HEAT PUMPS
AND
THERMAL COMPRESSORS

HEAT PUMPS
AND
THERMAL COMPRESSORS

by

S. J. Davies, D.Sc. (Eng.), Ph.D., M.I.Mech.E.

Professor of Mechanical Engineering, University of London King's College

Four Lectures delivered at King's College, London

LONDON

CONSTABLE & COMPANY LTD

LONDON
PUBLISHED BY
Constable and Company Ltd
10–12 ORANGE STREET, W.C.2

INDIA
Orient Longmans Ltd
BOMBAY CALCUTTA MADRAS

CANADA
Longmans, Green and Company
TORONTO

SOUTH AFRICA
Longmans, Green and Co., Ltd.
CAPE TOWN

First Published 1950

PRINTED IN GREAT BRITAIN BY W. & J. MACKAY & CO., LTD., CHATHAM

FOREWORD

The lectures published here, though originally intended for my more advanced undergraduates, were amplified, in view of the general interest among practising engineers and the importance of the subject, to form one of the Courses of Public Lectures at this College.

The principles set out here owe almost everything to the contributions, in the early part of the last century, of Kelvin, Joule and Stirling in this country, although most of the practical applications of these principles have been made abroad.

For the convenience of readers less familiar with British Units, equivalent Metric Units are included in the text in most cases.

My sincere thanks are offered : to those authorities mentioned in the text ; to the Institution of Mechanical Engineers, the Institute of Marine Engineers, Messrs. Escher Wyss, Zurich, Messrs. Brown Boveri, Baden, and Messrs. Philips, Eindhoven, for permission to reproduce figures from their publications ; to *Engineering*, for the loan of blocks for Figs. I, 2 to 5, I, 9 to 11, and II, 7 and 9, and, finally, to Mr. P. E. Glikin, one of my research students, for drawing the remaining figures.

S. J. D.

King's College, London.
February, 1949.

CONTENTS

LIST OF ILLUSTRATIONS

LECTURE I

No excuse is necessary in this country at the present time for a course of lectures having for its subject heat economy in any shape or form. But, quite apart from our present difficulties, grave as they are, there are other aspects to consider : even before the recent war, strong tendencies in the development of world trade and industry were forcing us to consider, much more carefully than formerly, possible methods of using our labour force with greater efficiency. Much was accomplished during and between the wars by improved works organisation, while, in addition, the productive capacity of the individual workman, as regards quantity and often quality, was increased by providing him with more mechanical equipment. Much remains to be done in this direction. Consideration of these matters is essential if we, as a country, are to make our best contribution to world production, namely, that of high-grade manufactured goods. It is now commonly realised that such a contribution is necessary in our own interests, if we are to maintain and to improve our average standard of living. In connection with the present subject, it is important to remember that increase of mechanical equipment often involves greater demands for electrical power, demands which can be met either by increasing the production of power or, better, by improving the load factor and thus producing this power with less avoidable waste.

While the efficiency with which electrical power is produced and distributed has improved steadily during the past fifty years, and has now reached a very high level, there are, mostly in branches of industry involving the utilisation of heat energy, many cases in which the most primitive methods are still employed and in which energy continues to be wasted on a very large scale. The main object of the present lectures, therefore, is to put forward certain means of correcting this unsatisfactory state of affairs, namely, by the use of heat pumps and thermal compressors in thermal installations of various kinds. It will, further, be seen that the application of these machines is of importance in relation to the better utilisation of our labour force. It should not need to be emphasised that, with the high level of intelligence to be found in the mass of British people, the employment of labour in any task in which muscular effort can be replaced or reduced by mechanical aids represents avoidable waste.

It is well to reflect for a moment on the historical background of industry in this country and, in particular, of those branches in which power production and utilisation are important. We have been and are pioneers in practical achievement and, although we have continued to make worthy contributions to the science of thermodynamics, as the names Joule, Rankine, Kelvin, Callendar, Parsons, and, more recently, Ricardo and Whittle, remind us, as a people we think first of practical results and only afterwards of the scientific basis : in fact, of the eminent scientists and engineers just mentioned, Callendar is the only one who is not as well known for his practical achievements as for his scientific contributions.

Our insistence upon practical achievement, with which is associated reliability of operation based on sound design and first-class workmanship, has stood us in good stead, but, while these good qualities must continue to be our earnest concern, they are not enough : we are now compelled to analyse more closely our methods in order to ascertain the possibilities of better utilisation of our sources of energy and our labour force. We must, in particular, balance our preference for those simple and direct methods which lead to high reliability against more complicated methods yielding higher efficiency. In the production of electrical energy, improvement in efficiency has gone hand-in-hand with increasing complexity in our installations ; in coal-winning, it is now recognised that mechanical aids, leading also to complexity, are essential to a long-term cheapening of production ; in the home, too, mechanical aids are accepted as essential to economy of effort. In all of these cases a reduction of manual labour has accompanied improved efficiency. It is now most desirable that this acceptance of mechanical complexity, in order to achieve higher efficiency and to economise manual labour, should be extended to thermal operations of all kinds.

The present book is based on a course of four lectures given by the author. In the first lecture, in which a general introduction to the subject is given, the fundamental bases of thermal operations are set out and some of the factors to be considered in connection with heat pumps and thermal compressors are surveyed.

In the second lecture, special attention is given to existing applications of heat pumps employing vapour compression cycles.

In the third lecture, the conditions determining the performance of heat pumps using gases as the working substance and operating on a reversed Joule, or constant pressure, cycle are considered in detail ; this is followed by a discussion of air-conditioning and the application of heat pumps to such installations.

In the last lecture, applications of thermal compressors of the piston and turbo types are described ; this is followed by discussions of the possibilities of utilising the waste heat from oil engine power installations ; consideration is then given to the jet thermo-compressor in a heat pump installation, and this is followed by a description of the Philips hot-air engine ; finally, the present position of heat pump installations is briefly summarised.

As a starting point, while a general acquaintanceship by the reader with the principles of operations of steam installations, internal combustion engines, air compressors and vapour-compression refrigerators will be assumed, a return to certain elementary matters that are fundamental to the present discussions cannot be avoided.

In all thermal processes forming a cycle we are brought back to the Carnot cycle as our ultimate standard of reference. For such a cycle, belonging to a heat engine, and given on a temperature-entropy diagram in Fig. I,

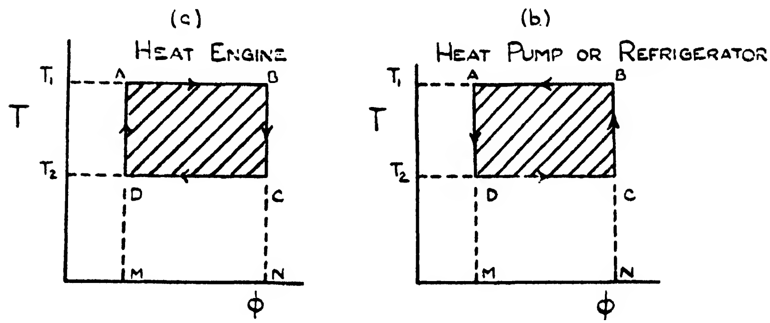

FIG. I—1.

Temperature-Entropy Diagrams of Carnot Cycle.

1(*a*), the heat, Q_1, is all given to the working substance at the highest temperature, T_1, and is represented by the area MABN ; the heat rejected, Q_2, is all given up at the lowest temperature, T_2, and is represented by the area MDCN.

