	0.01970	0.0994	55.5	76.6	132.1	1.0353	1.1826
64	789.4	0.02020	0.0918	58.6	72.0	130.6	1.0410	1.1803
66	810.3	0.02048	0.0880	60.2	69.5	129.7	1.0438	1.1780
68	831.6	0.02079	0.08422	61.9	66.8	128.7	1.0468	1.1754
70	853.4	0.02112	0.08040	63.7	63.8	127.5	1.0500	1.1724
74	898.2	0.02192	0.07269	67.3	57.2	124.5	1.0568	1.1690
76	921.3	0.02242	0.06875	69.4	53.4	122.8	1.0607	1.1655
78	944.8	0.02300	0.06473	71.6	49.3	120.9	1.0649	1.1605
80	968.7	0.02370	0.06064	73.9	44.8	118.7	1.0694	1.1555
82	993.0	0.02456	0.05648	76.4	40.2	116.6	1.0740	1.1505
84	1017.7	0.02553	0.05223	79.4	34.5	113.9	1.0790	1.1455
86	1043.0	0.02686	0.04789	83.3	27.1	110.4	1.0854	1.1381
87.8	1069.4	0.03454	0.03454	97.0	0.0	97.0	1.1098	1.1098

Table XIV.—Saturated Dichlorodifluoromethane (F12-Freon) (CCl_2F_2)

Temp, °F, t	Pressure, psig, p_d	Volume	Density	Total heat from —40°		Entropy from —40°	
		Vapor, cu ft per lb, v_g	Liquid, lb per cu ft, v/v_f	Liquid, Btu per lb, h_f	Vapor, Btu per lb, h_g	Liquid, Btu per lb, °F, s_f	Vapor, Btu per lb, °F, s_g
—40	10.92*	3.911	94.58	0	73.50	0	0.17517
—30	5.45*	3.088	93.59	2.03	74.70	0.00471	0.17387
—20	0.58	2.474	92.58	4.07	75.87	0.00940	0.17275
—10	4.50	2.003	91.57	6.14	77.05	0.01403	0.17175
0	9.17	1.637	90.52	8.25	78.21	0.01869	0.17091
2	10.19	1.574	90.31	8.67	78.44	0.01961	9.17075
4	11.26	1.514	90.11	9.10	78.67	0.02052	0.17060
5	11.81	1.485	90.00	9.32	78.79	0.02097	0.17052
6	12.35	1.457	89.88	9.53	78.90	0.02143	0.17045
8	13.48	1.403	89.68	9.96	79.13	0.02235	0.17030
10	14.65	1.351	89.45	10.39	79.36	0.02328	0.17015
12	15.86	1.301	89.24	10.82	79.59	0.02419	0.17001
14	17.10	1.253	89.03	11.26	79.82	0.02510	0.16987
16	18.38	1.207	88.81	11.70	80.05	0.02601	0.16974
18	19.70	1.163	88.58	12.12	80.27	0.02692	0.16961
20	21.05	1.121	88.37	12.55	80.49	0.02783	0.16949
22	22.45	1.081	88.13	13.00	80.72	0.02873	0.16938
24	23.88	1.043	87.91	13.44	80.95	0.02963	0.16926
26	25.37	1.007	87.68	13.88	81.17	0.03053	0.16913
28	26.89	0.973	87.47	14.32	81.39	0.03143	0.16900
30	28.46	0.939	87.24	14.76	81.61	0.03233	0.16887
32	30.07	0.908	87.02	15.21	81.83	0.03323	0.16876
34	31.72	0.877	86.78	15.65	82.05	0.03413	0.16865
36	33.43	0.848	86.55	16.10	82.27	0.03502	0.16854
38	35.18	0.819	86.33	16.55	82.49	0.03591	0.16843
40	36.98	0.792	86.10	17.00	82.71	0.03680	0.16833
42	38.81	0.767	85.88	17.46	82.93	0.03770	0.16823
44	40.70	0.742	85.66	17.91	83.15	0.03859	0.16813
46	42.65	0.718	85.43	18.36	83.36	0.03948	0.16803
48	44.65	0.695	85.19	18.82	83.57	0.04037	0.16794
50	46.69	0.673	84.94	19.27	83.78	0.04126	0.16785
52	48.79	0.652	84.71	19.72	83.99	0.04215	0.16776
54	50.93	0.632	84.50	20.18	84.20	0.04304	0.16767
56	53.14	0.612	84.28	20.64	84.41	0.04392	0.16758
58	55.40	0.593	84.04	21.11	84.62	0.04480	0.16749
60	57.71	0.575	83.78	21.57	84.82	0.04568	0.16741
62	60.07	0.557	83.57	22.03	85.02	0.04657	0.16733
64	62.50	0.540	83.34	22.49	85.22	0.04745	0.16725
66	64.97	0.524	83.10	22.95	85.42	0.04833	0.16717
68	67.54	0.508	82.86	23.42	85.62	0.04921	0.16709
70	70.12	0.493	82.60	23.90	85.82	0.05009	0.16701
72	72.80	0.479	82.37	24.37	86.02	0.05097	0.16693
74	75.50	0.464	82.12	24.84	86.22	0.05185	0.16685
76	78.30	0.451	81.87	25.32	86.42	0.05272	0.16677
78	81.15	0.438	81.62	25.80	86.61	0.05359	0.16669
80	84.06	0.425	81.39	26.28	86.80	0.05446	0.16662
82	87.00	0.413	81.12	26.76	86.99	0.05534	0.16655
84	90.1	0.401	80.87	27.24	87.18	0.05621	0.16648
86	93.2	0.389	80.63	27.72	87.37	0.05708	0.16640
88	96.4	0.378	80.37	28.21	87.56	0.05795	0.16632
90	99.6	0.368	80.11	28.70	87.74	0.05882	0.16624
92	103.0	0.357	79.86	29.19	87.92	0.05969	0.16616
94	106.3	0.347	79.60	29.68	88.10	0.06056	0.16608
96	109.8	0.338	79.32	30.18	88.28	0.06143	0.16600
98	113.3	0.328	79.06	30.67	88.45	0.06230	0.16592
100	116.9	0.319	78.80	31.16	88.62	0.06316	0.16584
102	120.6	0.310	78.54	31.65	83.79	0.06403	0.16576
104	124.3	0.302	78.27	32.15	88.95	0.06490	0.16568
106	128.1	0.293	78.00	32.65	89.11	0.06577	0.16560
108	132.1	0.285	77.73	33.15	89.27	0.06663	0.16551
110	136.0	0.277	77.46	33.65	89.43	0.06749	0.16542
120	157.1	0.240	76.02	36.16	90.15	0.07180	0.16495
130	180.2	0.208	74.46	38.69	90.76	0.07607	0.16438
140	205.5	0.180	72.73	41.24	91.24	0.08024	0.16363

* Inches of mercury below 1 atm.

TABLE XV.—SUPERHEATED DICHLORODIFLUOROMETHANE (F12)(CCl_2F_2)

Temp, °F t	Abs pressure 20 psi (Saturation temp, −8.2°F)			Abs pressure 30 psi (Saturation temp, 11.1°F)			Abs pressure 40 psi (Saturation temp, 25.9°F)			Abs pressure 50 psi (Saturation temp, 38.3°F)		
	v	h	s	v	h	s	v	h	s	v	h	s
(At saturation)	*(1.925)*	*(77.27)*	*(0.17160)*	*(1.323)*	*(79.47)*	*(0.17008)*	*(1.009)*	*(81.16)*	*(0.16914)*	*(0.817)*	*(82.52)*	*(0.16841)*
50	2.203	85.40	0.18858	1.448	85.03	0.18138	1.070	84.65	0.17612	0.842	84.24	0.17187
60	2.250	86.35	0.19138	1.480	86.48	0.18420	1.095	86.11	0.17896	0.863	85.72	0.17475
70	2.297	88.31	0.19415	1.512	87.95	0.18699	1.120	87.60	0.18178	0.884	87.22	0.17760
80	2.343	89.78	0.19688	1.544	89.43	0.18974	1.144	89.09	0.18455	0.904	88.72	0.18040
90	2.390	91.26	0.19959	1.576	90.91	0.19249	1.169	90.58	0.18731	0.924	90.23	0.18317
100	2.437	92.75	0.20229	1.608	92.41	0.19519	1.194	92.09	0.19004	0.944	91.75	0.18591
110	2.483	94.26	0.20494	1.640	93.93	0.19787	1.218	93.62	0.19272	0.964	93.29	0.18862
120	2.530	95.78	0.20759	1.672	95.46	0.20053	1.242	95.15	0.19538	0.984	94.83	0.19132
130	2.577	97.31	0.21020	1.703	97.00	0.20315	1.267	96.70	0.19803	1.004	96.39	0.19397
140	2.623	98.85	0.21280	1.735	98.54	0.20577	1.291	98.26	0.20066	1.024	97.96	0.19662
150	2.669	100.40	0.21537	1.767	100.11	0.20836	1.315	99.83	0.20325	1.044	99.54	0.19923
160	2.716	101.97	0.21792	1.799	101.69	0.21092	1.340	101.42	0.20583	1.064	101.14	0.20182
170	2.762	103.56	0.22045	1.829	103.28	0.21344	1.364	103.02	0.20838	1.084	102.75	0.20439
180	2.808	105.15	0.22297	1.860	104.88	0.21597	1.388	104.63	0.21092	1.103	104.36	0.20694
190	2.854	106.76	0.22545	1.891	106.49	0.21846	1.412	106.25	0.21343	1.123	105.98	0.20946
200	2.901	108.38	0.22794	1.923	108.12	0.22096	1.435	107.88	0.21592	1.142	107.62	0.21196
210	2.947	110.01	0.23039	1.954	109.76	0.22342	1.459	109.52	0.21840	1.162	109.28	0.21444
220	2.992	111.65	0.23283	1.986	111.41	0.22588	1.482	111.17	0.22085	1.181	110.95	0.21691
230	3.038	113.31	0.23524	2.017	113.08	0.22830	1.506	112.84	0.22329	1.200	112.62	0.21935
240	3.084	114.98	0.23766	2.048	114.75	0.23072	1.530	114.52	0.22572	1.220	114.31	0.22179

Temp, °F	Abs pressure 70 psi (Saturation temp, 57.9°F)			Abs pressure 80 psi (Saturation temp, 66.3°F)			Abs pressure 90 psi (Saturation temp, 73.9°F)			Abs pressure 100 psi (Saturation temp 80.9°F)		
(At saturation)	*(0.594)*	*(84.61)*	*(0.16749)*	*(0.521)*	*(85.45)*	*(0.16716)*	*(0.465)*	*(86.21)*	*(0.16685)*	*(0.419)*	*(86.89)*	*(0.16659)*
100	0.658	91.03	0.17943	0.568	90.68	0.17675	0.499	90.31	0.17433	0.442	89.93	0.17210
110	0.673	92.59	0.18219	0.582	92.26	0.17954	0.511	91.89	0.17713	0.454	91.54	0.17493
120	0.689	94.16	0.18493	0.599	93.84	0.18229	0.523	93.48	0.17990	0.465	93.15	0.17773
130	0.704	95.75	0.18763	0.606	95.43	0.18500	0.535	95.08	0.18262	0.477	94.76	0.18049
140	0.719	97.34	0.19030	0.623	97.03	0.18771	0.547	96.69	0.18533	0.488	96.37	0.18321
150	0.733	98.94	0.19293	0.636	98.64	0.19035	0.559	98.31	0.18799	0.499	97.99	0.18590
160	0.748	100.54	0.19555	0.649	100.26	0.19298	0.571	99.94	0.19065	0.510	99.63	0.18856
170	0.763	102.16	0.19814	0.662	101.88	0.19558	0.584	101.58	0.19327	0.521	101.28	0.19120
180	0.777	103.80	0.20071	0.675	103.52	0.19817	0.596	103.23	0.19588	0.531	102.94	0.19381
190	0.792	105.45	0.20325	0.688	105.18	0.20073	0.607	104.89	0.19845	0.542	104.61	0.19638

200	0.806	107.10	0.20579	0.701	106.84	0.20328	0.619	106.56	0.20101	0.553	106.29	0.19894
210	0.820	108.76	0.20829	0.714	108.51	0.20580	0.630	108.24	0.20353	0.563	107.98	0.20148
220	0.835	110.43	0.21079	0.726	110.19	0.20828	0.642	109.93	0.20603	0.574	109.68	0.20401
230	0.849	112.13	0.21325	0.739	111.88	0.21076	0.653	111.63	0.20852	0.585	111.39	0.20650
240	0.863	113.83	0.21570	0.751	113.58	0.21321	0.665	113.35	0.21100	0.595	113.11	0.20899
250	0.878	115.55	0.21815	0.764	115.30	0.21566	0.676	115.08	0.21345	0.606	114.84	0.21145
260	0.892	117.28	0.22057	0.777	117.03	0.21809	0.688	116.82	0.21589	0.616	116.58	0.21389
270	0.906	119.02	0.22296	0.789	118.78	0.22045	0.699	118.57	0.21831	0.626	118.33	0.21631
280	0.920	120.76	0.22534	0.802	120.54	0.22289	0.710	120.33	0.22070	0.636	120.10	0.21870
290	0.934	122.52	0.22770	0.814	122.30	0.22525	0.721	122.10	0.22306	0.646	121.88	

Temp, °F	Abs pressure 120 psi (Saturation temp, 93.4°F)			Abs pressure 140 psi (Saturation temp, 104.5°F)			Abs pressure 160 psi (Saturation temp, 114.5°F)			Abs pressure 180 psi (Saturation temp, 123.7°F)		
(At saturation)	*(0.350)*	*(88.05)*	*(0.16610)*	*(0.298)*	*(88.99)*	*(0.16566)*	*(0.259)*	*(89.77)*	*(0.16522)*	*(0.228)*	*(90.38)*	*(0.16476)*
130	0.387	94.01	0.17654	0.323	93.28	0.17306	0.273	92.40	0.16977	0.233	91.47	0.16665
140	0.397	95.65	0.17932	0.332	94.96	0.17590	0.282	94.12	0.17269	0.241	93.23	0.16964
150	0.407	97.30	0.18207	0.341	96.65	0.17868	0.290	95.84	0.17553	0.249	94.99	0.17254
160	0.417	98.96	0.18474	0.350	98.34	0.18142	0.298	97.57	0.17832	0.257	96.75	0.17541
170	0.426	100.63	0.18743	0.358	100.03	0.18412	0.306	99.31	0.18106	0.265	98.52	0.17823
180	0.436	102.31	0.19011	0.366	101.72	0.18678	0.313	101.05	0.18377	0.272	100.29	0.18102
190	0.445	104.00	0.19271	0.374	103.42	0.18941	0.321	102.80	0.18646	0.280	102.07	0.18377
200	0.454	105.70	0.19529	0.383	105.14	0.19205	0.329	104.55	0.18913	0.287	103.85	0.18648
210	0.463	107.41	0.19785	0.391	106.86	0.19466	0.336	106.31	0.19175	0.294	105.63	0.18912
220	0.472	109.13	0.20041	0.399	108.59	0.19724	0.344	108.07	0.19435	0.301	107.42	0.19174
230	0.482	110.86	0.20294	0.407	110.33	0.19976	0.351	109.83	0.19693	0.307	109.21	0.19433
240	0.491	112.60	0.20545	0.415	112.09	0.20229	0.358	111.60	0.19949	0.314	111.01	0.19693
250	0.500	114.35	0.20792	0.423	113.85	0.20479	0.366	113.38	0.20203	0.321	112.81	0.19947
260	0.508	116.11	0.21035	0.431	115.63	0.20728	0.373	115.17	0.20453	0.327	114.62	0.20199
270	0.517	117.88	0.21279	0.439	117.42	0.20974	0.380	116.97	0.20700	0.334	116.44	0.20449
290	0.534	121.45	0.21760	0.455	121.03	0.21461	0.394	120.60	0.21189	0.347	120.09	0.20944

TABLE XVI.—SATURATED METHYL CHLORIDE (CH_3Cl)

Temp. F, t	Pressure, psia, p	Density: Liquid, lb per cu ft, v_f	Volume: Vapor, cu ft per lb, v_g	Total heat from −32°F: Liquid, Btu per lb, h_f	Total heat from −32°F: Latent, Btu per lb, h	Total heat from −32°F: Vapor, Btu per lb, h_g	Entropy: Liquid, Btu per lb, °F, s_f	Entropy: Vapor, Btu per lb, °F, s_g
−20	11.75	63.185	8.09	4.4	186.4	190.8	0.0100	0.4340
−18	12.50	63.060	7.73	5.1	185.9	191.0	0.0116	0.4327
−16	13.10	62.935	7.38	5.8	185.5	191.3	0.0132	0.4314
−14	13.75	62.180	7.06	6.5	185.1	191.6	0.0148	0.4301
−12	14.40	62.685	6.75	7.2	184.7	191.9	0.0164	0.4289
−10	15.00	62.560	6.46	8.0	184.2	192.2	0.0180	0.4277
−8	15.80	62.435	6.18	8.7	183.8	192.5	0.0196	0.4265
−6	16.50	62.310	5.92	9.4	183.3	192.7	0.0212	0.4253
−4	17.08	62.185	5.67	10.1	182.9	193.0	0.0228	0.4241
−2	17.90	62.061	5.41	10.8	182.4	193.2	0.0244	0.4229
0	18.80	61.936	5.18	11.6	182.0	193.6	0.0259	0.4217
2	19.60	61.811	4.96	12.3	181.5	193.8	0.0275	0.4205
4	20.50	61.686	4.75	13.0	181.1	194.1	0.0291	0.4193
6	21.50	61.561	4.55	13.7	180.6	194.3	0.0307	0.4182
8	22.40	61.436	4.36	14.5	180.1	194.6	0.0322	0.4171
10	23.30	61.311	4.18	15.3	179.6	194.9	0.0337	0.4160
12	24.40	61.187	4.02	16.0	179.2	195.2	0.0353	0.4149
14	25.34	61.086	3.86	16.7	178.7	195.4	0.0369	0.4138
16	26.50	60.959	3.70	17.4	178.3	195.7	0.0384	0.4127
18	27.60	60.831	3.55	18.2	177.8	196.0	0.0399	0.4116
20	28.80	60.702	3.41	19.0	177.3	196.3	0.0414	0.4106
22	29.80	60.593	3.28	19.7	176.8	196.5	0.0430	0.4095
24	31.20	60.404	3.15	20.4	176.3	196.7	0.0445	0.4085
26	32.50	60.365	3.03	21.2	175.8	197.0	0.0460	0.4075
28	33.80	60.206	2.92	21.9	175.4	197.3	0.0475	0.4065
30	35.2	60.077	2.81	22.7	174.9	197.6	0.0490	0.4055
32	36.6	59.914	2.69	23.4	174.4	197.8	0.0505	0.4045
34	37.9	59.779	2.59	24.1	173.9	198.0	0.0520	0.4035
36	39.5	59.650	2.49	24.9	173.4	198.3	0.0535	0.4025
38	41.1	59.521	2.40	25.7	172.9	198.6	0.0550	0.4016
40	42.6	59.492	2.31	26.5	172.4	198.9	0.0565	0.4007
42	44.3	59.263	2.23	27.2	171.9	199.1	0.0580	0.3998
44	46.1	59.134	2.14	27.9	171.4	199.3	0.0595	0.3989
46	47.8	59.005	2.06	28.7	170.9	199.6	0.0610	0.3980
48	49.6	58.876	1.99	29.5	170.4	199.9	0.0625	0.3971
50	51.5	58.747	1.93	30.3	169.9	200.2	0.0639	0.3963
52	53.5	58.616	1.86	31.0	169.4	200.4	0.0654	0.3954
54	55.4	58.484	1.79	31.7	168.9	200.6	0.0669	0.3945
56	57.4	58.353	1.72	32.5	168.4	200.9	0.0684	0.3937
58	59.3	58.220	1.67	33.3	167.8	201.1	0.0698	0.3929
60	61.6	58.077	1.61	34.1	167.2	201.3	0.0712	0.3921
62	63.8	57.943	1.55	34.8	166.8	201.6	0.0727	0.3913
64	66.1	57.809	1.50	35.5	166.3	201.8	0.0742	0.3905
66	68.4	57.675	1.45	36.3	165.7	202.0	0.0756	0.3897
68	71.0	57.541	1.39	37.1	165.1	202.2	0.0770	0.3889
70	73.3	57.403	1.34	37.9	164.5	202.4	0.0784	0.3882
72	75.8	57.265	1.30	38.6	164.1	202.7	0.0799	0.3874
74	78.5	57.127	1.26	39.3	163.6	202.9	0.0813	0.3867
76	79.8	56.989	1.22	40.0	163.0	203.0	0.0827	0.3860
78	82.5	56.851	1.17	40.8	162.5	203.3	0.0841	0.3853
80	85.3	56.714	1.14	41.7	161.9	203.6	0.0855	0.3846
82	89.7	56.576	1.10	42.4	161.3	203.7	0.0869	0.3839
84	92.7	56.438	1.06	43.1	160.8	203.9	0.0883	0.3832
86	95.5	56.300	1.04	43.9	160.2	204.1	0.0897	0.3825
88	98.9	56.161	1.00	44.7	159.7	204.4	0.0911	0.3819
90	102.1	56.022	0.98	45.5	159.1	204.6	0.0925	0.3813
92	105.3	55.883	0.95	46.2	158.6	204.8	0.0939	0.3807
94	108.3	55.744	0.92	47.0	158.0	205.0	0.0953	0.3801
96	112.0	55.605	0.89	47.8	157.4	205.2	0.0967	0.3795
98	115.3	55.466	0.88	48.6	156.9	205.5	0.0981	0.3789
100	118.8	55.327	0.85	49.4	156.3	205.7	0.0994	0.3783
110	137.6	54.633	0.765	53.4	153.5	206.9	0.1062	0.3756
120	159.6	53.936	0.700	57.5	150.6	208.1	0.1128	0.3732

Table XVII.—Saturated Sulphur Dioxide (SO_2)

Temp, °F, t	Pressure, psia, p	Volume	Density	Total heat above −40°F			Entropy from −40°F		
		Vapor, cu ft per lb, v_f	Liquid, lb per cu ft, $1/v_f$	Liquid, Btu per lb, h_f	Latent, Btu per lb, h	Vapor, Btu per lb, h_g	Liquid, s_f	Evap s_{fg}	Vapor, s_g
−40	3.136	22.42	95.79	0.00	178.61	178.61	0.00000	0.42562	0.42562
−30	4.331	16.56	94.94	2.93	176.97	179.90	0.00674	0.41190	0.41864
−20	5.883	12.42	94.10	5.98	175.09	181.07	0.01366	0.39826	0.41192
−10	7.863	9.44	93.27	9.16	172.97	182.13	0.02075	0.38469	0.40544
0	10.35	7.280	92.42	12.44	170.63	183.07	0.02795	0.37122	0.39917
2	10.91	6.923	92.25	13.12	170.13	183.25	0.02941	0.36853	0.39794
4	11.50	6.584	92.08	13.78	169.63	183.41	0.03084	0.36586	0.39670
5	11.81	6.421	92.00	14.11	169.38	183.49	0.03155	0.36454	0.39609
6	12.12	6.266	91.91	14.45	169.12	183.57	0.03228	0.36319	0.39547
8	12.75	5.967	91.74	15.13	168.60	183.73	0.03373	0.36053	0.39426
10	13.42	5.682	91.58	15.80	168.07	183.87	0.03519	0.35787	0.39306
11	13.77	5.548	91.49	16.14	167.80	183.94	0.03592	0.35654	0.39246
12	14.12	5.417	91.41	16.48	167.53	184.01	0.03664	0.35521	0.39185
13	14.48	5.289	91.33	16.81	167.26	184.07	0.03737	0.35388	0.39125
14	14.84	5.164	91.24	17.15	166.97	184.14	0.03808	0.35257	0.39065
15	15.21	5.042	91.16	17.49	166.72	184.21	0.03880	0.35125	0.39005
16	15.59	4.926	91.07	17.84	166.44	184.28	0.03953	0.34993	0.38946
17	15.98	4.812	90.98	18.18	166.16	184.34	0.04026	0.34861	0.38887
18	16.37	4.701	90.89	18.52	165.88	184.40	0.04098	0.34729	0.38827
19	16.77	4.593	90.80	18.86	165.60	184.46	0.04169	0.34598	0.38767
20	17.18	4.487	90.71	19.20	165.32	184.52	0.04241	0.34466	0.38707
21	17.60	4.386	90.62	19.55	165.03	184.58	0.04313	0.34335	0.38648
22	18.03	4.287	90.53	19.90	164.74	184.64	0.04385	0.34204	0.38589
23	18.46	4.190	90.44	20.24	164.45	184.69	0.04457	0.34073	0.38530
24	18.89	4.096	90.33	20.58	164.16	184.74	0.04528	0.33943	0.38471
25	19.34	3.994	90.24	20.92	163.87	184.79	0.04600	0.33812	0.38412
26	19.80	3.915	90.15	21.26	163.58	184.84	0.04671	0.33683	0.38354
27	20.26	3.829	90.06	21.61	163.28	184.89	0.04743	0.33553	0.38296
28	20.73	3.744	89.96	21.96	162.98	184.94	0.04814	0.33422	0.38236
29	21.21	3.662	89.86	22.30	162.68	184.98	0.04886	0.33292	0.38178
30	21.70	3.581	89.76	22.64	162.38	185.02	0.04956	0.33163	0.38119
31	22.20	3.503	89.67	22.98	162.08	185.06	0.05027	0.33034	0.38061
32	22.71	3.437	89.58	23.33	161.77	185.10	0.05099	0.32904	0.38003
33	23.23	3.355	89.48	23.68	161.46	185.14	0.05171	0.32774	0.37945
34	23.75	3.283	89.39	24.03	161.15	185.18	0.05242	0.32645	0.37887
35	24.28	3.212	89.29	24.38	160.84	185.22	0.05312	0.32517	0.37829
40	27.10	2.887	88.81	26.12	159.25	185.37	0.05668	0.31873	0.37541
45	30.15	2.601	88.34	27.86	157.62	185.48	0.06020	0.31234	0.37254
50	33.45	2.348	87.87	29.61	155.95	185.56	0.06370	0.30599	0.36969
55	37.05	2.124	87.41	31.36	154.24	185.60	0.06715	0.29971	0.36686
60	40.93	1.926	86.95	33.10	152.49	185.59	0.07060	0.29345	0.36405
65	45.13	1.749	86.50	34.84	150.70	185.54	0.07401	0.28724	0.36125
70	49.62	1.590	86.02	36.58	148.88	185.46	0.07736	0.28110	0.35846
75	54.47	1.448	85.52	38.32	147.02	185.34	0.08070	0.27498	0.35568
80	59.68	1.321	85.03	40.05	145.12	185.17	0.08399	0.26897	0.35296
81	60.77	1.297	84.93	40.39	144.74	185.13	0.08462	0.26772	0.35234
82	61.88	1.274	84.84	40.73	144.36	185.09	0.08525	0.26652	0.35177
83	63.01	1.253	84.74	41.08	143.97	185.05	0.08589	0.26532	0.35121
84	64.14	1.229	84.64	41.43	143.58	185.01	0.08653	0.26412	0.35065
85	65.28	1.207	84.54	41.78	143.19	184.97	0.08718	0.26291	0.35009
86	66.45	1.185	84.44	42.12	142.80	184.92	0.08783	0.26171	0.34954
87	67.64	1.164	84.35	42.46	142.41	184.87	0.08847	0.26052	0.34899
88	68.84	1.144	84.25	42.80	142.02	184.82	0.08910	0.25933	0.34843
89	70.04	1.124	84.15	43.15	141.62	184.77	0.08974	0.25813	0.34787
90	71.25	1.104	84.05	43.50	141.22	184.72	0.09038	0.25693	0.34731
95	77.60	1.011	83.57	45.20	139.23	184.43	0.09349	0.25103	0.34452
100	84.52	0.9262	83.07	46.90	137.20	184.10	0.09657	0.24516	0.34173
110	99.76	0.7804	82.03	50.26	133.05	183.31	0.10254	0.23357	0.33611
120	120.93	0.6598	80.90	53.58	128.78	182.36	0.10829	0.22217	0.33046
140	158.61	0.4758	78.61	60.04	119.90	179.94	0.11893	0.19990	0.31883

Table XVIII.—Superheated Sulphur Dioxide (SO_2)

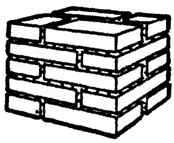
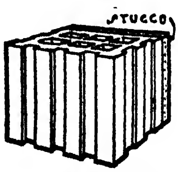
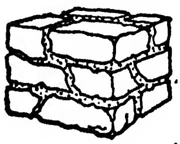
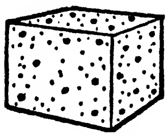
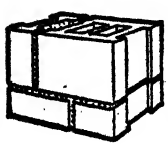
Temp, t, °F	Abs pressure 10 psi (Saturation temp, −1.34°F)			Abs pressure 15 psi (Saturation temp, 14.43°F)			Abs pressure 20 psi (Saturation temp, 26.44°F)			Abs pressure 25 psi (Saturation temp, 36.33°F)		
	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s	Volume, cu ft per lb, v	Total heat, Btu per lb, h	Entropy, Btu per lb, °F, s
At saturation	*(7.52)*	*(182.95)*	*(0.4000)*	*(5.110)*	*(184.17)*	*(0.39091)*	*(3.878)*	*(184.86)*	*(0.38329)*	*(3.123)*	*(185.26)*	*(0.37754)*
20	7.939	186.7	0.40802	5.192	185.4	0.39270						
30	8.030	188.4	0.41159	5.333	187.3	0.39672						
40	8.316	190.1	0.41505	5.470	189.2	0.40054	4.035	187.8	0.38959	3.181	186.1	0.37927
50	8.500	191.8	0.41837	5.604	191.0	0.40424	4.145	189.8	0.39346	3.273	188.4	0.38372
60	8.681	193.5	0.42161	5.734	192.8	0.40777	4.251	191.8	0.39719	3.363	190.6	0.38795
70	8.860	195.2	0.42480	5.862	195.6	0.41116	4.354	193.7	0.40080	3.451	192.7	0.39198
80	9.038	196.9	0.42795	5.988	196.4	0.41443	4.454	195.6	0.40429	3.536	194.7	0.39582
90	9.214	198.6	0.43104	6.112	198.2	0.41765	4.552	197.5	0.40758	3.6.8	196.7	0.39945
100	9.389	200.3	0.43407	6.233	199.9	0.42076	4.648	199.3	0.41093	3.696	198.6	0.40291
110	9.563	202.0	0.43705	6.353	201.6	0.42383	4.742	201.1	0.41415	3.772	200.5	0.40625
120	9.736	203.7	0.43997	6.471	203.3	0.42682	4.834	202.9	0.41726	3.848	202.4	0.40949
130	9.908	205.4	0.44283	6.588	205.6	0.42976	4.925	204.7	0.42027	3.923	204.2	0.41261
140	10.08	207.1	0.44565	6.705	206.7	0.43264	5.015	206.5	0.42322	3.998	206.0	0.41568
150	10.25	208.8	0.44842	6.821	208.4	0.43548	5.104	208.2	0.42613	4.073	207.8	0.41866
160	10.42	210.5	0.45116	6.937	210.1	0.43825	5.193	209.9	0.42898	4.145	209.6	0.42156
170	10.59	212.2	0.45296	7.052	211.8	0.44097	5.281	211.6	0.43176	4.216	211.4	0.42439
180	10.76	213.8	0.45651	7.167	213.5	0.44366	5.369	213.3	0.43449	4.287	213.2	0.42717
190	10.93	215.4	0.45913	7.282	215.2	0.44630	5.456	215.0	0.43716	4.358	215.0	0.42988
200	11.10	217.0	0.46171	7.396	216.9	0.44889	5.542	216.7	0.43977	4.428	216.7	0.43253
210	....	....	....	....	....	....	5.629	218.4	0.44234	4.498	218.4	0.43413