The Work Done, W, expressed in heat units, *by* the working substance in one cycle

$$= \text{Heat received} - \text{Heat rejected},$$

$$\text{that is, ABCD} = \text{MABN} - \text{MDCN},$$

$$\text{or} \quad W = Q_1 - Q_2.$$

On the diagram, $\phi_B - \phi_A = \phi_C - \phi_D = \phi_N - \phi_M$;

$Q_1 = T_1 (\phi_N - \phi_M)$; $Q_2 = T_2 (\phi_N - \phi_M)$, and

$W = (T_1 - T_2) (\phi_N - \phi_M) = Q_1 - Q_2$

$$\text{Efficiency} = \frac{\text{Work Done}}{\text{Heat Received}} = \frac{W}{Q_1} = \frac{Q_1 - Q_2}{Q_1}$$

$$= 1 - \frac{Q_2}{Q_1}.$$

Efficiency can also be expressed as : $\dfrac{\text{MABN} - \text{MDCN}}{\text{MABN}}$

$$= \frac{T_1(\phi_N - \phi_M) - T_2(\phi_N - \phi_M)}{T_1(\phi_N - \phi_M)} = \frac{T_1 - T_2}{T_1}$$

$$= 1 - \frac{T_2}{T_1};$$

so that—

$$1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2}{T_1} \text{ and } \frac{Q_2}{Q_1} = \frac{T_2}{T_1}, \text{ i.e., } \frac{Q_1}{T_1} = \frac{Q_2}{T_2}.$$

That is to say, the ratio of the quantities of heat received and rejected is equal to the ratio of the absolute temperatures at which the respective exchanges take place.

We have all considered this cycle, in its reversed form, seen in Fig. I, 1 (*b*), as the ideal cycle of reference of a refrigerator, the same relationships holding between Q_1, T_1 and Q_2, T_2. In this case, Q_2 is the quantity of heat received at the lowest temperature, T_2, and Q_1 is the quantity of heat rejected at the highest temperature, T_1, of the cycle. $W = Q_1 - Q_2$ is now the quantity of work done *on* the working substance during a cycle.

Refrigeration is defined as the removal of heat from a body at a lower temperature than that of its surroundings. In this ideal case, the quantity of heat refrigerated is, of course, Q_2 heat units per cycle ; W, expressed in heat units, represents the expenditure of mechanical energy to drive the machine through the reversed cycle. The efficacy of the operation of refrigeration is given by—

$$\frac{\text{Heat Refrigerated}}{\text{Work Done}} = \frac{Q_2}{W}.$$

This ratio is called the *Coefficient of Performance.* But, as before, $W = Q_1 - Q_2$ and $\frac{Q_1}{T_1} = \frac{Q_2}{T_2}$, so that $\frac{Q_2}{W}$

$$\frac{Q_2}{Q_1 - Q_2} = \frac{1}{\frac{Q_1}{Q_2} - 1} = \frac{1}{\frac{T_1}{T_2} - 1} = \frac{T_2}{T_1 - T_2}.$$

In the direct cycle, the efficiency is $\frac{T_1 - T_2}{T_1}$, an expression in which the desirability of making $(T_1 - T_2)$ as large as possible is indicated; in the reversed, or refrigerator cycle, the Coefficient of Performance, $\frac{T_2}{T_1 - T_2}$, shows an opposite desirability, namely, of making $(T_1 - T_2)$ as small as possible.

In the direct cycle, with a given temperature range, $(T_1 - T_2)$, the lowest possible value of T_1 is desirable; in the refrigerator cycle, with a given value of $(T_1 - T_2)$, the highest possible value of T_2 is desirable.

There is a second method of considering the reversed cycle, namely, that in which it forms the standard of reference for the heat pump. In this, the processes, and the organs necessary to carry them out, are precisely the same as in the refrigerator cycle; we are, however, not now primarily concerned with the heat, Q_2, taken in at the lowest temperature, T_2, but with the quantity of heat, Q_1, rejected or given up at the highest temperature, T_1, in its relationship to W, the work done in the cycle. In other words, the efficacy of the cycle is now measured by:

$$\frac{\text{Heat Delivered}}{\text{Work Done}} = \frac{Q_1}{W} = \frac{T_1}{T_1 - T_2}.$$

No agreement has been reached on a name for this relationship and the term, Coefficient of Performance, used for refrigeration, has been loosely, and, in the author's view, wrongly applied to it. He proposes "*Performance*

Energy Ratio," and this term will thus be used throughout these lectures.

Since the expression $\dfrac{T_1}{T_1 - T_2}$ is the reciprocal of $\dfrac{T_1 - T_2}{T_1}$, the efficiency of the corresponding heat engine, Mr. J. A. Sumner, of Norwich*, proposes "*Reciprocal Thermal Efficiency*". Dr. Oscar Faber, an authority in this subject, employs the term "*Advantage*"†. The French, it is interesting to note, use "*Coefficient of Amplification*". None of these expressions, however, emphasises that a ratio is in question and Performance Energy Ratio is preferred, since what we measure in actual cases is the ratio of those quantities of energy which determine the performance of a heat pump.

All three of these quantities, Efficiency of a heat engine, Coefficient of Performance of a refrigerator and Performance Energy Ratio of a heat pump, have their highest conceivable values when the operations are carried out in the hypothetical engine proposed by Carnot.

In the reversed cycle, we have adiabatic expansion from T_1 to T_2 along AD and adiabatic compression from T_2 to T_1 along CB. DC and BA are isothermal stages: during DC, heat is received at T_2 from an infinite cold body or source of heat, and, during BA, heat is rejected at T_1 to an infinite hot body or receiver of heat. In the two isothermal stages, the temperatures of the working substance are always assumed to be, respectively, the same as that of the body from which heat is received and of that to which heat is rejected.

The values of the Coefficient of Performance measured in actual refrigerators, or the values of the Performance Energy Ratio in actual heat pumps, will fall below the ideal values given by the Carnot cycle, in so far as the actual processes fall short of those postulated in the

*Proc.I.Mech.E., 1948, Vol. 158, p. 22.

†Proc.I.Mech.E., 1946, Vol. 154, p. 144.

Carnot engine. These deficiencies in actual machines may be derived from (*a*) the properties of the working substance used, (*b*) the cycle adopted, and (*c*) the design conditions. Having selected the working substance, we are often restricted as to the cycle we must adopt. We are technically more free as regards our design conditions, but usually are compelled to make the best practical compromise in accordance with the financial or other external circumstances forced upon us.

It may help at this point to focus our ideas if we encroach a little on the second lecture and consider the conditions in an actual heat pump installation. We will take as an example the installation shown in Fig. I, 2, built by Messrs. Escher Wyss, of Zurich, for a market-gardener in Switzerland.* In the two World Wars, that country, while fortunate in not being involved directly in hostilities, suffered greatly through having no considerable internal fuel supplies. As a result of their experience in the first war, their extensive development in installations producing hydro-electric power—the so-called "white coal"—was brought about; in the second war, their supplies both of hydro-electric power and of imported fuel proved inadequate and, in order to make the best use of these two sources of power, they were compelled to consider very seriously the application of mechanical aids in thermal operations. It is in the U.S.A., however, that the greatest experience in the control of air conditions by means of heat pumps has been gained. That country, with its large area, its regional and seasonal extremes of climate, and its large population who, for the most part, are mechanically-minded, has made great progress in this direction, but it should be remembered that Swiss conditions approach more closely to those obtaining in this country. In both cases, however, their experiences in practice may be studied with profit.

It will be recalled that the organs of a heat pump are precisely the same as those of a refrigerator. Those for a

*Bulletin, Association Suisse des Electriciens, 1946, No. 18, p. 539.

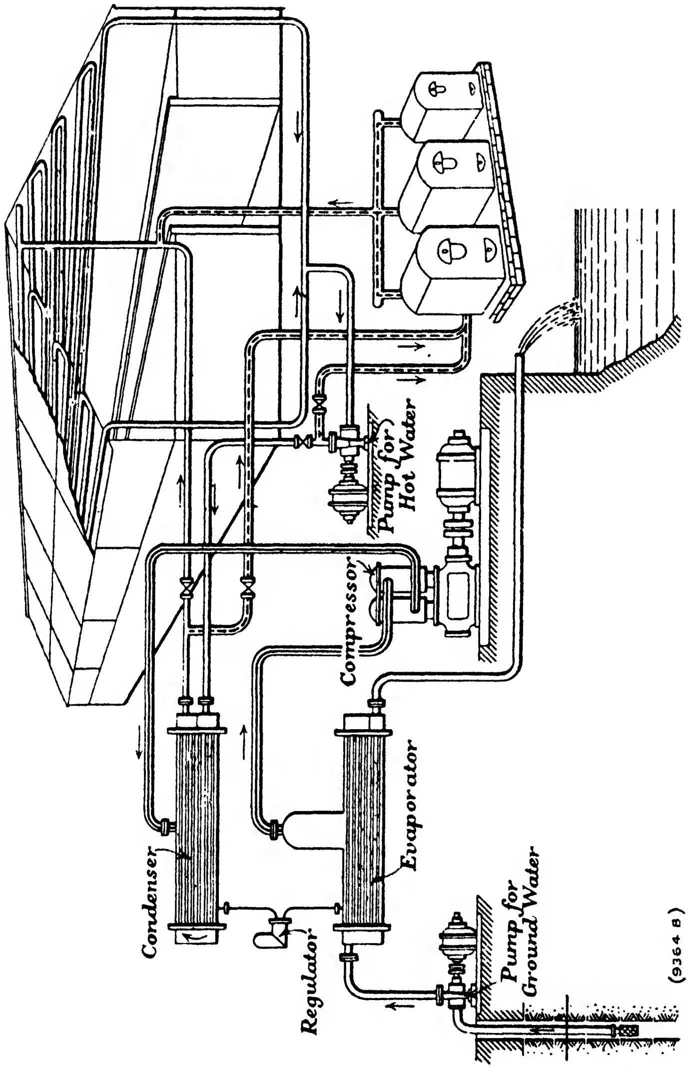

FIG. I—2.