Temp, °F	Abs pressure 40 psi (Saturation temp, 58.83°F)			Abs pressure 50 psi (Saturation temp, 70.40°F)			Abs pressure 60 psi (Saturation temp, 80.29°F)			Abs pressure 70 psi (Saturation temp, 88.97°F)		
(At saturation)	*(1.970)*	*(185.60)*	*(0.36470)*	*(1.577)*	*(185.45)*	*(0.35826)*	*(1.3144)*	*(185.16)*	*(0.35272)*	*(1.125)*	*(184.77)*	*(0.34789)*
100	2.246	196.1	0.38415	1.650	193.9	0.37369	1.448	191.4	0.36403	1.181	187.6	0.35443
110	2.304	198.3	0.38810	1.825	196.4	0.37815	1.500	194.3	0.36906	1.228	191.6	0.36020
120	2.360	200.4	0.39183	1.872	198.8	0.38234	1.548	197.0	0.37375	1.272	194.8	0.36545
130	2.413	202.5	0.39541	1.917	201.1	0.38627	1.590	199.5	0.37810	1.313	197.6	0.37028
140	2.465	204.6	0.39881	1.961	203.3	0.38998	1.629	201.9	0.38217	1.352	200.3	0.37478
150	2.515	206.5	0.40209	2.003	205.4	0.39353	1.669	204.2	0.38603	1.389	202.9	0.37897
160	2.565	208.5	0.40525	2.044	207.5	0.39691	1.708	206.5	0.38963	1.424	205.3	0.38291
170	2.614	210.4	0.40831	2.084	209.6	0.40015	1.744	208.6	0.39310	1.457	207.6	0.38662
180	2.662	212.3	0.41127	2.123	211.6	0.40327	1.778	210.7	0.39639	1.489	209.9	0.39014
190	2.709	214.2	0.41416	2.161	213.4	0.40628	1.808	212.8	0.39956	1.521	212.0	0.39348
200	2.755	216.0	0.41694	2.199	215.4	0.40919	1.842	214.8	0.40260	1.551	214.1	0.39670
210	2.800	217.9	0.41966	2.237	217.3	0.41200	1.874	216.8	0.40554	1.580	216.1	0.39978
220	2.845	219.7	0.42233	2.274	219.2	0.41477	1.907	218.7	0.40839	1.608	218.1	0.40275
230	2.889	221.5	0.42494	2.311	221.1	0.41748	1.939	220.7	0.41118	1.636	220.1	0.40564
240	2.933	223.3	0.42751	2.347	223.0	0.42015	1.970	222.6	0.41391	1.664	222.1	0.40845
250	2.977	225.1	0.43007	2.383	224.9	0.42275	2.000	224.5	0.41657	1.691	224.1	0.41120
260	3.021	227.0	0.43262	2.418	226.7	0.42535	2.032	226.4	0.41917	1.718	226.0	0.41389

Temp, °F	Abs pressure 80 psi (Saturation temp, 96.88°F)			Abs pressure 100 psi (Saturation temp, 110.15°F)			Abs pressure 120 psi (Saturation temp, 121.52°F)			Abs pressure 140 psi (Saturation temp, 131.64°F)		
(At saturation)	*(0.9809)*	*(184.33)*	*(0.34357)*	*(0.7786)*	*(183.30)*	*(0.33603)*	*(0.6430)*	*(182.19)*	*(0.33954)*	*(0.5451)*	*(181.04)*	*(0.32388)*
140	1.163	198.6	0.36819	0.8928	194.6	0.35528	0.7085	190.1	0.34264	0.5734	185.1	0.33089
150	1.199	201.3	0.37270	0.9255	197.9	0.36061	0.7403	193.9	0.34904	0.6055	189.7	0.33777
160	1.232	203.9	0.37692	0.9561	200.9	0.36558	0.7700	197.4	0.35484	0.6345	193.6	0.34442
170	1.263	206.4	0.38093	0.9848	203.7	0.37009	0.7972	200.6	0.36012	0.6613	196.3	0.35041
180	1.292	208.7	0.38461	1.012	206.4	0.37431	0.8228	203.7	0.36494	0.6861	200.8	0.35588
190	1.320	211.0	0.38813	1.038	209.0	0.37829	0.8470	206.7	0.36936	0.7092	204.0	0.36088
200	1.347	213.3	0.39150	1.062	211.5	0.38203	0.8699	209.4	0.37348	0.7309	207 1	0.36548
210	1.374	215.5	0.39471	1.086	213.8	0.38556	0.8916	212.0	0.37737	0.7513	210 0	0.36976
220	1.400	217.5	0.39780	1.109	216.1	0.38892	0.9124	214.5	0.38104	0.7707	212.7	0.37379
230	1.426	219.6	0.40079	1.131	218.4	0.39214	0.9324	217.0	0.38451	0.7892	215.4	0.37758
240	1.451	221.6	0.40369	1.152	220.5	0.39524	0.9515	219.3	0.38785	0.8070	217.9	0.38118

TABLE XIX.—COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides and are based on a wind velocity of 15 mph.

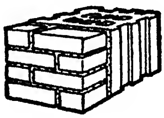
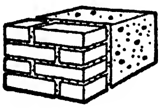
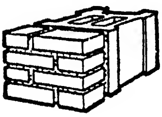
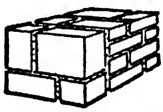
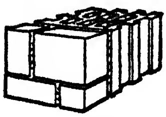
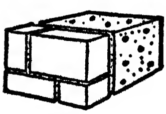
Typical construction	Type of wall	Thickness of masonry, in.	Wall
	Solid brick Based on 4-in. hard brick and the remainder common brick	8 12 16	1 2 3
	Hollow tile Stucco exterior finish The 8-in. and 10-in. tile figures are based on two cells in the direction of flow of heat. The 12-in. tile is based on three cells in the direction of flow of heat. The 16-in. tile consists of one 10-in. tile and one 6-in. tile each having two cells in the direction of heat flow	8 10 12 16	4 5 6 7
	Limestone or sandstone	8 12 16 24	8 9 10 11
	Concrete (monolithic) These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish	6 10 16 20	12 13 14 15
	Cinder (monolithic) Conductivity $k = 4.36$	6 10 16 20	16 17 18 19
	Haydite (monolithic) Conductivity $k = 3.96$	6 10 16 20	20 21 22 23
	Cinder blocks Cores filled with dry cinders, 69.7 lb per cu ft Cores filled with granulated cork, 5.12 lb per cu ft Cores filled with rock wool, 14.2 lb per cu ft Based on one air cell in direction of heat flow Cores filled with granulated cork, 5.24 lb per cu ft	8 8 8 8 12 12	24 25 26 27 28 29
	Concrete blocks Cores filled with granulated cork, 5.14 lb per cu ft Based on one air cell in direction of heat flow	8 8 12	30 31 32
	Haydite blocks Cores filled with granulated cork, 5.06 lb per cu ft	8 8	33 34
	Haydite blocks Cores filled with granulated cork, 5.6 lb per cu ft	12 12	35 36

[a] Computed from factors marked by * in Table 2; "A.S.H.V.E. Guide," Chapter 5, 1939.

Interior finish											
Uninsulated walls							Insulated walls				
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plaster board (⅜ in.)—furred	Decorated building board (½ in.) without plaster—furred	Plaster (½ in.) on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on corkboard (1½ in.) set in cement mortar (½ in.)	Plaster (¾ in.) on metal lath attached to furring strips—furred space (over ¾ in. wide) faced one side with bright aluminum foil	Plaster on metal lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b,c])	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.50	0.46	0.30	0.32	0.30	0.23	0.22	0.16	0.14	0.23	0.12	0.20
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.14	0.12	0.19	0.11	0.17
0.28	0.27	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.17	0.10	0.15
0.40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.30	0.29	0.22	0.22	0.22	0.17	0.17	0.14	0.12	0.18	0.10	0.16
0.25	0.24	0.19	0.19	0.19	0.16	0.15	0.12	0.11	0.16	0.097	0.14
0.71	0.64	0.37	0.39	0.37	0.26	0.25	0.18	0.15	0.26	0.13	0.23
0.58	0.53	0.33	0.34	0.33	0.24	0.23	0.17	0.14	0.24	0.13	0.21
0.49	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.79	0.70	0.39	0.42	0.39	0.27	0.26	0.19	0.16	0.27	0.13	0.23
0.62	0.57	0.34	0.37	0.34	0.25	0.24	0.18	0.15	0.25	0.13	0.22
0.48	0.44	0.29	0.31	0.29	0.22	0.21	0.16	0.14	0.22	0.12	0.20
0.41	0.39	0.27	0.28	0.27	0.21	0.20	0.15	0.13	0.21	0.12	0.18
0.46	0.43	0.29	0.30	0.29	0.22	0.21	0.16	0.14	0.22	0.12	0.19
0.33	0.31	0.23	0.24	0.23	0.18	0.18	0.14	0.12	0.18	0.11	0.16
0.22	0.22	0.17	0.18	0.17	0.15	0.14	0.12	0.10	0.15	0.09	0.13
0.19	0.18	0.15	0.15	0.15	0.13	0.13	0.11	0.09	0.13	0.09	0.12
0.44	0.41	0.28	0.29	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
0.30	0.29	0.22	0.23	0.22	0.17	0.17	0.14	0.12	0.18	0.10	0.16
0.21	0.20	0.16	0.17	0.16	0.14	0.14	0.11	0.10	0.14	0.09	0.13
0.17	0.17	0.14	0.14	0.14	0.12	0.12	0.10	0.09	0.12	0.08	0.11
0.42	0.39	0.27	0.28	0.27	0.21	0.20	0.16	0.13	0.21	0.12	0.19
0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
0.22	0.21	0.17	0.18	0.17	0.14	0.14	0.12	0.11	0.14	0.09	0.13
0.23	0.22	0.19	0.18	0.18	0.15	0.14	0.12	0.10	0.15	0.09	0.14
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.20	0.19	0.17	0.16	0.16	0.13	0.13	0.11	0.10	0.14	0.09	0.13
0.56	0.52	0.32	0.34	0.32	0.24	0.23	0.17	0.14	0.24	0.12	0.21
0.41	0.39	0.27	0.28	0.27	0.21	0.20	0.15	0.13	0.21	0.12	0.18
0.49	0.46	0.30	0.32	0.30	0.23	0.22	0.16	0.14	0.23	0.12	0.20
0.36	0.34	0.26	0.26	0.24	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.18	0.17	0.15	0.15	0.14	0.13	0.12	0.10	0.09	0.13	0.08	0.12
0.34	0.32	0.25	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.15	0.14	0.13	0.13	0.12	0.11	0.11	0.09	0.08	0.11	0.08	0.10

[b] Based on the actual thickness of 2-in. furring strips.
Source: Copyright, A.S.H.V.E. from "Heating, Ventilating and Air Conditioning Guide," Chapter 5, 1939.
[c] A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

Typical construction	Type of wall: Facing	Type of wall: Backing	Wall No.
	4-in. brick veneer[d]	6-in. 8-in. hollow tile[c] 10-in. 12-in.	37 38 39 40
	4-in. brick veneer[d]	6-in. 10-in. concrete 16-in.	41 42 43
	4-in brick veneer[d]	8-in. cinder blocks 8-in. cinder blocks—cores filled with granulated cork, 5.12 lb per cu ft 12-in. cinder blocks 12-in. cinder blocks—cores filled with granulated cork, 5.24 lb per cu ft.	44 45 46 47
		8-in. concrete blocks 8-in. concrete blocks—cores filled with granulated cork, 5.14 lb per cu ft 12-in. concrete blocks 8-in. Haydite block 8-in. Haydite block—cores filled with granulated corck, 5.06 lb per cu ft 12-in. Haydite block 12-in. Haydite block—cores filled with granulated cork, 5.6 lb per cu ft.	48 49 50 51 52 53 54
	4-in. cut-stone veneer[d]	8-in. 12-in. common brick 16-in.	55 56 57
	4-in. cut-stone veneer[d]	6-in. 8-in. hollow tile[c] 10-in. 12-in.	58 59 60 61
	4-in. cut-stone veneer[d]	6-in. 10-in. concrete 16-in.	62 63 64

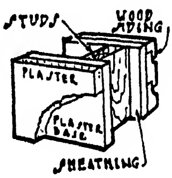
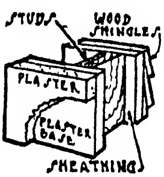
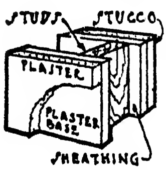
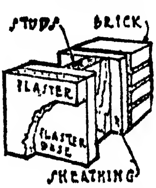
[a] Computed from factors marked by * in Table 2, "A.S.H.V.E. Guide," Chap 5, 1939.
[b] Based on the actual thickness of 2-in. furring strips.
[c] The 6-, 8-, and 10-in. tile figures are based on two cells in the direction of heat flow. The 12-in. tile, is based on three cells in the direction of heat flow.

MASONRY WALLS WITH VARIOUS TYPES OF VENEERS[d]

Interior finish											
Uninsulated walls						Insulated walls					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plaster board (⅜ in.)—furred	No plaster—decorated rigid or building board interior finish (½ in.)—furred	Plaster (½ in.) on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on cork board (1½ in.) set in cement mortar (½ in.)	Plaster (¾ in.) on metal lath attached to furring strips—furred space (over ¾ in. wide) faced one side with bright aluminum foil	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—rock-wool fill (1⅝ in.[b])[f]	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—flexible insulation (½ in.) between furring strips (one air space)
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*J*	*K*	*L*
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.34	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.34	0.32	0.23	0.24	0.23	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.27	0.26	0 20	0.21	0.20	0.16	0.16	0.13	0.11	0.16	0.10	0.15
0.57	0.53	0.33	0.35	0.33	0.24	0.23	0.17	0.14	0.24	0.13	0.21
0.48	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
0.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.20	0.19	0.16	0.16	0.16	0.13	0.13	0.11	0.10	0.13	0.09	0.12
0.31	0.30	0.22	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
0.18	0.18	0.15	0.15	0.15	0.13	0.12	0.10	0.09	0.13	0.08	0.12
0.44	0.42	0.28	0.30	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
0.34	0.32	0.24	0.25	0.23	9.19	0.18	0.14	0.12	0.19	0.11	0.17
0.40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
0.17	0.16	0.14	0.14	0.14	0.12	0.12	0.10	0.09	0.12	0.08	0.11
0.29	0.28	0.21	0.22	0.21	0.17	0.17	0.13	0.12	0.17	0.10	0.16
0.14	0.14	0.12	0.12	0.12	0.10	0.10	0.09	0.08	0.10	0.07	0.10
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.28	0.27	0.21	0.21	0.21	0.17	0.16	0.13	0.12	0.17	0.10	0.15
0.23	0.22	0.18	0.18	0.18	0.15	0.14	0.12	0.11	0.15	0.095	0.14
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.28	0.26	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.17	0.10	0.15
0.61	0.56	0.34	0.86	0.34	0.25	0.24	0.18	0.15	0.25	0.13	0.22
0.51	0.47	0.31	0.32	0.31	0.23	0.22	0.17	0.14	0.23	0.12	0.20
0.41	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.21	0.11	0.18

[d] Calculations include cement mortar (½ in.) between veneer or facing and backing.
[e] Based on one air cell in direction of heat flow.
[f] A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.
SOURCE: Copyright, A.S.H.V.E., from "Heating, Ventilating and Air Conditioning Guide," Chap. 5, 1939.

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

Typical construction	Exterior finish	Type of sheathing	Wall No.
	Wood siding or clapboard	1-in. wood[d]	65
		25/32-in. rigid insulation	66
		½-in. plaster board	67
	Wood shingles	1-in. wood[d]	68
		25/32-in. rigid insulation[e]	69
		½-in. plaster board[e]	70
	Stucco	1-in. wood[d]	71
		25/32-in. rigid insulation	72
		½-in. plaster board	73
	Brick[f] veneer	1-in. wood[d]	74
		25/32-in. rigid insulation	75
		½-in. plaster board	76

[a] Computed from factors marked by * in Table 2, "A.S.H.V.E. Guide," Chap. 5, 1939.
[b] These coefficients may also be used with sufficient accuracy for plaster on wood lath or plaster on plaster board.
[e] Based on the actual width of 2- by 4-in. studding, viz. 3⅝ in.

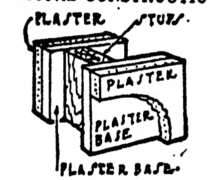
VARIOUS TYPES OF FRAME CONSTRUCTION[a]

Interior finish													
	No insulation between studding								Insulation between studding				
	Plaster on wood lath on studding	Plaster (3/4 in.) on metal lath on studding	Plaster (1/2 in.) on plaster board (3/8 in.) on studding	Plaster (1/2 in.) on rigid insulation (1/2 in.) on studding	Plaster (1/2 in.) on rigid insulation (1 in.) on studding	Plaster (1/2 in.) on corkboard (1 1/2 in.) on studding	No plaster—decorated rigid or building board interior finish (1/2 in.)	1 in. wood sheathing,[d] furring strips, plaster (1/2 in.) on wood lath	Plaster (3/4 in.) on metal lath—stud space faced one side with bright aluminum foil	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1/2 in.) between studding and in contact with sheathing	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1/2 in.) between studding—2 air spaces	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1 in.) between studding—2 air spaces	Plaster (3/4 in.) on metal lath[b] on studding—rock wool fill (3 5/8 in.[c]) between studding[g,h]
	A	B	C	D	E	F	G	H	I	J	K	L	M
	0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.17	0.19	0.17	0.15	0.12	0.072
	0.19	0.20	0.19	0.16	0.13	0.10	0.16	0.14	0.16	0.15	0.13	0.10	0.068
	0.31	0.33	0.31	0.22	0.17	0.13	0.23	0.19	0.24	0.20	0.17	0.13	0.760
	0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.17	0.20	0.17	0.15	0.12	0.072
	0.17	0.17	0.17	0.14	0.11	0.092	0.14	0.14	0.14	0.13	0.11	0.094	0.064
	0.24	0.25	0.24	0.19	0.15	0.11	0.19	0.19	0.19	0.17	0.15	0.12	0.071
	0.30	0.32	0.30	0.22	0.16	0.12	0.22	0.19	0.23	0.20	0.17	0.13	0.076
	0.22	0.23	0.22	0.17	0.14	0.11	0.19	0.15	0.18	0.16	0.14	0.11	0.071
	0.40	0.43	0.40	0.26	0.19	0.14	0.28	0.22	0.29	0.24	0.20	0.14	0.081
	0.27	0.28	0.27	0.20	0.15	0.12	0.21	0.17	0.21	0.18	0.16	0.12	0.074
	0.21	0.21	0.21	0.16	0.14	0.10	0.17	0.15	0.17	0.15	0.13	0.11	0.068
	0.35	0.37	0.35	0.24	0.18	0.13	0.25	0.21	0.26	0.22	0.18	0.14	0.079

[d] Yellow pine or fir—actual thickness about 25/32 in.
[e] Furring strips between wood shingles and sheathing.
[f] Small air space and mortar between building paper and brick veneer neglected.
[g] A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.
[h] Stud and rock wool fill areas combined.

TABLE XXII.—COEFFICIENTS OF TRANSMISSION (U) OF FRAME INTERIOR WALLS AND PARTITIONS[a]

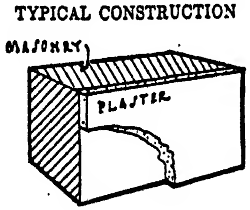
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION 	Wall	Single partition (finish on one side of studding)	Double partition (finished on both sides of studding)				
			Air space between studding	Flaked gypsum fills between studding	Rock wool fills between studding	½-in. flexible insulation between studding (one air space)	Stud space faced one side with bright aluminum foil
Type of wall		A	B	C	D	E	F
Wood lath and plaster on studding	77	0.62	0.34	0.11	0.076	0.21	0.24
Metal lath and plaster[c] on studding	78	0.69	0.39	0.11	0.078	0.23	0.26
Plaster board (⅜ in.) and plaster[d] on studding	79	0.61	0.34	0.10	0.075	0.21	0.24
½-in. rigid insulation and plaster[d] on studding	80	0.35	0.18	0.083	0.063	0.14	0.15
1-in. rigid insulation and plaster[d] on studding	81	0.23	0.12	0.066	0.054	0.097	0.10
1½-in. corkboard and plaster[d] on studding	82	0.16	0.081	0.052	0.044	0.070	0.073
2-in. corkboard and plaster[d] on studding	83	0.12	0.063	0.045	0.038	0.057	0.059

[a] Computed from factors marked by * in Table 2, "A.S.H.V.E. Guide," Chap. 5, 1939.
[b] Thickness assumed 3⅝-in.
[c] Plaster on metal lath assumed ¾-in. thick.
[d] Plaster assumed ½-in. thick.
SOURCE: Copyright, A.S.H.V.E., from "Heating, Ventilating and Air Conditioning Guide," Chap. 5, 1939.

TABLE XXIII.—COEFFICIENTS OF TRANSMISSION (U) OF MASONRY PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

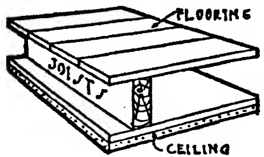
TYPICAL CONSTRUCTION 	No.	Plain walls (no plaster)	Walls plastered on one side	Walls plastered on both sides
Type of wall		A	B	C
4-in. hollow clay tile	84	0.45	0.42	0.40
4-in. common brick	85	0.50	0.46	0.43
4-in. hollow gypsum tile	86	0.30	0.28	0.27
2-in. solid plaster	87	....	....	0.53

[a] Computed from factors marked by * in Table 2, "A.S.H.V.E. Guide," Chap. 5, 1939.
SOURCE: Copyright, A.S.H.V.E., from "Heating, Ventilating and Air Conditioning Guide," Chap. 5, 1939.

Table XXIV.—Coefficients of Transmission (U) of Frame Construction Floors and Ceilings[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



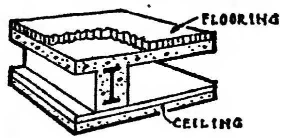
Type of ceiling	Insulation between joists	No.	Type of flooring: No flooring	Yellow pine flooring[b] on joists	Yellow pine flooring on rigid insulation (1/2 in.) on joists	Maple or oak flooring[c] on yellow pine subflooring[b] on joists	1/8-in. battleship linoleum on yellow pine flooring[b]
			A	*B*	*C*	*D*	*E*
No ceiling	None	1	....	0.46	0.27	0.34	0.34
Metal lath and plaster (3/4 in.)	None	2	0.69	0.30	0.21	0.25	0.25
Wood lath and plaster	None	3	0.62	0.28	0.20	0.24	0.24
Plaster board (3/8 in.) and plaster (1/2 in.)	None	4	0.61	0.28	0.20	0.24	0.23
Rigid insulation (1/2 in.) and plaster (1/2 in.)	None	5	0.35	0.21	0.16	0.18	0.18
Rigid insulation (1 in.) and plaster (1/2 in.)	None	6	0.23	0.16	0.13	0.14	0.14
Metal lath and plaster	Bright aluminum foil[e]	7	0.59	0.22	0.17	0.19	0.19
Metal lath and plaster	Flexible[d] insulation (1 in.)	8	0.17	0.13	0.11	0.12	0.12
Metal lath and plaster	Flexible[d] insulation (2 in.)	9	0.10	0.086	0.076	0.081	0.081
Metal lath and plaster	Rock-wool fill (3 5/8 in.)	10	0.079	0.068	0.063	0.066	0.066
Corkboard (1 1/2 in. and plaster (1/2 in.)	None	11	0.16	0.12	0.10	0.11	0.11
Corkboard (2 in.) and plaster (1/2 in.)	None	12	0.12	0.10	0.087	0.094	0.094

[a] Computed from factors marked by * in Table 2, "A.S.H.V.E. Guide," Chap. 5, 1939.
[b] Thickness assumed to be 25/32 in.
[c] Thickness assumed to be 13/16 in.
[d] Based on one air space with no flooring, and two air spaces with flooring. The value of U will be the same if insulation is applied to under side of joists and separated from lath and plaster ceiling by 1-in. furring strips.
[e] Air space faced on one side with bright aluminum foil.
Source: Copyright, A.S.H.V.E., from "Heating, Ventilating and Air Conditioning Guide," Chap. 5, 1939.

TABLE XXV.—COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



Type of Ceiling	Thickness of concrete, in.	No	Type of flooring: No flooring (concrete bare)[b]	Yellow pine flooring[c] on wood sleepers embedded in concrete[d]	Maple or oak flooring[e] on yellow pine subflooring[c] on wood sleepers mbedded in concrete	Tile or terrazzo[f] flooring on concrete	¼-in. battleship linoleum directly on concrete
			A	B	C	D	E
No Ceiling	4	1	0 65	0 40	0 31	0.61	0.44
	6	2	0.59	0.37	0 30	0.56	0.41
	8	3	0.53	0.35	0.28	0.51	0.38
	10	4	0.49	0.33	0 27	0.47	0.36
¾ in. plaster applied directly to underside of concrete	4	5	0.59	0.38	0.30	0.56	0.41
	6	6	0.54	0.35	0.28	0.52	0.38
	8	7	0.50	0.33	0.27	0.47	0.36
	10	8	0.45	0.32	0 26	0.44	0.34
Suspended or furred metal lath and plaster (¾ in.) ceiling	4	9	0.37	0.28	0 23	0.36	0.29
	6	10	0.35	0.26	0.22	0.34	0.28
	8	11	0.33	0.25	0 21	0.32	0.27
	10	12	0.32	0 24	0 21	0.31	0.25
Suspended or furred ceiling of plaster board (⅜ in.) and plaster (½ in.)	4	13	0 35	0.26	0.22	0.34	0.28
	6	14	0.33	0.25	0.21	0.32	0.26
	8	15	0.31	0.24	0 21	0.30	0.25
	10	16	0.30	0.23	0.20	0 29	0.24
Suspended or furred ceiling of rigid insulation (½ in.) and plaster (½ in.)	4	17	0.24	0.20	0.17	0.24	0.21
	6	18	0.23	0.19	0.17	0.23	0.20
	8	19	0.22	0.18	0.16	0.22	0.19
	10	20	0.22	0 18	0 16	0.21	0.19
Plaster (½ in.) on corkboard (1½ in.) set in cement mortar (½ in.) on concrete	4	21	0.15	0.13	0.12	0.14	0.14
	6	22	0.14	0.13	0.12	0.14	0.13
	8	23	0.14	0.12	0.11	0.14	0.13
	10	24	0.14	0.12	0.11	0.14	0.13

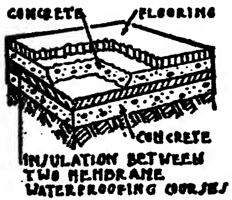
[a] Computed from factors marked by * in Table 2, "A.S.H.V.E. Guide," Chap. 5, 1939.
[b] The figures in column A may be used with sufficient accuracy for concrete floors covered with carpet.
[c] Thickness of yellow pine flooring assumed to be $^{25}/_{32}$ in.
[d] The figures in column B may be used with sufficient accuracy for maple or oak flooring[e] applied directly over the concrete on wood sleepers.
[e] Thickness of maple or oak flooring assumed to be $^{13}/_{16}$ in.
[f] Thickness of tile or terrazzo assumed 1 in.

SOURCE: Copyright, A.S.H.V.E., from "Heating, Ventilating and Air Conditioning Guide," Chap. 5, 1939.

TABLE XXVI.—COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH[e] FLOORING[a,b]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the ground and the air over the floor, and are based on still air (no wind) conditions.

TYPICAL CONSTRUCTION

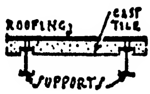
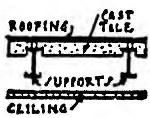
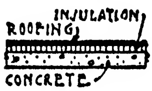
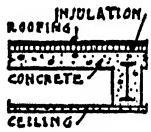
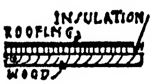
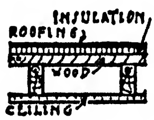
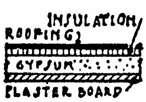
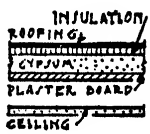
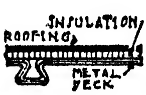
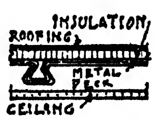


Type and thickness of insulation	Thickness of concrete in.	No.	Type of finish flooring: No flooring (concrete bare) A	Yellow pine flooring[b] on wood sleepers resting on concrete B	Maple or oak flooring[c] on yellow pine subflooring on wood sleepers resting on concrete C	Tile or terrazzo[d] on concrete D	¼-in. battleship linoleum directly on concrete E
None	4	1	1.07	0.35	0.28	0.98	0.60
	6	2	0.90	0.33	0.27	0.84	0.54
	8	3	0.79	0.32	0.26	0.74	0.50
	10	4	0.70	0.30	0.25	0.66	0.46
None	4	5	0.66	0.29	0.24	0.63	0.44
	8	6	0.54	0.27	0.23	0.52	0.39
1-in. rigid insulations	4	7	0.22	0.16	0.14	0.22	0.19
1-in. rigid insulations	8	8	0.21	0.15	0.13	0.20	0.18
2-in. corkboards	4	9	0.12	0.099	0.093	0.12	0.11
2-in. corkboards	8	10	0.12	0.096	0.090	0.12	0.11

[a] Computed from factors marked by * in Table 2, "A.S.H.V.E. Guide," Chap. 5, 1939.
[b] Assumed 25/32 in. thick.
[c] Assumed 13/16 in. thick.
[d] Assumed 1 in. thick.
[e] The figures for Nos. 5 to 10 include 3-in. cinder concrete placed directly on the ground. The insulation is applied between the cinder concrete and the stone concrete. Usually the insulation is protected on both sides by a waterproof membrane, but this is not considered in the calculations.

SOURCE: Copyright, A.S.H.V.E., from "Heating, Ventilating and Air Conditioning Guide," Chap. 5, 1939.

TABLE XXVII.—COEFFICIENTS OF TRANSMISSION (U) OF VARIOUS TYPES OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING[a]

Typical Construction: Without ceilings	Typical Construction: With metal lath and plaster ceilings[d]	Type of roof deck	Thickness of roof deck, in.	No.
		Precast cement tile	1⅝	1
		Concrete Concrete Concrete	2 4 6	2 3 4
		Wood Wood Wood Wood	1[b] 1½[b] 2[b] 4[b]	5 6 7 8
		Gypsum fiber concrete[c] (2 in.) on plaster board (⅜ in.) . . . . Gypsum fiber concrete[c] (3 in.) on plaster board (⅜ in.) . . . . Gypsum fiber concrete[c] (2 in.) on rigid insulation board (½ in.) . . . . Gypsum fiber concrete[c] (2 in.) on rigid insulation board (1 in.)	2⅜ 3⅜ 2½ 3	9 10 11 12
		Flat metal roofs Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per hr per sq ft of projected area per degree Fahrenheit difference in temperature, based on an outside wind velocity of 15 mph	. . . .	13

[a] Computed from factors marked by * in Table 2, "A.S.H.V.E. Guide," Chap. 5, 1939.
[b] Nominal thicknesses specified—actual thicknesses used in calculations.
[c] Gypsum fiber concrete—87½ per cent gypsum, 12½ per cent wood fiber.
SOURCE: Copyright, A.S.H.V.E., from "Heating, Ventilating and Air Conditioning Guide," Chap. 5, 1939.