Arrangement of Heat Pump Installation for a Market-Gardener.

heat pump in which a liquid and its vapour are used as the working substance are shown diagrammatically in Fig. I, 3.

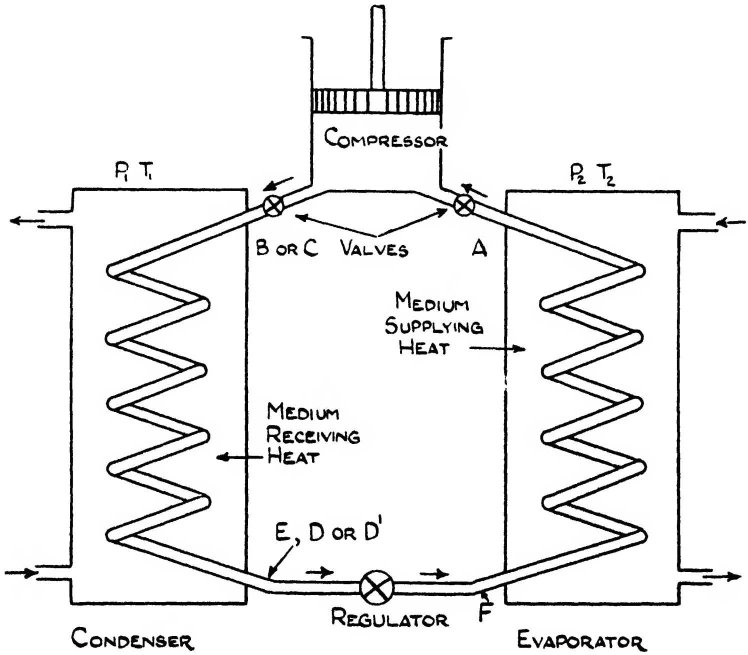

FIG. I—3.
Organs of Heat Pump or of Refrigerator Working on a Vapour-Compression Cycle.

The four stages of the cycle are : (1) heat reception in the evaporator, at the lower pressure, from the medium supplying the low temperature heat ; (2) compression of the working substance by the compressor from the lower to the higher pressure ; (3) delivery of heat by the working substance, at the higher pressure, to the medium receiving the high temperature heat ; (4) expansion from the higher pressure to the lower pressure through the regulator, a throttling operation ideally at constant total heat. (The points, A, B . . . will be referred to later.)

Stages 1 and 3 involve, respectively, the transfer of heat to and from the working substance ; stages 2 and 4 primarily involve changes of pressure only. Pumps to circulate the media supplying or receiving heat may be necessary auxiliaries.

In the installation shown in Fig. I, 2, the source of heat at low temperature is ground water. About 110 gal. (500 lit.) per minute are necessary and this quantity is available at a depth of 23 ft. (7 m.) and at steady temperatures, at all seasons, ranging from 50 to 54°F. (10 to 12°C.). This water is pumped through the evaporator and its temperature is reduced, by the transfer of heat, by about 7°F. (4°C.) ; it then flows to waste. The working substance in the heat pump proper is ammonia, and this is vaporised in the evaporator, at 73·3 lb. per sq. in., abs. (5·15 kg. per sq. cm. abs.) by means of the heat received from the ground water. After evaporation, the pressure of the ammonia vapour is raised in a two-stage compressor to about 299·7 lb. per sq. in., abs. (21 kg. per sq. cm. abs.). In the condenser, heat is transferred from the ammonia to the hot water of the heating system. The ammonia, as a liquid or very wet vapour, is expanded through the regulator and returns to the evaporator. The hot water circuits may be followed on the diagram : hot water is delivered from the condenser at 122°F. (50°C.) and, after giving up its heat in the hot houses, is returned at 104°F. (40°C.) to the condenser. For the power required by the compressor a motor of 35 kW is provided ; power of 1·5 kW is supplied to the motor driving the pump for the ground water, while a small motor suffices for driving the pump for circulating the hot water.

The installation has replaced a direct-heating electric boiler and a set of three coal-fired boilers and, in its first year of service, was subjected to certain conditions by the suppliers of electrical power : at the daily peak load of the supply, from 10.30 a.m. to 12.0 noon, all electricity was cut off and, in order to bridge this interval, the boilers, without firing, were left in circuit to serve as heat accumulators. The control of the heat pump is fully automatic and is achieved through the medium of thermostats mounted on the supply and return pipes of the heating system. In the last year before the installation of the heat pump, 196,240 kW-hrs. of power and a quantity of coal not stated were consumed ; with the heat pump in service, this was reduced to 65,000 kW-hrs., and no coal. The equivalent overall annual Performance Energy Ratio thus exceeds 3/1. This represents to the owner a saving of over 135,000 units and some tons of coal per annum ; against this must be placed the interest on the capital and the costs of maintenance. To the electricity undertaking it represents a saving of fuel of, say, 55 tons per annum, as well as a reduction in the load. Such a reduction in load was not sought by electricity undertakings before the war : quite the contrary. But, today, in practically every country, shortages of fuel, plant and equipment, combined with steadily increasing real and potential demands, continue to force undertakings to seek all possible means of economising in their outputs.

The temperatures at various points, as far as these are available, are given in Figs. I, 4, and I, 5, and are tabulated with the saturation pressures and temperatures in the condenser and evaporator, respectively, in Table I, 1.

TABLE I, 1

Heat pump installation in Fig. I, 2

Ground Water Temperatures :	inlet	50°F. (10°C.) ;	outlet	43°F. (6°C.)
Ammonia Evaporator Temperatures :	,,	40°F. (4·5°C.) ;	,,	$\nless$40°F. (4·5°C.)
Ammonia Compressor Temperatures :	,,	$\nless$40°F. (4·5°C.) ;	,,	$\ngtr$203°F. (95°C.)
Ammonia Condenser Temperatures :	,,	$\ngtr$203°F. (95°C). ;	,,	$\ngtr$123°F. (50·5°C.)
Hot Water Temperatures :	,,	104°F. (40°C.) ;	,,	122°F. (50°C.)
Pressure in Condenser :	299·7 lb. per sq. in. abs. (21·0 kg. per sq. cm. abs.) Saturation temperature 123°F. (50·5°C.).			
Pressure in Evaporator :	73·3 lb. per sq. in. abs. (5·15 kg. per sq. cm. abs.) Saturation temperature, 40°F. (4·5°C.).			

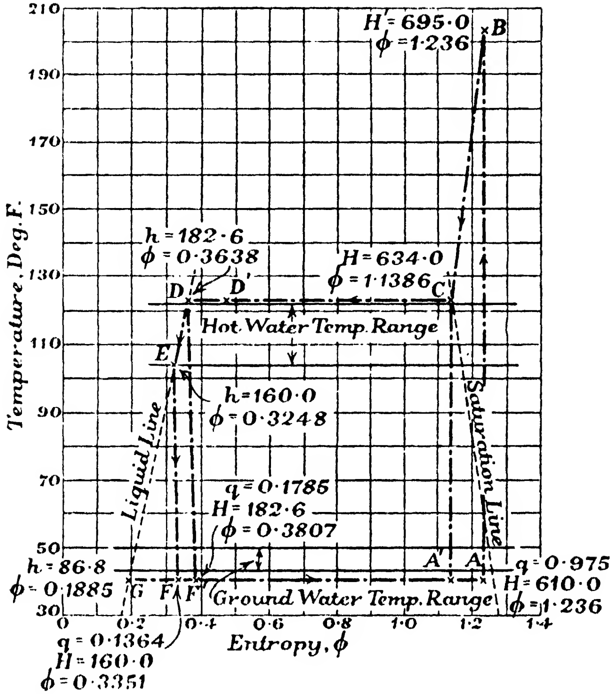

FIG. I—4.
Temperature-Entropy Diagram of Installation of FIG. I—2.