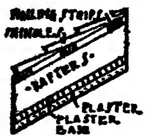
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

Without ceiling—underside of roof exposed								With metal lath and plaster ceiling[d]							
No insulation	Rigid insulation (½ in.)	Rigid insulation (1 in.)	Rigid insulation (1½ in.)	Rigid insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)	No insulation	Rigid insulation (½ in.)	Rigid insulation (1 in.)	Rigid insulation (1½ in.)	Rigid insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*J*	*K*	*L*	*M*	*N*	*O*	*P*
0.84	0.37	0.24	0.18	0.14	0.22	0.16	0.13	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.82	0.37	0.24	0.17	0.14	0.22	0.16	0.13	0.42	0.26	0.18	0.15	0.12	0.18	0.14	0.11
0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11
0.64	0.33	0.22	0.16	0.13	0.21	0.15	0.12	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11
0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10
0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095
0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.92
0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	0.18	0.14	0.12	0.10	0.087	0.11	0.096	0.082
0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097
0.32	0.22	0.16	0.13	0.11	0.15	0.12	0.10	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091
0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.10	0.20	0.16	0.13	0.11	0.09	0.12	0.10	0.087
0.19	0.15	0.12	0.10	0.09	0.12	0.10	0.08	0.16	0.13	0.11	0.09	0.08	0.10	0.09	0.077
0.95	0.39	0.25	0.18	0.14	0.23	0.17	0.13	0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11

[d] These coefficients may be used with sufficient accuracy for wood lath and plaster, or plaster board and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

TABLE XXVIII.—COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

Typical construction	Type of roofing and roof sheathing	Insulation between roof rafter	No.	Type of ceiling (applied directly to roof rafters): No ceiling (rafters exposed)	Metal lath and plaster (¾ in.)	Plaster board (⅜ in.) and plaster (½ in.)	Wood lath and plaster	Rigid insulation (½ in.)	Rigid insulation (½ in.) and plaster (½ in.)	Rigid insulation (in.) and plaster (½ in.)	Corkboard (1½ in.) and plaster (½ in.)	Corkboard (2 in.) and plaster (½ in.)
				A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*
	Wood shingles on wood strips[b]	None	1	0.46	0.30	0.29	0.29	0.22	0.21	0.16[g]	0.12	0.10
		Bright aluminum foil[g]	2		0.22	0.21	0.21	0.18	0.17	0.14	0.11	0.089
		1 in. flexible[c]	3	..	0.13	0.12	0.12	0.11	0.11	0.092	0.078	0.069
		2 in. flexible[c]	4	.	0.086	0.083	0.083	0.076	0.075	0.068	0.060	0.054
		3⅝ in. rock wool[e]	5	....	0.063	0.062	0.062	0.058	0.058	0.053	0.048	0.044
	Asphalt shingles Rigid asbestos Shingles, composition roofing, or slate or tile roofing[d] on wood sheathing[f]	None	6	0.56	0.34	0.32	0.32	0.24	0.23	0.17	0.13	0.11
		Bright alumninum foil[g]	7		0.24	0.24	0.24	0.19	0.18	0.14	0.11	0.098
		1 in. flexible[c]	8		0.13	0.13	0.13	0.11	0.11	0.095	0.080	0.071
		2 in. flexible[c]	9		0.088	0.087	0.087	0.079	0.078	0.070	0.062	0.056
		3⅝ in. rock wool[e]	10	...	0.065	0.064	0.064	0.060	0.059	0.054	0.049	0.045

[a] Computed from factors marked by * in Table 2, "A.S.H.V.E. Guide," Chap. 5, 1939. Nos. 6 to 10 based on ½-in.-thick slate.
[b] Based on 1 in. by 4 in. strips spaced 2 in.
[c] Figures based on two air spaces. Insulation may also be applied to underside of roof rafters with furring strips between.
[d] Roofing felt between roof sheathing and slate or tile neglected in calculations.
[e] Assumed 3⅝ in. thick based on the actual width of 2- by 4-in. rafters.
[f] Sheathing assumed ²⁵⁄₃₂ in. thick.
[g] Air space faced on one side with bright aluminum foil.
SOURCE: Copyright, A.S.H.V.E.— from "Heating, Ventilating and Air Conditioning Guide," Chap. 5, 1939.

TABLE XXIX.—COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS, SKYLIGHTS AND GLASS WALLS

Coefficients are based on a wind velocity of 15 mph, and are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air inside and outside of the door, window, skylight or wall.

A. Windows and Skylights

Description	U
Single	1.13[a,c]
Double	0.45[a]
Triple	0.281[a]

B. Solid Wood Doors[b,c]

Nominal Thickness Inches	Actual Thickness Inches	U
1	25/32	0.69
1¼	1 1/16	0.59
1½	1 5/16	0.52
1¾	1⅜	0.51
2	1⅝	0.46
2½	2⅛	0.38
3	2⅝	0.33

C. Glass Walls

Description	U
Hollow glass tile wall, 6 x 6 x 2 in. thick blocks, wind velocity 15 mph, outside surface; still air, inside surface	0.60
Still air, outside and inside surface	0.48

[a] See Harding and Willard, "Heating, Ventilating and Air Conditioning," rev. ed., 1932.
[b] Computed using $C = 1.15$ for wood; $f_i = 1.65$ and $f_o = 6.0$
[c] It is sufficiently accurate to use the same coefficient of transmission of doors containing thin wood panels as that of single panes of glass, namely, 1.13 Btu per hour per square foot per degree difference between inside and outside air temperatures.

ANSWERS TO PROBLEMS

CHAPTER I

1. All matter is composed of groups of small particles known as molecules. Kinetic energy is the energy possessed by all moving objects, due to their motion.

2. The three states in which matter can exist are as a solid, a liquid, or a gas.

In order to bring about a change of state, heat (called "latent heat") must be added to or taken away from a substance.

The addition of heat increases and the removal of heat reduces the kinetic energy of the molocules; in fact, this increase or decrease is the essential factor in the change of state.

3. According to the kinetic theory, a gas is not the uniform stationary substance it appears to be when seen as a whole body, but in reality is composed of small rapidly moving particles which are the individual molecules comprising the gas. The rapid motion of these molecules in every direction creates the illusion that they fill the container completely, where s actually (so the theory states) the space that the molecules would occupy if brought to rest and "laid end to end" would be only a minute fraction of the total volume of the space. These molecules constantly collide with each other and with the walls of the container, thus explaining the phenomenon of pressure exerted by a gas.

The moving particles possess kinetic energy; by the addition or removal of heat, we are able to raise or lower the kinetic energy of these molecules and thus increase or decrease their rate of motion.

In ice, for example (and the following discussion will apply to any change from solid to liquid to gas), the H_2O molecules, while not stationary, possess only sufficient kinetic energy to enable them to vibrate in more or less restricted paths. The addition of heat to ice increases the amount of their vibratory motion, but owing to the limited path of motion, the ice possesses the characteristic properties of a solid, *i.e.*, definite volume and shape, until sufficient heat energy has been applied to increase the temperature to the melting point. A transformation in the nature of the paths of motion of the molecules takes place accompanied by the absorption of a considerable amount of heat energy with no change in temperature.

The path of motion of molecules forming liquid water is not so restricted as that for ice; nevertheless the moving particles do not have the pratically unlimited freedom of motion that gaseous particles have. Consequently a liquid, while possessing freedom of shape (it will take the shape of the container it occupies), nevertheless is still restricted to a definite volume.

The addition of heat energy, which goes to increasing the kinetic energy of the molecules, raises the temperature of the liquid water to the boiling point; addition of heat to water at the boiling point, however, instead of raising the temperature, brings about a transformation in the paths of motion of the individual particles. With the addition of the final amount of the required latent heat, we find the liquid has been transformed to a gas and the kinetic energy of the individual particles increased to the point where their paths of motion are entirely free, restricted only by the walls of the container and the possibility of collision with other particles.

4. Substances when placed in close contact with each other may unite to form an entirely new substance (or substances) which bears no similarity to the initial substances and in which no trace of the original substances can be found; or they may merely remain in contact with each other without any interaction, no matter how finely divided or well-mixed the two substances may be. The union in the first case is known as a "chemical combination"; the second, a "physical mixture." With a mixture, we may, with more or less effort, always separate the union back to its original components,

without requiring a chemical change of any sort; with a chemical combination, the final substance cannot be subdivided into the original substances, except by a chemical reaction.

5.

Substance	Chemical symbols for one molecule	Molecular weights
Carbon	C	12
Hydrogen	H_2	2
Nitrogen	N_2	28
Oxygen	O_2	32
Sulphur	S	32
Mercury	Hg	200
Carbon monoxide	CO	28
Carbon dioxide	CO_2	44
Water	H_2O	18

6. $C + O_2 = CO_2$ (carbon dioxide)

7. $2H_2 + O_2 = 2H_2O$ (water)

Two molecules of water are formed. (The water is in a gaseous state when formed, but readily condenses to ordinary liquid water.)

CHAPTER II

1. Each student will have an individual answer.

2. In constructing a graph, if after plotting all the known points and drawing the curve connecting these points, we extend the curve in the same general direction beyond these known points and use these extensions to obtain additional corresponding values of two variables, we are extrapolating.

If our two variables varied according to some simple mathematical equation or law (which we could recognize from the straight-line or single-curve nature of our graph), we can feel fairly certain that if we had additional points to plot, beyond the two ends of our graph as it stands, these points when plotted would fall on the extension of our present graph.

Only when a definite simple mathematical relation exists between the two variables are we safe in using extrapolated values. With discontinuous functions, *i.e.*, variables that bear no definite relation to each other, we have *no* assurance, where we extrapolate, that the variables will continue to intersect according to the relation that exists for the known portion of the graph. The actual relationship may change in the region beyond the curve from that existing along the known portion of the curve; since when extrapolating you assume that the relationship does not change, the values obtained by extrapolating will not be correct.

3. 80 per cent efficiency.

4. 34 per cent thermal efficiency. 0.41 lb of fuel per brake horsepower.

5. 0.04 in. of water required draft. 63 per cent efficiency. 5.8 lb per sq ft grate per hour.

6. PWS — 20 will be the proper condenser for the given conditions.

7. Check your graph with the graph shown in Fig. 15, page 37.

CHAPTER III

1. Nitrogen, oxygen, carbon dioxide, and argon. (Air contains also a certain percentage of water in the gaseous state, which is important to us, but which we shall discuss in a later chapter.) 78.9 per cent nitrogen, 21.1 per cent oxygen.

2. By the phrase "an absolute pressure of one atmosphere," we mean a pressure measured on the absolute scale of pressures, equivalent to that exerted by the atmosphere itself, at some stated elevation above or below sea level. The "absolute pressure of one atmosphere" at sea level, which is taken as standard, is 14.696 psi. The absolute pressure at 2,000 ft above sea level is 13.60 psi (see table, page 44).

3. For all but extremely large tanks, the pressure gauges would all read exactly the same, demonstrating that the pressure exerted by a gas in a container is the same in all directions.

Owing to the variation in atmospheric pressure at different levels, with exceptionally large tanks, there may be sufficient difference in altitude between the top and the bottom to cause a difference in pressure-gauge readings at these points. For ordinary work, however, this slight discrepancy will be too small to be noticeable.

4.

Absolute-pressure Scale, psi	Gauge-pressure Scale, psi
100	85.3
75	60.3
50	35.3
25	10.3
14.7	0.0
0.0	−14.7

Absolute-pressure Scale, psi	Vacuum, As Read on a Compound Gauge, in. Hg
14.7	0
12.2	5.0
9.7	10.0
7.2	15.0
4.7	20.0
2.2	25.0
0.0	29.9

5. $P_1V_1 = P_2V_2$

$$(60 + 14.7)(50) = (120 + 14.7)(V_2)$$

$$V_2 = \frac{(50)(74.7)}{134.7}$$

$$= 27.7 \text{ cu ft} \quad \textit{Ans.}$$

6. $P_1V_1 = P_2V_2$

$$(2{,}000{,}000)(16.7) = (15.7)(V_2)$$

$$V_2 = \frac{(2{,}000{,}000)(16.7)}{15.7}$$

$$= 2{,}130{,}000 \text{ cu ft} \quad \textit{Ans.}$$

7. $P_1V_1 = P_2V_2$

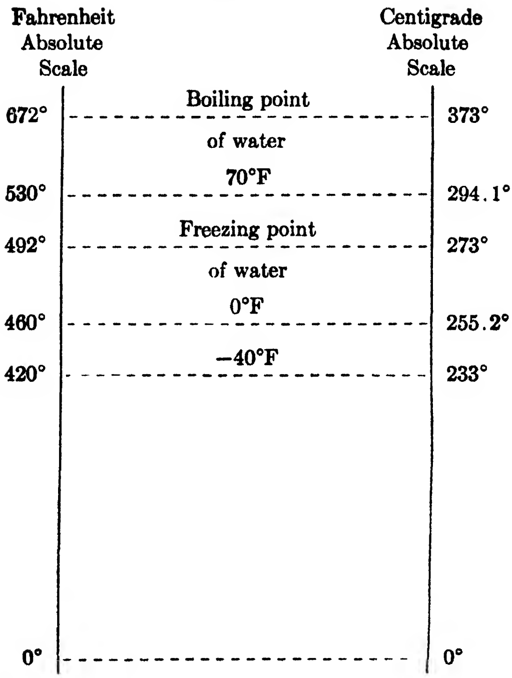
$$(14.7)(2) = (P_2)(0.25)$$

$$P_2 = \frac{(14.7)(2)}{0.25}$$

$$117.6 \text{ psia.} \quad \textit{Ans.}$$
$$\underline{-14.7 \text{ psi}}$$
$$= 102.9 \text{ psig.} \quad \textit{Ans.}$$

CHAPTER IV

1.



2. $P_1 = 0$ psig $= 14.7$ psia

$V_1 = 20$ cu ft

$P_2 = 58.5$ psig $= 73.2$ psia

$V_2 = ?$

$$P_1V_1 = P_2V_2$$

$$(14.7)(20) = (73.2)(V_2)$$

$$V_2 = \frac{(14.7)(20)}{73.2}$$

$$= 4.02 \text{ cu. ft.}$$

3. $V_1 = 15$ cu ft $\qquad V_2 = ?$

$T_1 = 0°F = 460°F$ abs $\quad T_2 = 110°F = 570°F$ abs

$$\frac{V_1}{V_2} = \frac{T_1}{T_2}$$

$$\frac{15}{460} = \frac{V_2}{570}$$

$$V_2 = \frac{(15)(570)}{460}$$

$$= 18.6 \text{ cu ft} \quad \textit{Ans.}$$

4. Since the gas fills the same tank before and after the heating process, its volume is of course constant.

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}$$

$P_1 = 30 + 14.7 = 44.7$ psia $\quad T_1 = 40°F + 460°F = 500°F$

$P_2 = ?$ $\quad T_2 = 340°F + 460°F = 800°F$

$$\frac{47.7}{P_2} = \frac{500}{800}$$

$$P_2 = \frac{(800)(44.7)}{500}$$

$$= 71.6 \text{ psia}$$

$$= 56.9 \text{ psig} \quad \textit{Ans.}$$

5. Neglecting other gases, air is 21 per cent oxygen and 79 per cent nitrogen by volume. Hence:

$$\begin{aligned} (14.7)(0.21) &= 3.09 \text{ psia partial pressure of oxygen} \\ (14.7)(0.79) &= \underline{11.61} \text{ psia partial pressure of nitrogen} \\ & 14.7 \text{ psia total pressure} \end{aligned}$$

6. When two gases are mixed together, either a chemical combination of the two will result, or they will form a physical mixture. Whether the two gases have united to form a chemical combination or merely a physical mixture will be indicated by the nature of the substances present after the original mixing has been completed. If entirely new substances have been formed, a chemical combination has taken place. If the original gases are still present, chemically unchanged, after the mixing is complete, then a physical mixture, not a chemical combination of these gases, has been formed.

7. The four gases in the tank are oxygen, nitrogen, hydrogen, and argon.

a. Since the tank was originally open to the atmosphere, we can at once obtain the partial pressure of the first two gases, oxygen and nitrogen, thus (see Prob. 5 above):

$$\begin{aligned} \text{Oxygen } 14.7(0.21) &= 3.09 \text{ psia} \\ \text{Nitrogen } 14.7(0.79) &= \underline{11.61} \text{ psia} \\ & 14.7 \text{ psia, or 0 psig} \end{aligned}$$

After adding the hydrogen, the tank is at 58.8 psig, or 73.5 psia; hence

$$73.5 - 14.7 = 58.8 \text{ psia partial pressure of the hydrogen}$$

The final absolute pressure in the tank after adding the argon is 100 psia, so we have

$$100 - 73.5 = 26.5 \text{ psia partial pressure of argon}$$

b. Since the volumes of the gases are in the same proportions to the total volume as their respective partial pressures are to the total pressure (by Dalton's law) and the total final pressure in the tank is 100 psia, we have

			% by Volume		Volume, cu ft
Oxygen	$\frac{3.09}{100}$,	or	3.09,	or	0.309
Nitrogen	$\frac{11.61}{100}$,	or	11.61,	or	1. 61
Hydrogen	$\frac{58.8}{100}$,	or	58.81,	or	5.88
Argon	$\frac{26.5}{100}$,	or	26.5,	or	2.65
Total			100.0		10.00

c. Although the individual volumes and partial pressures of the component gases are changed by the reduction in pressure, the percentage ratios of the separate volumes to each other and to the total volume remain the same.
Call the total final volume V (at 0 psig, or 14.7 psia)

$$P_1V_1 = P_2V_2$$

$$(100)(10) = (14.7)(V_2)$$

$$V_2 = \frac{(100)(10)}{14.7}$$

$$V_2 = 68.0 \text{ cu ft}$$

Using the per cent by volume obtained in *b*, we have

Oxygen 68 (0.0309)	2.1 cu ft
Nitrogen 68 (0.1161)	7.9 cu ft
Hydrogen 68 (0.588)	40 0 cu ft
Argon 68 (0.265)	18.0 cu ft
Total	68.0 cu ft

CHAPTER V

1. Heat is a form of energy. It is measured in American thermodynamic circles in "British thermal units," abbreviated "Btu." The specific heat of a substance is the number of Btu required to raise the temperature of 1 lb of the substance 1°F. The specific heat of most substances varies depending on the temperature and pressure, but the variation is slight.

2. $Q = cW(T_2 - T_1)$

where Q = heat, Btu
c = specific heat, Btu per lb
W = weight, lb
T_1 = initial temperature, °F
T_2 = final temperature, °F

3. Using the formula in Prob. 2, we have

$$Q = ? \qquad T_1 = 35°$$

$$c = 0.65 \qquad T_2 = -5°$$

$$W = 100(11.5)$$

$$Q = 0.65(100)(11.5)(-5° - 35°)$$

$$Q = 748(-40) = -29{,}900 \text{ Btu}$$

The minus sign means a *removal* of 29,900 Btu *Ans.*

4. Work equals force in pounds multiplied by distance in feet, or $W = F(D)$.

$$W = 24{,}000(400) = 9{,}600{,}000 \text{ ft-lb} \quad \textit{Ans.}$$

5. 9,600,000 ft-lb in 30 *min* must be changed to the work done in 1 *min*. This is the "output" of the hoist.

$$\frac{9{,}600{,}000}{30} = 320{,}000 \text{ ft-lb per min output}$$

The engine horsepower is the "input" of the hoist mechanism, and efficiency is 65 per cent. (See page 77 for method on the work below.)

$$0.65 = \frac{\text{output work}}{\text{engine work}} = \frac{320{,}000}{\text{engine work}}$$

$$\text{Engine work} = \frac{320{,}000}{0.65} = 492{,}000 \text{ ft-lb per min}$$

$$\frac{492{,}000}{33{,}000} = 14.9 \text{ engine horsepower.} \quad \textit{Ans.}$$

In one step we can get the result, thus,

$$\text{Engine horsepower} = \frac{320{,}000}{0.65(33{,}000)} = 14.9 \text{ engine horsepower} \quad \textit{Ans.}$$

CHAPTER VI

1. The amount of heat added to the water will be equal to the difference between the total heat in the 400 lb per hr of steam leaving the boiler at 100 psi and the total heat in the 400 lb of water entering the boiler at 60°F (see sketch on page 621).

From the Steam Tables,

$$h_f \text{ at } 60° = 28.06 \text{ Btu per lb}$$

$$h_g \text{ at } 114.0 \text{ lb} = 1189.5 \text{ Btu per lb}$$

$$\text{at } 116.0 \text{ lb} = 1189.8 \text{ Btu per lb}$$

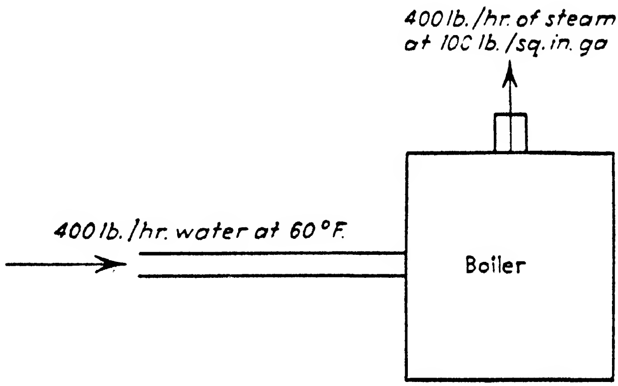
$$h_g \text{ at } 114.7 \text{ lb} = 1189.6 \text{ Btu per lb}$$

(Or, we could take h_g at 115 psia = 1189.7 Btu per lb.)

$400(1189.6 - 28.06) = 464{,}600$ Btu per hr total heat added by the boiler using 60°F feed water *Ans.*

Introducing feed water at 180° rather than 60° changes the h_f of the entering water to (from the Steam Tables) h_f at 180° = 147.92 Btu per lb

$400(1089.6 - 147.9) = 416{,}700$ Btu per hr total heat added by the boiler, using 180°F feed water



The savings effected by the increase in feed-water temperature from 60 to 180° is then

$$464{,}600 - 416{,}700 = 47{,}900 \text{ Btu per hr} \quad \textit{Ans.}$$

The saving effected, you may have noticed, is due entirely to the greater total heat of the 180° feed water compared with the total heat of the 60° feed water. The problem can be solved by the following alternate method:

$$h_f \text{ at } 60° = 28.06 \text{ Btu per lb}$$

$$h_f \text{ at } 080° = 147.92 \text{ Btu per lb}$$

$$400(147.92) - 400(28.06) = 47{,}900 \text{ Btu per hr} \quad \textit{Ans.}$$

2. In order to cool water down to 60°, the pressure of the chamber must be that which corresponds to a 60° evaporating or boiling temperature. This pressure, obtained from column (3) of Steam Table 1, is 0.5218 in. Hg.

The amount of heat to cool 10 lb per min of water at 70 to 60° can be found by using the sensible heat-transfer equation:

$$Q = cW(T_2 - T_1)$$

$$= (1)(10)(60 - 70)$$

$$= 100 \text{ Btu per min}$$

The problem states that we are to obtain this amount of cooling by evaporating a portion of the 10 lb of water. This water enters at 70° and cools to 60°.

Since the pressure in the chamber corresponds to a boiling-point temperature of 60°, the entering water at 70° is above the boiling-point temperature. It will, therefore, begin to boil and continue to boil from the moment it enters the vacuum chamber until it reaches a temperature of 60°.

The portion of the water that we evaporate, in order to obtain our cooling effect of 100 Btu per min, is boiling through the range of temperatures from 70 to 60°, rather than exactly at 60°. Should we use the h_{fg} value at 60, 63, 68, or 70° as representing the latent heat required to vaporize the water we are evaporating? We do not know *exactly*, so as a good approximation we can take the h_{fg} at 70°, the h_{fg} at 60°, and average them. We then use this value for the entire boiling process.

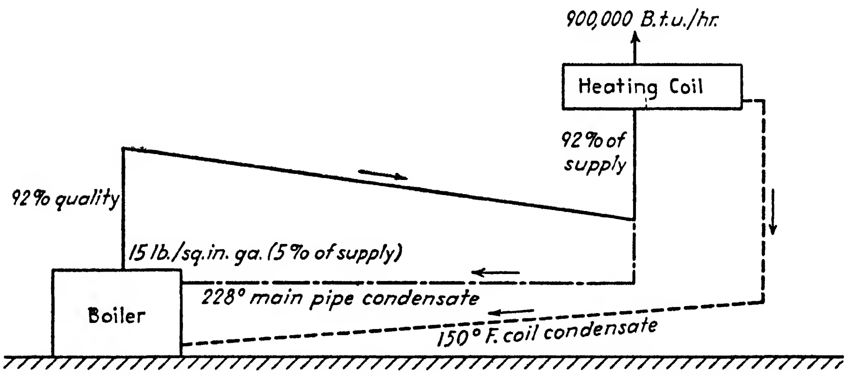
$$h_{fg} \text{ at } 70° = 1054.3 \text{ Btu per lb}$$

$$h_{fg} \text{ at } 60° = 1059.9 \text{ Btu per lb}$$

$$h_{fg} \text{ average} = 1057.1 \text{ Btu per lb}$$

$$\frac{100}{1057.1} = 0.0946 \text{ lb of water (or 0.095 lb)} \quad \textit{Ans.}$$

3. Steam enters the heating coil at 15 psig pressure, 92 per cent quality and leaves as water at 228°. The difference in the total heat between the entering and receiving conditions, which we shall obtain from the Steam Tables, then represents the heat given up to the coil (and to whatever the coil is heating) *per pound of steam* (see sketch).



$15 + 14.7 = 29.7$ psia pressure. The heat in the liquid, h_f, for this pressure is, interpolating from the tables,

$$h_f \text{ at } 28.0 \text{ lb} = 214.83 \text{ Btu per lb}$$

$$h_f \text{ at } 30.0 \text{ lb} = 218.82 \text{ Btu per lb}$$

$$h_f \text{ at } 29.7 \text{ lb} = 218.22 \text{ Btu per lb}$$

(Or we could with very little error simply use h_f at 30 lb = 218.82.) h_{fg}, for this pressure, interpolating from the tables, is

$$h_{fg} \text{ at } 28.0 \text{ lb} = 947.9 \text{ Btu per lb}$$

$$h_{fg} \text{ at } 30.0 \text{ lb} = 945.3 \text{ Btu per lb}$$

$$h_{fg} \text{ at } 29.7 \text{ lb} = 945.7 \text{ Btu per lb}$$

(Again we could with very little error use h_{fg} at 30 lb = 945.3)

If the steam leaving the boiler were dry saturated steam, *i.e.*, steam with 100 per cent quality, it would take 945.7 Btu to vaporize 1 lb at this pressure. Since the quality as given is only 92 per cent, the amount of heat of vaporization in the final wet steam is

$$(945.7)(0.92) = 870 \text{ Btu per lb}$$

The actual latent heat of vaporization added to the heat of the liquid, h_f, previously determined, will give the total heat of the steam leaving the boiler and entering the heating coil, thus

$$218.2 + 870 = 1088.2 \text{ Btu per lb}$$

The amount of heat in the condensate water eaving the heating coil, for a temperature of 150°F, is, from the tables,

$$h_f \text{ at } 150°\text{F} = 117.89 \text{ Btu per lb, or } 118 \text{ Btu per lb}$$

For the amount of heat rbsorbed by the heating coil, *per pound of steam passing through the coil*, is then

$$1088.2 \times 118 = 970 \text{ Btu per lb}$$

To furnish 900,000 Btu per hr of heat to the coil, then we must supply 900,000/970 = 928 lb of steam per hour required at the entrance to the heating coils.

In order to find out how many Btu per hour must be transferred from the combustion chamber into the water in the boiler, we can do two things: (1) find the total amount of heat that leaves the system, which will be the sum of the heating-coil output plus the loss in the supply main to the coil; or (2) find the amount of heat leaving the boiler and subtract the amount returning to the boiler from this amount. (The former is easier.)

In method 1, we already know the heating-coil output; hence, we have to find only the loss in the supply main.

If 5 per cent of the steam leaving the boiler condenses in the main, 95 per cent must reach the coil. We have found this 95 per cent to be 928 lb per hr. Hence

$$\frac{x}{1.00} = \frac{928}{0.95}, \quad \text{and} \quad x = \frac{928}{0.95} = 977 \text{ lb per hr}$$

the amount condensing in the supply main then is

$$(977)(0.05) = 48.8 \text{ lb per hr}$$

From the Steam Tables, the amount of heat in each pound of condensate returning to the boiler at a temperature of 228° is 196.2 Btu per lb. We have already found the amount of heat in the 48.8 lb of steam as it leaves the boiler, before condensing into steam. It is 1088.2 Btu per lb. Therefore the amount lost from the supply main is

$$48.8(1088.2 - 196.2) = 43.500 \text{ Btu per hr}$$

which, added to the 900,000 Btu per hr given up to the air around the heating coil, gives 943,500 Btu per hr as the total heat output of the boiler. Therefore, this amount of heat must be transferred from the combustion chamber into the water in the boiler. To find the gallons per hour of oil required,

$$\frac{943{,}500}{(140{,}000)0.63} = 10.7 \text{ gal per hr}$$

Working out the oil requirements according to the other method, the amount of heat leaving the boiler is

$$(977)(1088.2) = 1{,}063{,}000 \text{ Btu per hr}$$

The amount of heat returning in the condensate from the supply main is

$$(48.8)196\ 2 = 9580 \text{ Btu per hr}$$

The amount of heat returning in the condensate from the heating coil is

$$(928)(118) = 109{,}500 \text{ Btu per hr}$$

The difference between the amount of heat leaving the boiler and the total of the amounts returning to the boiler will evidently be the amount of heat that must be supplied by the oil burner in order that the process may continue.

$$\text{Heat returned} = 109{,}500 + 9580 = 119{,}080 \text{ Btu per hr}$$

$$1{,}063{,}000 - 119{,}080 = 943{,}920 \text{ Btu per hr}$$

(We obtained a very slightly different result in method 1. The difference is due to arithmetic and slide-rule work.)
Then,

$$\frac{943{,}920}{140{,}000(0.63)}\ 10\ 7 \text{ gal per hr}$$

4. First, 30 psig = 45 psia. Next we must assume a reasonable temperature at which the warm condensate (condensed steam) will leave the condenser. Since we have 80° cooling water, it is reasonable to assume that the condensate may be cooled to about 100°F. Now from the Steam Tables,

$$\text{Steam at 45 psia and 300°F} - h = 1185\ 6 \text{ (from page 566, Table III}^1\text{)}$$

$$\text{Water at 100°F} - h_f = 68 \text{ (from page 556, Table I)}$$

The difference between the above two heat-content values is the heat given up in the condenser *per pound of steam condensed.* Multiplying this difference by the number of pounds of steam condensed per minute gives the Btu per minute which the cooling water must remove. Let us call this Q. Then

$$Q = 10(1185.6 - 68) = 11{,}176 \text{ Btu per min}$$

But for the cooling water, Q is as follows:

$$Q = cW(T_2 \times T_1)$$

$$Q = 11{,}176$$

$$c = 1$$

$$W = 25 \text{ gpm } (8.33) = 208.3$$

$$T_1 = 80°\text{F}$$

$$T_2 = ?$$

Substituting numbers for letters, we have

$$11{,}176 = 1(25)(8.33)(T_2 - 80)$$

$$11{,}176 = 208(T_2 - 80)$$

[1] Tables with roman numerals are given in the Appendix.

From here you may use either one of two methods to solve this equation. This one is best—divide both sides by 208, thus:

$$\frac{11{,}176}{208} = T_2 - 80 \qquad \frac{11{,}176}{208} + 80 = T_2 \qquad 53.7 + 80 = T_2 \qquad T_2 = 133.7°\text{F}$$

The other solution of the equation is given below, but is longer and hence not advisable:

$$11{,}176 = 208(T_2) - 208(80)$$

$$11{,}176 = 208(T_2) - 16{,}640$$

$$\frac{11{,}176 + 16{,}640}{208} = \frac{27{,}816}{208} = T_2 \qquad T_2 = 133.7°\text{F}$$

CHAPTER VII

1. *a*. Opposite 0°F, Table V, page 571, read values shown at left below. Values to the right are caculated as indicated.

Air at 0°, 100% Relative Humidity	Air at 0°, 55% Relative Humidity
Vapor pressure −0.03773 in. Hg	0.55(0.03773) = 0.0207 in. Hg
Grains per lb = 5.5	0.55(5.5) = 3.02 grains per lb From Table IV, page 570, the dew point is −11°
Specific volume dry = 11.58 Specific volume saturated = 11.59 Difference = 0.01	Since the air is more than 50% relative humidity, call specific volume 11.59 cu ft per lb
Sensible heat = 0.00 Total heat = 0.832 Difference = 0.832	The difference is latent heat if 100% relative humidity; 0.55(0.832) = 0.46 actual latent heat. Actual total heat is then 0.00 + 0.46 = 0.46 Btu per lb From Table IV, page 570, by interpolation, this corresponds to −1.34° wet bulb

b. To get the properties at 110°F, first note the following:

Dew point, latent heat, grains per pound, and the pounds per minute of air, vapor pressure. *Do not change*, because the coil adds *sensible heat only* to the air.