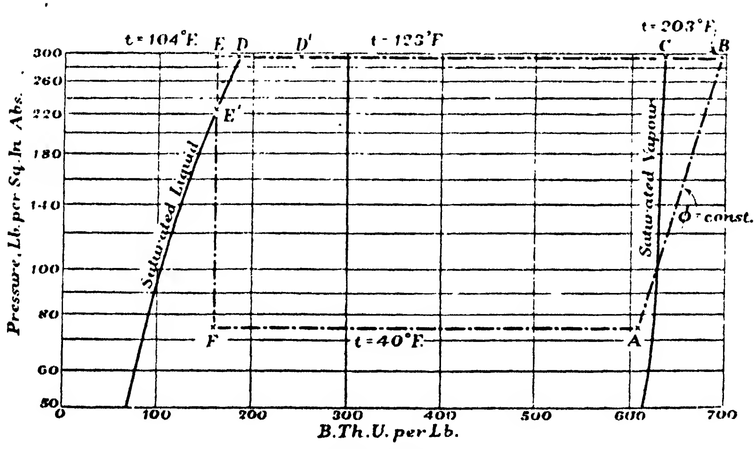

FIG. I—5.
Pressure-Total Heat Diagram of Installation of FIG. I—2.

The values at entry to the evaporator are not given in the article but in view of the lowest temperature of the ground water, 43°F. (6°C.), the saturation temperature may be taken as 40°F. (4·5°C.), corresponding to a pressure of 73·3 lb. per sq. in. abs. (5·15 kg. per sq. cm. abs.). This gives 4·08/1 as the pressure ratio of the compressor. The temperature in the condenser may range from the saturation temperature of the ammonia, 123°F. (50·5°C.) or more, according to the degree of superheat after compression, down at lowest to the outlet temperature, 104°F. (40°C.), of the water from the hot water system.

The ideal temperature-entropy diagram corresponding to this cycle is plotted in Fig. I, 4, on which the values of Total Heat and Entropy for various points of the cycle are also given. These values were taken from the Tables and Chart giving the thermodynamic properties of ammonia issued by the U.S. Bureau of Standards, Circular No. 142. We have used them at King's College for some twelve years and the author would like at this point to pay tribute to the ingenuity and convenience of the chart.

The temperature-entropy diagram is ABCDEFA, the letters giving the states corresponding to the same letters on Fig. I, 3, and the following conditions were taken: 1, compression is assumed to be adiabatic and is shown by AB, the dryness fraction at A being 0·975; 2, the greatest cooling of the liquid in the condenser is assumed, namely, to the lowest temperature in the hot water, 104°F. (40°C.) and is shown by BCDE; 3, all pressure losses due to the circulation of the ammonia have been neglected; 4, physical equilibrium and homogeneity in the working substance are assumed at all points; 5, all heat losses and gains are neglected. Lines showing the ranges of temperature of the hot water and of the ground water, respectively, have been added to the diagram. The same cycle, with the same letters for corresponding points, is plotted in Fig. I, 5, on the pressure-total heat basis employed in the Bureau of Standards Chart. Pressures are plotted vertically and Total Heats horizontally.

The Performance Energy Ratio $= \dfrac{\text{Heat Delivered}}{\text{Work Done}}$. This ratio has been calculated for two cases : in the first, the liquid is cooled to E = 104°F. (40°C.) ; in the second case, the fluid leaves the condenser as liquid at the saturation temperature, 123°F. (50·5°C.), as at D. In the former—

$$\text{P.E.R.} = \frac{H_B - H_E}{H_B - H_A} = \frac{695 - 160}{695 - 610} = \frac{535}{85} = \frac{6{\cdot}29}{1}.$$

In the latter case—

$$\text{P.E.R.} = \frac{H_B - H_D}{H_B - H_A} = \frac{695 - 182{\cdot}8}{695 - 610} = \frac{513{\cdot}2}{85} = \frac{6{\cdot}03}{1}$$

The best possible theoretical value of the Performance Energy Ratio is seen to be 6·29 ; in the absence of cooling of the liquid from 123°F. to 104°F. (50·5°C. to 40°C.), the value is reduced to 6·03. The work of compression, as measured by $H_B - H_A$, is unaffected by the conditions of the working substance on leaving the condenser and on entering the evaporator, so that the denominator remains unchanged. The numerator, however, depends directly on the heat delivered in the condenser and, as a minimum requirement, all the vapour should be condensed : any uncondensed vapour, as at D′, passing to the regulator would take with it a proportionately high quantity of heat and would thus reduce directly the Performance Energy Ratio.

It was pointed out earlier that the stages involving the evaporator and condenser are concerned primarily with the transfer of heat. In the evaporator, the ground water, the medium giving up heat to the working substance, suffers a fall in temperature from 50°F. to 43°F. (10°C. to 6°C.), while the working substance remains at 40°F. (4·5°C.). This differs from the conditions in the ideal Carnot heat pump, in which the source of heat does not suffer a fall in temperature and the temperature of the working substance is assumed to be equal to that of the source of heat. On entering the condenser, the working substance at B is in the form of superheated vapour at a maximum temperature of 203°F. (95°C.). This first gives up its superheat and is cooled to the saturation temperature, 123°F. (50·5°C.) ; it is then condensed at constant temperature and may or may not be cooled as liquid from 123°F. to 104°F. (50·5°C. to 40°C.), or to some intermediate temperature. The medium receiving heat has its temperature raised in the condenser from 104°F. to 122°F. (40°C. to 50°C.). This again differs from the conditions in the Carnot heat pump, in which both the working substance and the receiver of heat would be at the same constant temperature.

The value of the Performance Energy Ratio of the Carnot cycle corresponding to the actual cycle of Figs. I, 4, and I, 5, is—

$$\frac{T_1}{T_1 - T_2} = \frac{123 + 460}{123 - 40} = \frac{583}{83} = 7{\cdot}03.$$

It is of interest to compare this value, 7·03, with those values, 6·29 and 6·03, for the cycle on the temperature-entropy diagram, ABCDEFA, and to obtain some idea of the reasons for the small differences. ABCDEFA does not differ greatly in form from the rectangle of the Carnot cycle : although a considerably higher temperature is shown at B, the area of the part above CD, the temperature of which would be the upper temperature, T_1, of the Carnot cycle, is only small and its influence correspondingly limited. Similarly, the deviation from the Carnot cycle during DEF or DF′ is also small.

The deviation of the form of ABCDEFA from the diagram for the Carnot cycle being small, the temperature differences between the working substance and the heat conveying media in the condenser and evaporator are of vital importance in the influence they exert on the overall range of temperature of the cycle, and thus on the Performance Energy Ratio of a possible Carnot cycle.

The actual mean temperature of the hot water $= \frac{122 + 104}{2} = 113$°F. (45°C.) which, it may be presumed, would be a satisfactory value if the water were circulated very rapidly and thus would not suffer an appreciable fall of temperature. With an infinitely large condenser, this is the saturation value at the upper pressure of the working substance. Similarly, with an infinitely large evaporator, and a large flow of the heat-supplying medium, the mean temperature of the ground-water would approach 50°F. (10°C.). Under these conditions, the corresponding Carnot cycle has a Performance Energy Ratio $= \frac{113 + 460}{113 - 50} = \frac{573}{63} = 9{\cdot}1$. This is much greater than 7·03, the value for the Carnot cycle corresponding to the actual cycle. It would thus be possible, with larger heat exchangers, to reduce the temperature differences between the working substance and the media transferring heat, and in this way to increase the Performance Energy Ratio. But, especially in the case of the condenser, in which the pressure of the working substance is high, increase in size would be both difficult and costly.

Summarising these results, it is seen that the deficiencies of this installation follow, to a small extent only, from the properties of the working substance and from the cycle adopted, but are mainly the result of limitations imposed, in connection with the heat exchange operations, by the practical design conditions.

Considering the three fluid circuits, let W_G, W_A and W_H lb. per minute be the mass flows of the ground water, the ammonia, and the hot water, respectively. W_G may be taken as 100 gal. per min., that is, 1,000 lb. per min. (455 lit. per min.) ; this ground water suffers a fall of temperature of 7°F. (4°C.), and thus gives up 7,000

B.Th.U. (1,770 kcal.) per min. The hot water receives heat in the condenser to raise its temperature from 104°F. to 122°F. (40°C. to 50°C.), giving a rise of 18°F. (10°C.), and a mean temperature in the system of 113°F. (45°C.) ; $W_H = 1000 \cdot \frac{7}{18} \cdot \frac{6}{5} = 466$ lb. (212 kg.) per min., assuming the Performance Energy Ratio to be 6/1. Doubling the rate of circulation, with the system otherwise unchanged, would require four times the power for the hot-water circulating pump and, with the same mean temperature of the hot water, 113°F. (45°C.), would halve its temperature range. Heat transfer in the condenser, however, would be improved and it might be possible to reduce the upper pressure and temperature of the ammonia cycle, thus improving the Performance Energy Ratio. The power thus saved in driving the compressor should more than compensate for the increase of power to the circulating pump.

The mass flow of ammonia, neglecting losses, can be found from the relationship :

$$W_A\,(H_A - H_F) = W_A\,(610 - 160)$$
$$= W_A.\ 450 = 7{,}000 \text{ B.Th.U. per min.}$$

This gives $W_A = \frac{7000}{450} = 15{\cdot}55$ lb. (7·06 kg.) per min. The specific volume at A [ϕ_W being 0 at $-$ 40°F. ($-$ 40°C.)] $= \frac{1{\cdot}236}{1{\cdot}262} \times {\cdot}3{\cdot}971 = 3{\cdot}88$ cu. ft. per lb. The necessary effective displacement of the compressor is thus :

$$15{\cdot}55 \times 3{\cdot}88 = 60{\cdot}4 \text{ cu. ft. (171 lit.) per min.}$$

Certain assumptions in our discussions must be emphasised, namely, that points on the temperature-entropy diagram are taken to represent accurately the states of the working substance and that pressure differences from point to point have been neglected. As with most successful engineering designs, the fullest information must be sought but the final result will be a compromise between various conflicting conditions. These are, then, some of the considerations affecting the design of heat pump installations in which vapour-compression cycles are employed ; others will be dealt with in the next lecture.

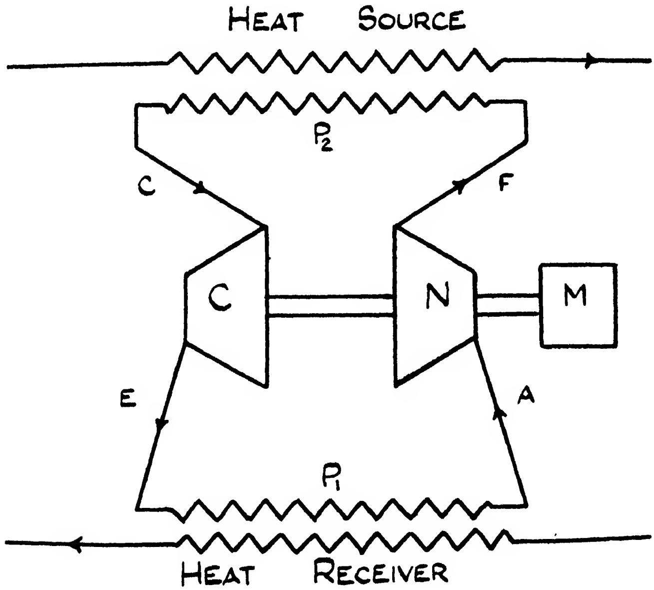

FIG. 1—6.

Organs of Heat Pump Working on a Reversed Joule Cycle.

Coming to heat pumps in which air or other gas is the working substance, it will be recalled that the first refrigerator to be successful in practice was the Bell-Coleman machine, which, using air as the working substance, operated on a reversed Joule or constant pressure cycle. In this case, the fall of pressure must take place in an expansion organ since, with a perfect gas, there is no "step-down" of temperature in a throttling operation. The organs of such a heat pump are as shown in Fig. I, 6, and the cycle is as follows : a compressor, C, takes in the air at the lower pressure, P_2, compresses it to the upper pressure P_1 and delivers it to a heat exchanger in which heat is delivered at constant pressure to a heat receiving medium ; the air, after giving up its heat, passes to the expansion organ, N, in which it expands doing positive work ; from N the air passes, at the lower pressure P_2, to a heat exchanger, in which it receives heat, at constant pressure, from the source of heat at low temperature. The difference between the negative work of the compression of the air in C and the positive work of expansion in N is supplied by a driving unit or motor M.

The pressure-volume and temperature-entropy diagrams for 1 lb. of air, passing through such a cycle under ideal conditions, are shown as CEAF in Fig. I, 7 (*a*) and (*b*). The cycle consists of two adiabatic stages, CE and AF, and two constant pressure stages, EA and FC. The points C, E, A, and F, are given in Fig. I, 6. Incidentally, in its direct form, AECF, this is the ideal cycle of the simple open gas turbine. The difference of entropy between E and A and between C and F $= C_p \log_e \frac{T_E}{T_A} = C_p \log_e \frac{T_C}{T_F}$. Assuming a constant value for C_p, the specific heat at constant pressure, a reasonable assumption when the temperature ratios are limited, as they are in this case, the Performance Energy Ratio of the ideal Joule cycle is given by :

$$\frac{\text{Heat Delivered}}{\text{Work Done}} = \frac{C_p (T_E - T_A)}{C_p(T_E - T_C) - C_p(T_A - T_F)} =$$

$$\frac{T_E - T_A}{(T_E - T_C) - (T_A - T_F)} = \frac{T_E - T_A}{(T_E - T_A) - (T_C - T_F)}$$

$$= \frac{1}{1 - \frac{T_C - T_F}{T_E - T_A}} = \frac{1}{1 - \frac{T_C}{T_E}} = \frac{1}{1 - \frac{T_F}{T_A}},$$

since $$\frac{T_F}{T_A} = \frac{T_C}{T_E} = \frac{T_C - T_F}{T_E - T_A}.$$

Also, $$\frac{T_A}{T_F} = \frac{T_E}{T_C} = (R)^{\frac{\gamma - 1}{\gamma}}.$$

in which γ is the ratio of the specific heat at constant pressure to that at constant volume, and R is the ratio of the upper pressure, P_1, to the lower pressure, P_2. This ratio, R, will be referred to as the Pressure Ratio

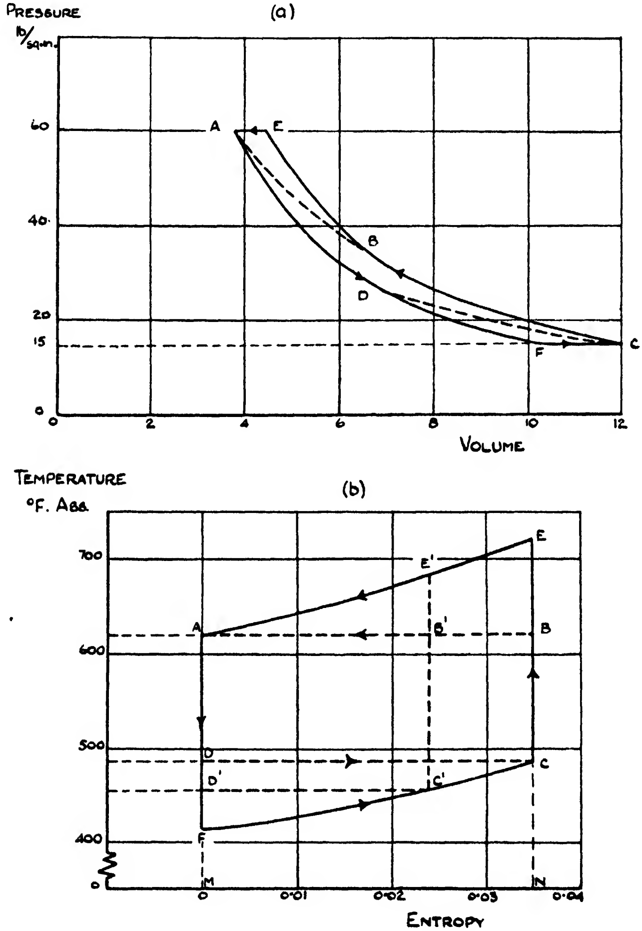

FIG. I—7.