Now, opposite 110°F, Table V, page 575, read the following for 100 *per cent saturated air* at this temperature:

Vapor pressure	= 2.59 in. Hg
Grains per pound	= 413.3
Specific volume dry	= 14.35
Specific volume saturated	= 15.71
Difference	= 1.36

Sensible heat 26.4 Btu per lb. (This is correct for the answer.)
Using vapor pressures, the relative humidity is found to be 0.8 per cent; using absolute humidity (grains per pound) relative humidity may be calculated as 0.74 per cent.

This disagreement is not great and occurs because the relative humidity is so very low. In practice the air would be referred to as "less than 1 per cent relative humidity." All results so far should now be tabulated in the table which presents the final answer. The table will be complete except for the three 110° values marked *. These latter three are found as follows:

Total heat = sensible heat + latent heat = 26.86 Btu per lb
26.86 Btu per lb total heat corresponds to 60.8° wet bulb (Table V, page 573)
Specific volume = 14.35 + 0.008(1.36) = 14.25 + approximately 0.01 = 14.36

	Outdoor air	After heating
Dry bulb, °F	0	110
Wet bulb, °F	−1.34	60.8*
Dew point, °F	−11	−11
Relative humidity, %	55	Less than 1
Total heat, Btu per lb	0.46	26.86*
Sensible heat, Btu per lb	0.00	26.40
Latent heat, Btu per lb	0.46	0.46
Grains per pound	3.02	3.02
Specific volume, cu ft per lb	11.59	14.36*
Vapor pressure, in. Hg	0.0207	0.0207
Pounds per minute of air	$\frac{10{,}000}{11.59} = 863$	863

Each pound of air increases in total heat 26.4 Btu per lb

$$Q = 26.4(863 \text{ lb}) = \text{Btu per min added to the air.} \quad \textit{Ans.}$$

2. Opposite 60 in the grains per pound column, Table V, page 573, we see that the dew point leaving the washer is about 53.2°F. Let us take the dew point as 53°, which corresponds to 59.74 grains per lb and is accurate enough. *Ans.* On this line in the table, read the sensible and total heats, and the difference is the latent heat.

Total heat	21.95 Btu per lb
Sensible heat	12.71 Btu per lb
Latent heat	9.24 Btu per lb

The latent heat of the air entering the washer was 0.46 Btu per lb; the washer increased this by

$$9.24 \times 0.46 = 8.78 \text{ Btu per lb}$$

Since the *total* heat of the air did not change in the air washer, the *sensible* heat must have decreased by the amount that the *latent* heat increased. *Ans.*

Sensible heat entering washer (from Prob. 1)	26.40 Btu per lb
Less	8.78 Btu per lb
Sensible heat leaving washer	17.62 Btu per lb

This corresponds to 73.5° dry bulb (Table V, page 573). *Ans.* Calculating relative humidity by Eq. (3), page 125, we get

$$\% \text{ R.H.} = \frac{60}{124.5}(100) = 48.2\% \quad \textit{Ans.}$$

The Btu per minute of latent-heat increase in the air washer is

$$8.78(863 \text{ lb per min}) = 7577 \text{ Btu per min} \quad \textit{Ans.}$$

The water added in the air washer in pounds per minute is found thus

$$\frac{\text{Grains per pound increase (pounds air per minute)}}{7{,}000}$$

The increase in absolute humidity is 60 − 5.5 = 54.5 grains per lb. Then

$$\frac{54.5(863)}{7{,}000} = 6.75 \text{ lb per min} \quad \textit{Ans.}$$

This is a little over 3 qt. per min.)

3. From Table V, page 571, and the equation "latent heat = total heat − sensible heat"

Latent heat at 95°, 100% saturated = 63.05 − 22.80 = 40.25 Btu per lb
Actual latent heat of our air = Total heat of our air, at 70° wet bulb, minus sensible heat of dry air at 95°
Actual latent heat of our air = 33.95 − 22.80 = 11.15 Btu per lb

$$\% \text{ R.H.} = \frac{11.15}{40.25}(100) = 27\ 7\%$$

From the table grains, per pound at 95°, 100% saturated = 255.6
Actual absolute humidity of our air = 0.277(255.6) = 70.8 grains per lb
From Table V, page 573, this is a dew point of 57.6°.

Since the air is cooled only at 60°, which is *not down to its dew point*, the absolute humidity, dew point, and associated properties *do not change.* Therefore, we do not have to calculate all the properties of the air on both sides of the coil. Instead, we can say with certainty that the following properties of the air are constant:

Absolute humidity
Dew point
Latent heat
Vapor pressure *Ans.*

No moisture condenses. *Ans.*

4. The dew point and relative humidity are found by the same method used in Prob. 3, in which we also started by knowing dry bulb and wet bulb. Total and sensible heats are read from the table, and latent heat is the difference. Specific volume is obtained in the usual way. We then make the table on page 628, which includes the answer to the first part of the problem. (Blanks appear in the table because we do not need certain properties in this problem.)

The table shows the "pounds per minute of air," meaning by the word "air," "dry air plus vapor." We obtain the number of pounds of dry air per minute (not including the weight of the vapor) by the method shown on page 113.

$$\frac{7{,}000}{7{,}108}(69.7) = 68.6 \text{ lb } \textit{dry air} \text{ per minute}$$

	Before cooling	After cooling	Change
Dry bulb, °F	95	55	
Wet bulb, °F	78	53	
Dew point, °F	69.5	51.6	
Relative humidity, %	46 2	88	
Grains per pound	108	56.6	−51.4
Total heat, Btu per lb	41.42	21.95	−19.47
Sensible heat, Btu per lb	22.80	13.19	−9.61
Latent heat, Btu per lb	18.62	8.76	−9.86
Specific volume, cu ft per lb	14.62		
Vapor pressure, in. Hg			
Pounds per min of air	69.7		

Ans

We need this value because our total-heat and absolute-humidity values are "per *pound of dry air* plus its vapor."

$19.47(68.6) = 1{,}336$ Btu per min removed *Ans.*

$\frac{9.61}{19.47}(100) = 49.45\%$ sensible heat *Ans.*

$\frac{9.86}{19.47}(100) = 50.55\%$ latent heat *Ans.*

$\frac{51.4(68.6)}{7{,}000} = 0.504$ lb water per minute removed, or about ½ lb per min *Ans.*

5.

	Absolute Pressures		
	In. Hg	Mm Hg	Psi
A vacuum of 28 in. Hg =	1.9	48.3	0.943
1 psia =	2.03	51.6	1.0
1 psig =	31.95	811.5	15.7

From page 110:

$$1 \text{ in. Hg} = 25.4 \text{ mm Hg}$$

$$1 \text{ psi} = 2.03 \text{ in. Hg}$$

6. Sensible heat at 80°F = 19.19 Btu per lb
Total heat at 67° wet bulb = 31.51 Btu per lb

Actual latent heat of our air is the difference between the above.

$$(\cancel{30}.51 - 19.19) = \cancel{11}.32$$

Latent heat of air 100% saturated at 80°F = (43.51 − 19.19) = 24.32

$$\% \text{ R.H.} = \frac{11.32}{24.32}(100) = 46.5\% \quad \textit{Ans.}$$

At 80°F, specific volume of air saturated = 14.08
At 80°F, specific volume of dry air = 13.59
Difference 0.49

$$0.465(0.49) = 0.23 \text{ (approximately)}$$

$$13.59 + 0.23 = 13.82 \text{ cu ft per lb} \quad \textit{Ans.}$$

7. We get the total weight of air plus vapor using the specific volume of 13.82 as figured in Prob. 6.

$$\frac{40{,}000}{13.82} = 2{,}890 \text{ lb of air plus vapor}$$

100% saturated air at 80° contains 155.5 grains per lb; our air contains 0.465(155.5) = 73.2 grains per lb. (Hence our air has a dew point of 58.5°F.)

Using the method on page 113, we get

$$\frac{7{,}000}{7{,}073.2}(2{,}890) = 2{,}870 \text{ lb dry air}$$

$$\frac{78.7}{7{,}073.2}(2{,}890) = 32 \text{ lb vapor}$$

8. Because of the low partial pressure of the water vapor in air, some water will evaporate from the wet-bulb wick, provided only that the air is moving past the wick at a reasonable speed. This evaporation of water absorbs heat and cools the water that remains on the wick. As soon as the wick drops below the dry-bulb temperature of the air, sensible heat flows from the air to the wick. The lower the vapor pressure of the water vapor in the air being measured, the faster water evaporates from the wick. The colder the wick becomes, the faster the sensible heat flows to it from the air. Thus the wet-bulb thermometer is affected by both the sensible heat and the latent heat in the air.

9. Your chart to the left of the saturation curve should resemble the Peerless psychrometric chart, page 134, except that you will have only a few wet-bulb lines extended. The numerical total heat values for your wet-bulb lines should be the same as those shown in columns (6) and (7) of Table IV, page 570.

10. At 0°F, 100% saturated, directly from the table we read the values below:

Dry bulb..................	0°F	Grains per pound	5.5
Wet bulb..................	0°	Total heat.......	0.83 Btu per lb
Dew point..................	0°	Sensible heat....	0.00 Btu per lb
Relative humidity...........	100%	Specific volume..	11.59 cu ft per lb

We get the weight of air plus vapor and then the weight of the dry air by the same method as used in Prob. 7.

$$\frac{10{,}000}{11.59} = 863 \text{ lb of air plus vapor, per minute}$$

$$\frac{7{,}000}{7{,}005.5}(863) = 862 \text{ lb of dry air}$$

From the above you should learn and remember that the former value, 863 lb, may be used in practical work, as the error is very small and is in the conservative direction. For instance, if we used 863 in figuring the heat (see below), our answer would be slightly *too high*, which is an error in the *safe* direction.

Grains per pound at 70° saturated = 110.2, and 0.4(110.2) = 44.1 grains per lb

Total heat of air at 70° dry bulb and 40% relative humidity = sensible heat at 70° + 0.4 (latent heat at 70°, 100% saturated)

Total heat at 70° dry bulb, 40% relative humidity

$$= 16.79 + 0.4\,(33.96 - 16.79)$$
$$= 16.79 + 0.4(17.17)$$
$$= 16.79 + 6.87 = 23.66$$

Then

$$862(23.66 - 0.83) = 20{,}500 \text{ Btu per min} \quad \textit{Ans.}$$

$$\frac{862(44.1 - 5.5)}{7{,}000} = 4.75 \text{ lb per min of water} \quad \textit{Ans.}$$

CHAPTER VIII

1. The vapor pressure of a liquid depends upon its temperature. If the absolute pressure over the surface of a liquid is decreased to a point where this absolute pressure is lower than the vapor pressure of the liquid, some of the liquid will change state and become a gas. Such a change of state is often referred to as "flashing to a vapor." Since the change of state of any substance from a liquid to a gas requires latent heat to increase the kinetic energy of the molecules, the heat content of a body of liquid is decreased when a portion of the liquid flashes to a vapor.

As an example of cooling of water by the method described above, water at a temperature of 102°F has a vapor pressure of approximately 1.00 psi. If water at 1020 is in a chamber in which the absolute pressure can be reduced below 1.00 psi, some of the water will flash to steam and the remainder of the body of water will be cooled. If mechanical means are provided to remove the vapor and to continue lowering the absolute pressure to about 0.25 psi, the water will be cooled to about 59°F.

2. Because of the large volume of vapor that must be removed from a chamber of the type described in Prob. 1, a reciprocating compressor is unsuitable for this service. (Note the large volume of vapor in your solutions to Prob. 4 and Prob. 5*d*.) Centrifugal compressors, sometimes called "turbo-compressors," are satisfactory for handling these large volumes of water vapor. Steam-jet compressors, sometimes called "thermal compressors," are also satisfactory for this service.

3. From Table I, page 555:

$$h_{fg} \text{ at } 60° = 1059.9$$

$$h_{fg} \text{ at } 70° = 1054.3$$

$$\text{Average } h_{fg} = 1057.1$$

$$h_{fg} \text{ at } 65° = \text{exactly } 1057.1, \text{ from the table}$$

(This is a coincidence, as the two values might have differed slightly.)

We shall use $h_{fg} = 1{,}057$ as the average latent heat of vaporization for this evaporation process.

$$Q = cW(T_2 - T_1)$$

$$Q = 1\,(1{,}000)(70-60) = 10{,}000 \text{ Btu per min}$$

$$\frac{Q}{h_{fg}} = \text{pounds of water evaporated per minute}$$

$$\frac{10{,}000}{1{,}057} = 9.46 \text{ lb of water per minute}$$

Since our h_{fg} value is not exact, let us call our answer 9.5 lb of water per minute *Ans.*

4. From Table I, page 555,

$$v_g \text{ at } 40° = 2444 \text{ cu ft per lb}$$

$$2(2{,}444) = 4888 \text{ cfm} \quad \textit{Ans.}$$

5. From Table I, page 555, we get *a* and *b*:

(*a*) h_{fg} at 50°F = 1065.6, say 1066 Btu per lb *Ans.*

(*b*) v_g (dry saturated) at 50°F = 1,703 cu ft per lb *Ans.*

(*c*) $\dfrac{Q}{h_{fg}} = \text{pounds evaporated}$

$Q = 2{,}000$ Btu per min for a 10-ton machine

$\dfrac{2{,}000}{1{,}006} = 1.875$ lb water per minute evaporated *Ans.*

(*d*) $v_g \times$ pounds per minute = cfm. Hence

$$\text{cfm} = V_g(1.875)$$
$$= 1{,}703(1.875) = 3{,}195 \text{ cfm of vapor} \quad \textit{Ans.}$$

6. In Fig. 2, page 139, a vacuum pump or air ejector is shown connected to the shell of the condenser within which the water vapor is condensed. A similar air ejector is required on a steam-jet refrigeration machine, a two-stage ejector being shown on the condenser of the steam-jet machine illustrated in Fig. 4, page 145. These ejectors are also called "noncondensible ejectors." The function of all such ejectors is to remove any noncondensible gases such as air, which are bound to be present in small quantities in the steam that is condensed. If these noncondensible gases were not removed from the condenser, they would build up in quantity and raise the absolute pressure within the condenser.

7. The answer to this question should consist of a simplified sketch illustrating the equipment shown in Fig. 4, page 145.

8. The answer to this question should consist of a table presenting the data shown in Figs. 8 and 9, page 150. The tabular data may be approximate, as the two graphs differ, but they differ only slightly.

9. Steam-jet machines should not be used because of the large quantity of condenser water required. Since the problem states that no cooling tower could be used, this condensing water would have to be city water, the cost of which would be excessive. Furthermore, the installation cost of the steam-jet plant would be higher than that of an ordinary mechanical plant. A centrifugal water-vapor refrigeration plant would also cost more than an ordinary mechanical plant. Therefore, if the owner of the theater did not plan to operate the plant for 15 years or more, so that operating-cost savings would make the higher first cost worth while, the centrifugal water-vapor plant could not be recommended either. On the other hand, if the theater were a new building and if the management planned to operate for 15 to 20 years, with high-type operating personnel, the centrifugal water-vapor plant might offer sufficiently attractive operating savings to justify its higher initial cost.

10. The condenser must condense both the *main-jet steam* and the *flash-tank steam.* First we must get the number of pounds per minute of water evaporated in the flash

tank. We use the formula used in Probs. 3 and 5.

$$Q = 10 \text{ tons } (200) = 2{,}000$$

$$\frac{2{,}000}{1{,}065} = 1.878 \text{ lb water evaporated per minute}$$

The problem states that for each pound of such vapor drawn out of the evaporator, the jet needs 2 lb of main-jet steam.

$$2(1.878) = 3.756 \text{ lb per min main-jet steam}$$

Then, adding flash-tank steam + main-jet steam:

$$1.878 + 3\ 756 = 5\ 634 \text{ lb per min.} \quad \textit{Total steam} \text{ to be condensed}$$

The problem gives h_{fg} in the condenser to 1,040 Btu per lb, which corresponds (Table I, page 555) to a *condenser temperature of* 95°F. *Ans.*

$$1{,}040(5.634) = 5{,}860 \text{ Btu per min} \quad \textit{Ans.}$$

which is

$$\frac{5{,}860}{10} = 586 \text{ Btu per min per ton} \quad \textit{Ans.}$$

(This solution neglects the possibility that the steam entering the condenser may be slightly superheated.)