Pressure-Volume and Temperature-Entropy Diagrams for Ideal Joule Cycle (Reversed).

of the cycle. So that the Performance Energy Ratio is also given by :

$$\frac{1}{1 - \left(\frac{1}{R}\right)^{\frac{\gamma - 1}{\gamma}}}.$$

Taking $\gamma = 1{\cdot}4$, calculations give, for a range of values of the Pressure Ratio, the following relationships between the Pressure Ratio and the Performance Energy Ratio :

Pressure Ratio	4	3	2	1·5
Performance Energy Ratio	3·06	3·71	5·56	9·20

These results show that, on the ideal cycle, the Performance Energy Ratio increases as R is reduced and that high values of the Performance Energy Ratio may be obtained in cases in which the Pressure Ratio can be kept at a low value. Since, as we have seen, the Pressure Ratio depends on the range of temperature of the cycle that it is necessary to cover, it is this range that should be kept as low as possible.

It is of interest to consider the relationships of the ideal cycle to the corresponding Carnot cycle. The pressure-volume and temperature-entropy diagrams of Fig. I, 7, are drawn to scale for a particular cycle CEAF, in which R = 4. The corresponding Carnot cycle, namely, that with the same overall pressure ratio, R, is CBAD. Data for these points are given in Table I, 2.

TABLE I, 2

Data for points on Fig. I, 7 and I, 8

Points	A	B	C	D	E	F	G	H
Pressure, lb. per sq. in. absolute	60	35.5	15	25.4	60	15	60	15
Temperature, °F. abs.	620	620	485	485	720	417	746	437
Entropy	0	0·0356	0·0356	0	0·0356	0	0·0441	0·0115

Divergencies are at once seen to follow the different conditions under which heat is received by the working substance during FC in the Joule cycle and during DC in the Carnot cycle and, similarly, rejected during EA and BA in the respective cycles. Although the heat delivered on the Joule cycle, NEAM, is slightly greater than that delivered on the Carnot cycle, NBAM, it is seen that the necessary work done on the former, CEAF, is much greater than CBAD on the latter. This leads to a considerably lower value of the Performance Energy Ratio for the ideal Joule cycle : 3·06 for the latter against 4·59, which is the value for the Carnot cycle in this case.

The Performance Energy Ratio of the ideal Joule cycle for a given value of R is independent of the ranges of temperature of the constant pressure stages ; the value for the corresponding Carnot cycle will, however, vary with these ranges. For example, if the temperature ranges be reduced in Fig. I, 7 (*b*) so that the Joule cycle becomes C′E′AF, the corresponding Carnot cycle is C′B′AD′, which clearly has a lower Performance Energy Ratio than CBAD. In the limit, when the temperature ranges during the constant pressure stages of the Joule cycle are infinitely small, the values of this ratio for the two cycles become equal, a fact that is, of course, merely of academic interest.

A matter of great practical importance is that of the efficiencies with which compression and expansion may be carried out. If the components for compression and expansion are of the rotary type, as is desirable, for reasons of space and cost, if large volumes of air come into consideration, the efficiencies of compression and expansion, relative to adiabatic conditions, will be lower than for piston machines and it is desirable to study the effects of these efficiencies. Fig. I, 8, shows the ideal cycle, CEAF, of Fig. I, 7 (*b*), together with the modifications introduced when these efficiencies are less than those corresponding to isentropic conditions. If η_C is the adiabatic efficiency of compression, the final temperature,

T_G, at the end of compression is given by the relationship :

$$\eta_C = \frac{T_E - T_C}{T'_G - T_C}.$$

The compression line is CG on Fig. I, 8. Similarly, if η_t is the adiabatic efficiency of expansion, the expansion line is AH and the final temperature, T_H, is given by the relationship : $\eta_t = \frac{T_A - T_H}{T_A - T_F}$. Data for points G and H are also given in Table I, 2.

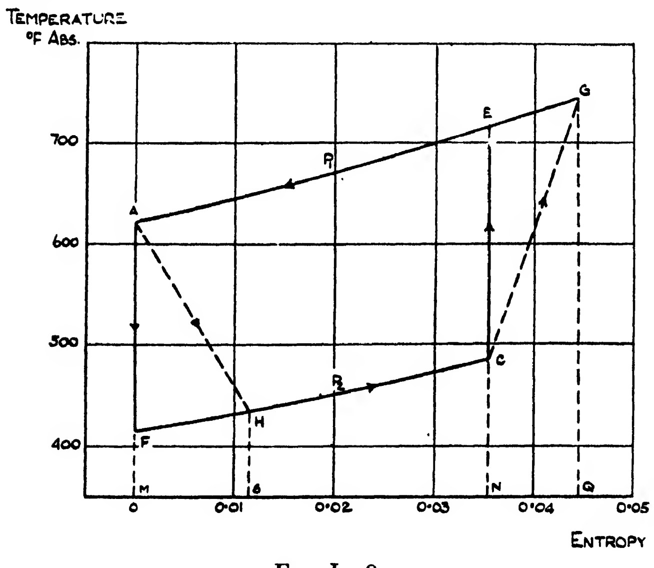

FIG. I—8.

Temperature-Entropy Diagrams for Ideal Joule Cycle and for Cycle in which $\eta_C = \eta_t = 0{\cdot}9$.

The heat delivered, QGAM $= C_p\,(T_G - T_A)$, a greater quantity than NEAM. The net work done is $C_p\,(T_G - T_C) - C_p\,(T_A - T_H)$, a quantity that is proportionately considerably greater than :

$$C_p\,(T_E - T_C) - C_p\,(T_A - T_F).$$

The Performance Energy Ratio is now :

$$\frac{C_p\,(T_G - T_A)}{C_p\,(T_G - T_C) - C_p\,(T_A - T_H)} = \frac{T_G - T_A + \dfrac{1}{\eta_C}(T_E - T_C)}{\dfrac{1}{\eta_C}(T_E - T_C) - \eta_t\,(T_A - T_F)}.$$

The modified Joule cycle, CGAH, in Fig. I, 8, is based on the assumption that both η_C and η_t have the value 0·9. For this case, the Performance Energy Ratio is 1·59, little more than half of that for the ideal cycle. The following figures show the values of the Performance Energy Ratio resulting from calculations in which, with R = 4/1, a series of equal values for the component efficiencies of compression and expansion is assumed :

$\eta_C = \eta_t$	1·0	0·95	0·9	0·85	0·8	0·75
P.E.R.	3·06	2·04	1·59	1·35	1·20	1·10

The results emphasise the importance of the component efficiencies being as high as possible.

While there remains a great deal yet to be discussed, it is clear that, whether the Joule or constant pressure cycle is carried out, therefore, with ideal or with real efficiencies of compression and expansion, as regards its Performance Energy Ratio, it falls considerably short of the Carnot cycle and, working between the same temperature ranges, is inferior to the cycle involving vapour-compression, even before the other practical conditions, such as those affecting heat transfer, are taken into account. In heating and ventilating and in some industrial processes, however, by using the air or gas to be heated as the actual working substance, heat transfer operations may be facilitated.

Whether the working substance of a heat pump is a liquid and its vapour, or a gas, two basic conditions must be fulfilled for satisfactory performance : firstly, that a considerable source of heat is available at the lower value of the temperature range in question ; secondly, that the temperature range through which it is desired to operate is small, the smaller the better in every way.

The last subject to be considered in this lecture is that of thermal compressors. In a large number of industrial operations involving drying, concentration by evaporation and quenching with water, very large quantities of heat are lost to the atmosphere in the form of the latent heat in the steam or water vapour which is often driven off in these processes. In many of these cases, the heat supplied to bring about the evaporation is conveyed in live steam, which is led through suitable tubes or coils in the vats, pans or other vessels containing the substance from which water is to be removed by evaporation, and the resulting water vapour is left to escape to the atmosphere. After this live steam has given up its latent heat it may be returned to the hot-well of the boiler or may pass to waste as condensate : in either case, the latent heat of the steam has been given up and even if the condensate is saved, its sensible heat is but a small proportion of the total heat energy given originally to the water in the boiler. Thermal compressors, which use but a small fraction of the energy represented by the latent heat, can, under suitable conditions, bring about notable economies in such installations.

As a simple example of the method of operation of a thermal compressor, a plant for distilling water will be considered. In Fig. I, 9, live steam supplied from a boiler is used, in the ordinary way, as the heat-conveying medium. The vapour from the water being distilled in the evaporator passes to a condenser where it gives up to circulating water the latent heat, and most of the sensible heat, received from the live steam. In Fig. I, 10, with the thermal compressor installation in operation, the vapour driven off in the still, passes to a compressor in which its pressure and thus its temperature are raised. This steam is then led through the heating coil in the still and gives up its heat to evaporate further water. The only energy required is that to drive the compressor, and the condensing plant is rendered unnecessary, the still in this case becoming a combined evaporator and condenser.

Assume fresh water at 60°F. (15·5°C.) to be supplied in both cases and distilled water at 80°F. (26·7°C.) to be produced. In Fig. I, 9, the heat necessary to raise the supply water from 60°F. (15·5°C.) and to evaporate it at

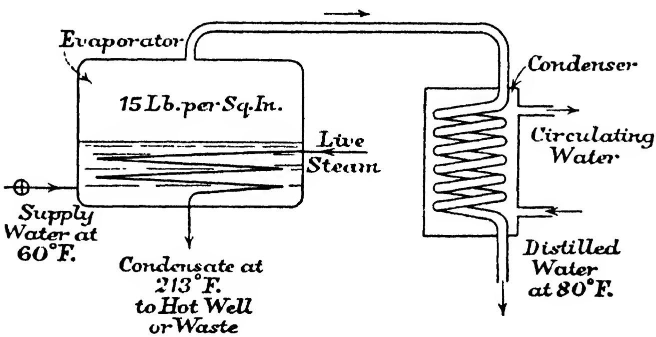

FIG. I—9.

Diagram of Distillation Plant.

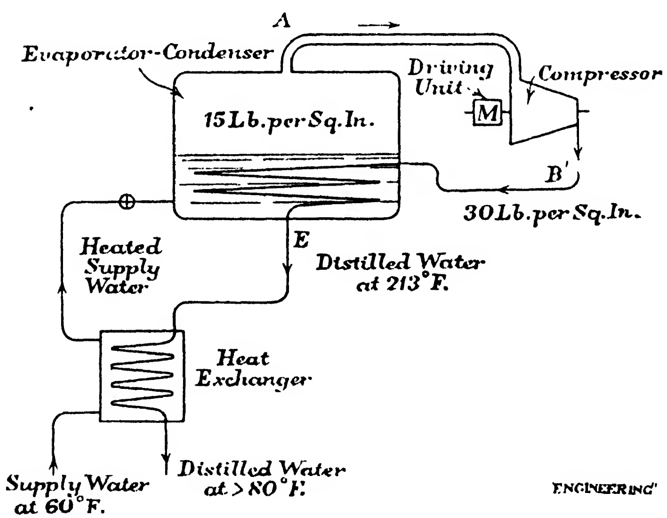

FIG. I—10.

Diagram of Thermal Compressor Plant.

atmospheric pressure must be supplied by live steam. Apart from the dimensions of the heat exchanger, it is immaterial at what higher pressure this is supplied. In Fig. I, 10, the pressure of the heating steam is taken as 30 lb. per sq. in. abs. and this may, for purposes of comparison, be taken as the supply pressure in Fig. I, 9. In the latter case, each lb. of evaporated water has a final total heat of 181·2 + 970·0 = 1151·2 B.Th.U. It thus receives 1151·2 − (60 − 32) = 1123·2 B.Th.U. per lb. from the steam at the upper pressure.

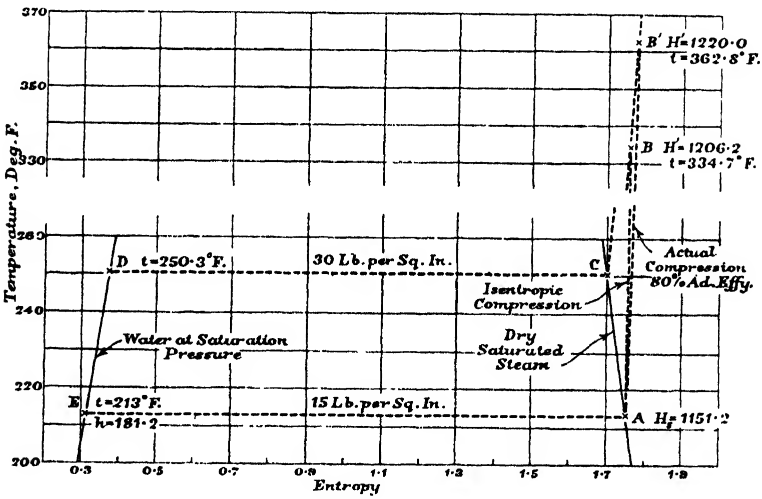

FIG. I—11.

Temperature-Entropy Diagram for Plant of FIG. I—10.

Let the overall efficiency of driving unit and compressor be 72 per cent, of which 80 per cent is the isentropic efficiency of compression and 90 per cent that of the driving unit, in this case an electric motor. The condition of the evaporated vapour, assumed to be dry saturated, is shown by point A on the temperature-entropy diagram of Fig. I, 11. Adiabatic compression is shown by the isentropic AB and gives an increase of total heat of 55 B.Th.U. per lb. Compression with 80 per cent isentropic efficiency is shown by AB′, with an increase of total heat of $55 \cdot \frac{100}{80} = 68{\cdot}8$ B.Th.U. per lb., so that the superheated steam at B′ has a total heat of 1220·0 B.Th.U. per lb. This, when passed to the still, in which it gives up heat to evaporate the water, follows the line BCDE and leaves at 213°F., (100·5°C.), with sensible heat 181·2 B.Th.U. per lb., having given up 1220·0 − 181·2 = 1038·8 B.Th.U. per lb. The fresh water has taken 970 B.Th.U. per lb. of this for its latent heat, leaving 1038·8 − 970 = 68·8 B.Th.U. per lb. available for sensible heat. The fresh water requires 181·2 − (60 − 32) = 153·2 B.Th.U. per lb. as sensible heat and 68·8 of this can be given in the distiller, leaving 153·2 − 68·8 = 84·4 B.Th.U. per lb. to be supplied. The distilled water is at 213°F. and thus has 181·2 − (80 − 32) = 133·2 B.Th.U. per lb. to dispose of in the exchanger, giving the ample reserve of 133·2 − 84·4 = 48·8 B.Th.U. per lb. against loss.

The energy given to the compressor motor is equivalent to $68{\cdot}8 \cdot \frac{100}{90} = 76{\cdot}4$ B.Th.U. per lb. of distilled water. This is only $\frac{76{\cdot}4}{1123{\cdot}2} = 1/14{\cdot}7$ of the energy supplied in Fig. I, 9, in the live steam. This solution ignores heat losses by radiation in both cases. In an actual practical case, however, in which 278 gal. per hr. of distilled water were produced, 79 B.Th.U. per lb. was the equivalent energy supplied at the coupling, giving, on the basis of Fig. I, 9, and I, 10, an input of 64.4 kW at the coupling and an electrical input of about 72 kW, and an overall ratio of 1/12·8.

In concluding this lecture, I would point out that the cases considered have all been taken in simple terms. Various combinations of each and all of these arrangements may, of course, be made to suit individual installations. Further, I have not attempted at this stage to discuss the financial circumstances or the obvious importance of such matters as load factor.