CHAPTER IX

1. You will find the properties you want in Table XI, page 582. Be careful to note that the pressure in this table is *absolute* pressure. Note also that the specific volume of the liquid ammonia is actually given as cubic feet per pound (v_f), whereas some refrigerant tables give the density of liquid in pounds per cubic foot ($1/v_f$). Regarding heat content, note that the ammonia table shows latent heat as h, whereas we ordinarily called this quantity h_{fg}. The numerical answers to Prob. 1 are included in Prob. 5 below.

2. The answers to this problem are given in Table XIII, page 591. Be careful of the precautions indicated in the discussion of Prob. 1 above. The numerical answers to Prob. 2 are included in Prob. 5 below. Carbon dioxide cannot exist as liquid above 87.8°F regardless of how high the pressure, because this temperature is its critical point.

3. The answers to this problem are given in Table XIV, page 593. Note that the pressures in this table are gauge pressures, although we ordinarily use absolute pressures in tables of this type. This table does not give h_{fg}; instead, you must find it by using the formula $h_{fg} = h_g - h_f$. The numerical answers for Prob. 3 are tabulated in Prob. 5 below.

4. The properties of methyl chloride are given in Table XVI, page 596. In using this table, observe the precautions mentioned in the discussion of Prob. 1 above. In addition, note that the density of the liquid is shown in pounds per cubic foot and is then incorrectly labeled V_f. This column to be consistent with the other refrigerant tables should be labeled "$1/v_f$."

The sulphur dioxide properties given are in Table XVII, page 597, the use of which should be easy after having worked with all the previous tables. The numerical answers to Prob. 4 are included in the table below:

5.

Refrigerant	At saturation		Specific volume		Latent heat
	Temp	Gauge pressure	v_g	v_f	h_{fg}
Ammonia	100°	197.2	1.42	0.0275	478
	40°	58.6	3.97	0.0253	536
Carbon dioxide	100°	Critical temperature + 87°F			
	40°	553.1	0.144	0.0179	95.0
Freon	100°	117	0.319	0.01272	57.4
	40	37	0.792	0.01161	65.7
Methyl chloride	100°	104.1	0.85	0.0181	156.3
	40°	27.9	2.31	0.01695	172.4
Sulphur dioxide	100°	69.8	0.93	0.01204	137.2
	40°	12.4	2.89	0.01113	159.3

Note: Where tables gave $1/v_f$, the table value has been changed so all values above are v_f. Where the table did not give h_{fg}, the above values are calculated by $h_{fg} = h_g - h_f$.

6. From Table XV, page 594, the lower left corner (values are labeled simply v and h in the table):

	v_g	h_{fg}
Freon 12 at 140 psia $T = 130°F$	0.323	93.28
Freon 12 at 140 psia $T = 120°F$	0.313	91.27

7. From the standpoint of thermodynamics, or heat and heat flow, an ideal refrigerant should exhibit the following characteristics: (*a*) large latent heat, (*b*) a resonable evaporator pressure corresponding to the evaporator temperatures desired, (*c*) at ordinary condensing temperatures (90 to 110°) a corresponding pressure that is not too high. (So long as the condensing temperature is not more than about five times as high as the evaporator temperature, the condensing temperature is not considered unreasonably high.)

Carbon dioxide has a critical temperature of 87.8° which corresponds to a pressure of about 1,070 psi. (Even with very cool condensing water, carbon dioxide will have a condensing pressure of over 700 to 800 psi.) Therefore, carbon dioxide is useless as an air-conditioning refrigerant.

8. The thermodynamic properties of Freon and methyl chloride are clearly illustrated at typical evaporator and condenser conditions by the table in Prob. 5 above. Both refrigerants are satisfactory as to thermodynamic properties. From the table in Prob. 5, it can be deduced that the displacement per ton is nearly the same for Freon and methyl chloride. Table IX, page 580, shows in column (1) that this is true.

As to general properties (see the summary list in Table 1, page 157) both Freon and methyl chloride are satisfactory in most respects. Methyl chloride has the disadvantage of producing combustible and explosive mixtures with air, while Freon is not considered inflammable. When decomposed by very high temperatures, Table VI, page 578, shows that Freon 12 must be ten times more concentrated than methyl chloride to kill or seriously injure guinea pigs in 2 hr. These latter reasons are probably responsible for the fact that methyl chloride is outlawed for air conditioning in many cities.

9.

	Ammonia	Freon	Water	Air
Proper condensing			Very	
Temperature and pressure	+	+	low	No
			Absolute	
Proper evaporator			pressure	
Temperature and pressure	+	+		No
Large latent heat	+	+	+	?
Nontoxic	No	+	+	+
No bad odor	No	+	+	+
Noncombustable	No	+	+	+

+ Indicates satisfaction.
? Indicates some question.

The above table indicates that ammonia is better thermodynamically than Freon because of its high latent heat. The compressor displacement per ton and the weight of refrigerant in pounds per minute are smaller for ammonia than Freon. Water has the highest latent heat but fails to meet the thermodynamic qualifications of an ideal refrigerant because its evaporator pressure and condenser pressure are both so nearly absolute zero. Because water operates at very low absolute pressures, its specific volume is extremely large, and huge quantities of vapor must be removed from the evaporator for each ton of refrigerating capacity. The latter is the biggest disadvantage of water as a refrigerant. The table above indicates, as you know, that air cannot be easily liquified by compressing to a reasonable pressure. Liquid air can be produced, but even at an evaporator pressure of 100 psi it boils at a temperature much lower than the lowest temperature shown for dry ice in Table XIII, page 591.

10. The answer to Prob. 5 consists of information taken from the tables giving the properties of refrigerants. From this information or from Table IX, page 580, it is possible to conclude the following: Ammonia would give roughly 1.8 times as much refrigerating capacity as Freon or methyl chloride. The latter two refrigerants would give substantially the same refrigerating capacity. The power requirements *per ton* would be approximately the same for the three refrigerants [see columns (7), (8), and (9) of Table IX, page 580]. Although Freon 12 is not illustrated in these columns, columns (1) to (6) in the table indicate that Freon 12 would be nearly the same as ammonia and methyl chloride for the items in columns (7), (8), and (9). Since its refrigerating capacity is greatest, ammonia would require the greatest amount of power to drive the constant-speed compressor discussed in this problem.

CHAPTER X

1. Refrigeration is the production of a lower temperature than the prevailing outdoor temperature within some enclosed space. The unit in which refrigeration or refrigerating effect is measured is called "the ton of refrigeration." One ton of refrigeration is the equivalent of 200 Btu per min or 12,000 Btu per hr of heat removal

2. $Q = cW(T_2 - T_1)$
$Q = (1)(300)(8.33)(54° - 48°)$
$Q = 15{,}000$ Btu per min

$$\frac{15{,}000}{200} = 75 \text{ tons} \quad \textit{Ans.}$$

3. The refrigeration compressor performs two functions. (1) It draws away vapor from the evaporator maintaining the desired low-side pressure in the evaporator. (2) It then compresses this low-side vapor to the necessary higher condensing pressure and drives the compressed vapor into the condenser.

4. An expansion valve is a pressure-reduction mechanism or barrier between the high and low sides of the refrigeration cycle, at the entrance to the evaporator. The expansion valve also serves to a certain extent to regulate the flow of refrigerant into the evaporator. A thermostatic expansion valve is operated by two sets of bellows. The first set of bellows acts to hold the valve closed if the low-side pressure is abnormally high. Provided this first bellows is satisfied by a normally low evaporator pressure, the valve is permitted to operate under the control of a second bellows. This second bellows tends to open the valve if the refrigerant vapor in the suction line becomes superheated more than a predetermined number of degrees. (Such valves are normally set to 5 to 10° of superheat.)

5. Refrigerating effect per pound of refrigerant is the heat content of the gas leaving the evaporator minus the heat content of the liquid entering the evaporator.

From Table XIV, page 593. h_g at 45° = 83.25 Btu per min
From Table XIV, page 593. h_f at 85° = 27.48 Btu per min
55.77 Btu per min

$$\frac{200}{55.77} = 3.59 \text{ lb of Freon per minute per ton} \quad \textit{Ans.}$$

6. From Table XV, page 594, upper left corner, we have

Evaporator A	Evaporator B
v, at 50 psia and at 45°F = 0.831	v, at 50 psia and at 55°F = 0.852
(The above is obtained by interpolating 7/12 of the way from 0.817 to 0.842)	(The above is obtained by interpolation 1/2 way between the v values at 50° and at 60°.)

7. Volume of liquid Freon 12 in evaporator *in cubic feet* is 0.7(0.25) = 0.175 cu ft

At 90°, Freon 12 density, or $1/v_f$, = 80.1 lb per cu ft
At 50°, Freon 12 density, or $1/v_f$, = 84.9 lb per cu ft

Weight of liquid in evaporator = 0.175 (84.97 = 14.85 lb
Volume of this Freon 12 at liquid receiver conditions = 14.85/80.1 = 0.185 cu ft
Adding 20% safety factor, we have

$$1.20(0.185) = 0.222 \text{ cu ft} \quad \textit{Ans.}$$

or

$$0.185 + 0.20(0.185) = 0.185 + 0.037 = 0.222 \text{ cu ft} \quad \textit{Ans.}$$

(A receiver 1 ft 2 in. long by 6 in. diameter would be about right. See if you can verify this.)

8. The specific-volume values below are the same as those given in the solution to Probs. 1 to 5, Chap. IX. The heat contents are looked up in the tables of properties of refrigerants.

	(1) h_g at 40°	(2) h_f at 100°	(3) $h_g - h_f$	(4) Lb per min $= \dfrac{2{,}000}{h_g - h_f}$	(5) Suction line spec vol = V_g at 40°F	(6) Q = cfm = (lb per min) times (V_g)	(7) Pipe area, sq ft $A = \dfrac{Q}{V}$
Ammonia ...	623	155	468	4.27	3.97	16.95	0.0113
Freon.......	82.7	31.2	51.5	38.8	0.792	30.7	0.0205
Methyl chloride...	199	49	150	13.3	2.31	31	0.0207
Sulphur dioxide....	185	47	138	14.5	2.89	41.9	0.028

Column (6) presents the *answer* requested. The various columns are explained below.

Columns (1), (2), and (5) give data from refrigerant tables.

Column (3), see page 162 and Fig. 1, page 162.

Column (4), see page 165.

Column (6). We are multiplying the *pounds per minute* of refrigerant by the specific volume in *cubic feet per pound.* The result is *cubic feet per minute.*

Column (7). Not needed in this problem, but needed in solving Prob. 9 below.

9. From column (7) in the table in Prob. 8, we get the required area of the suction line in *square feet* to be 0.0205 *sq ft.* (Note that this is in square *feet.*) If we multiply this by 144, we get the area in square *inches.* Using our formula

$$A = \frac{\pi d^2}{4} \text{ we get}$$

$$144(0.0205) = \frac{\pi d^2}{4}$$

$$d^2 = 144(0.0205)(4) = 3.76 \text{ sq in.}$$

$$d^2 = \sqrt{3.76} = 1.94 \text{ in.}$$

(We would of course use a $2\frac{1}{8}$-in. outside diameter refrigerant pipe, the *next larger* size commercially available, which has an inside diameter of approximately 2 in.) As to the liquid line, v_f at 100° is found (in Table XIV, page 593) to be 1/78.8, or 0.0127 cu ft per lb. Then we multiply this by the pounds per minute of Freon from Prob. 8.

$$0.0127(38.8) = 0.493 \text{ cfm of liquid}$$

$$Q = A(V), \text{ so } A = \frac{Q}{V} = \frac{0.493}{200} = 0.00247 \text{ sq ft}$$

then

$$144(0.0247) = \frac{\pi d^2}{4}$$

$$d = \sqrt{\frac{4\ (144)(0.00247)}{\pi}} = \sqrt{0.452} = 0.672 \text{ in.}$$

For this we would use ¾ in. outside diameter tubing.

10. From the table in Prob. 8, we see that sulphur dioxide, having the largest number of cubic feet per minute of gas in the suction line, requires the largest suction line.

We cannot say which would need the largest compressor displacement, based only on the work in Probs. 8 and 9. A compressor cylinder has a theoretical displacement, equal *per stroke* to area of the piston multiplied by the length of the stroke. Actually, a given cylinder pumps *fewer* cubic feet per minute of gas than its theoretical displacement. The ratio of actual displacement to theoretical displacement is called "volumetric efficiency," which depends on the compressor clearance, compressor ratio, speed, valve design, and the nature of the gas being pumped. Neglecting this, which we do not know in this problem (and which would be different for the different gases even with the same compressor), we can safely make the general statement that ammonia would require the smallest displacement, SO_2 the largest, with Freon and methyl chloride in between and nearly the same.

CHAPTER XI

1. A positive-displacement compressor is one that draws the intake gas into a space or chamber, this chamber then becoming smaller and smaller, positively compressing the gas and forcing it out the discharge. The Norge Rollator, the rotary-vane-type compressors, and all ordinary reciprocating compressors are examples of positive-displacement units. The centrifugal compressors operate by throwing the vapor off the periphery of a very high-speed rotor. They are not capable of compressing to high pressures and ordinarily must have three or more separate centrifugal compressors or "stages." Stage 1 discharges into the inlet of stage 2, etc. The Carrier centrifugal compressor using Carrene and the centrifugal water-vapor compressor are examples of this type.

2. Page 178 and Fig. 1 describe the Rollator compressor. If the end of the wedge tends to wear because of friction the spring behind the wedge compensates for this wear and the compressor's efficiency is not imparied.

3. Page 181 and Fig. 2 illustrate a vane-type rotary compressor. The vanes are held outward away from the center by centrifugal force when the rotor is spinning. When the rotor stops, the vanes need no longer be held outward, since the check valve shown in Fig. 2 prevents the high-pressure hot gas from passing backward through the compressor into the suction line. Hence the vanes *need* no springs behind them at all. (Some designs of rotary-vane-type compressors use very light springs behind the vanes.)

4. Page 183 and Figs. 3 and 4 describe and illustrate low-side-float and high-side-float expansion valves.

5. Page 184 and Fig. 5 describe and illustrate the automatic expansion valve. Page 189 and Fig. 8 describe and illustrate the capillary-tube expansion valve.

6. The thermostatic expansion valve is shown in a simple sketch in Fig. 6, while Fig. 7 shows a drawing of such valves as actually constructed. The thermostatic valve is ideal for dry expansion air-conditioning evaporators because under conditions of high load the valve opens and permits the greatest flow of refrigerant. Under conditions of lower than normal load, the valve shuts down on the supply of refrigerant. Regardless of load, dry gas, superheated a few degrees, is assured in the suction line, yet this gas does not become 20 to 30° superheated under conditions of high-heat load on the evaporator.

7. Because of the formation of flash gas and the boiling of the refrigerant and also because of the pressure difference due to frictional resistance in the tubes, it is difficult to distribute low-side liquid refrigerant to all tubes of an extended surface coil by means of a liquid header of the conventional type. Hence it has been difficult to operate such coils even with the header horizontal, and almost impossible if the header were vertical. The use of a chamber out of which tubes slightly larger than capillary tubes lead to each individual row of tubes of the coil, as illustrated in Figs. 9 and 9*a*, overcomes this difficulty and eliminates the header. It should be noted that these modified capillary tubes do not have sufficient resistance to provide the pressure drop between the high and low side so that an expansion valve is needed in the liquid line at the entrance to the reservoir chamber.

8. Figure 13, page 194, provides a complete answer to this question. In words we may say that the *absorber*, *strong aqua pump*, and *generator* serve the same function as the compressor in a mechanical system. Otherwise the two systems are identical.

9. The actual displacement of the compressor, *i.e.*, the number of cubic feet per minute of low-side gas which it can remove, determines the suction pressure in a flooded-evaporator mechanical-compression system with a *fixed evaporator heat load.* With the constant Btu per hour input into the evaporator, which we have assumed, a fixed number of pounds of refrigerant would be vaporized in the evaporator per unit of time. Obviously the rate at which this vapor is removed will determine the low-side pressure. Under the same conditions in an absorption system, the low-side pressure is determined by the ability or capacity of the absorber to absorb the low-side vapor, *i.e.*, the rate at which the absorber can remove the vapor from the evaporator.

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