LECTURE II

In the first lecture the fundamental principles of heat pumps were studied and, as an illustration of those principles, an application of an electrically-driven heat pump to space-heating was described. In this lecture, other actual heat pump installations will be considered and certain further conclusions will be drawn concerning their conditions of operation. One installation of outstanding importance is that designed and built by Mr. J. A. Sumner,* the City Electrical Engineer of Norwich, to provide space-heating for the Stores and Workshop Building of the Norwich Electricity Department. This is of interest not only because it is the first large-scale heat pump installation† in this country but also because the circumstances under which its application was carried out were such that it was possible for Mr. Sumner to collect reliable data concerning its operation. These data are of great value in relation to the further development of space-heating by heat pumps under the climatic and other conditions that are to be met in this country ; the evidence of the results of what will be called the "Norwich experiment" is direct and does not involve translation from foreign conditions to our own. As will be seen, this installation was built in the face of great difficulties and the author would offer his tribute to Mr. Sumner for the engineering skill and courage which he displayed in overcoming these difficulties.

The Stores and Workshop Building is shown in Fig. II, 1.

*loc. cit.

†A very early, but small, heat pump installation was applied to his house by Mr. T. G. N. Haldane : *Journal I.E.E.*, Vol. 68, No. 402, 1930, p. 666.

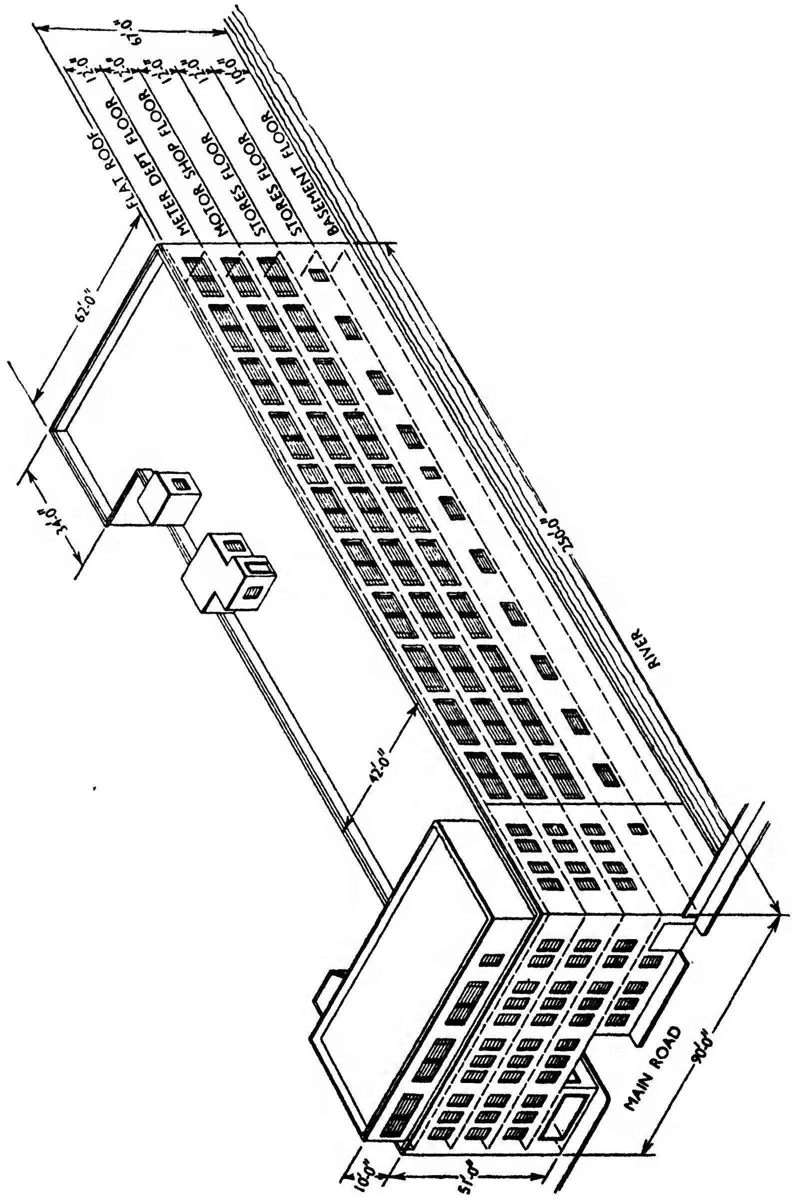

FIG. II—1.
Stores and Workshop Building of the Norwich Electricity Department.

This building adjoins the River Wensum, from which the necessary low temperature heat is derived. The minimum rate of flow of this river is 30,000,000 gal. (136,000 cu. m.) per day. The building is of steel and reinforced concrete frame construction, brick filled ; its volume is about 500,000 cu. ft. (14,160 cu. m.) and its total wall and roof area is 42,500 sq. ft. (3,940 sq. m.), of which the large area of 9,200 sq. ft. (853 sq. m.) is occupied by windows. In addition to storage space and workshops, the building includes certain offices.

In 1940, when the building was nearing completion, Mr. Sumner attempted, but without success, to obtain a heat pump installation to be applied to the central heating plant that had already been installed. This comprised a hot water circulating system with wall radiators, embedded panel heaters and electric fan unit heaters. It was therefore decided at that time to install coal-fired boilers, as a temporary measure, and, using this coal-fired system as a basis, to provide equipment that would render possible a careful study of the heat requirements of the building. This proved to be a particularly valuable preparation for the later application of a heat pump, since, in due course, the measuring equipment facilitated the collection of reliable data upon which an accurate comparison was made between a heating system embodying a heat pump as the source of heat with one in which coal-fired boilers were used.

The heating system as originally fitted was designed for hot water flow temperatures of 180°F. (82°C.) from the boilers to 160°F. (71°C.) on return to the boilers. Experiments were made in the winter of 1940–1 to find the minimum flow temperatures necessary to maintain, on the coldest days, the offices at a temperature of 62·5°F. (17°C.), and the workshops at 60°F. (15·5°C.). It was observed that, without the unit fan heaters, a temperature of 130°F. (54·5°C.) was necessary during the short periods of very low ambient temperatures, but that a maximum temperature of 120°F. (49°C.) was sufficient during the greater part of the heating period. The unit fan heaters, which require unduly high flow temperatures, were therefore replaced by radiators. From 1940–5 the boilers remained in exclusive use.

The diagram showing the organs of the installation, together with the temperatures at various points in the circuits followed, respectively, by the river water and by the hot water, is given in Fig. II, 2. The designer of the

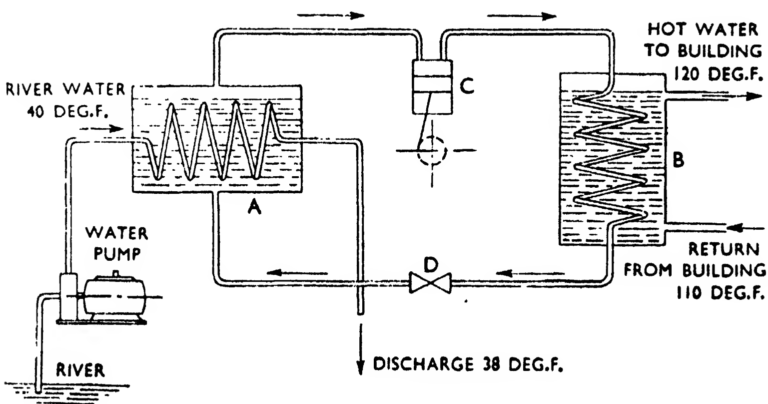

FIG. II—2.
Diagram of Circuits of Norwich Heat Pump.

plant was, however, seriously handicapped, as regards the various features of the design, by the difficulty of obtaining the desired materials and fittings : as events proved, until it was found possible in 1945 to procure a suitable second-hand compressor, no practical steps could be taken towards the realisation of the plant. It was also found that the condenser and the evaporator could not be ordered from specialist suppliers and both were built up on the site from available materials. Only the automatic float valve and the safety valves for the heat pump were obtained from manufacturers of refrigeration equipment. The compressor used at first was a second-hand Brotherhood single-acting piston machine of 11 in. (259 mm.)

bore, 9 in. (229 mm.) stroke with a maximum speed of 350 r.p.m., which had been designed for use with ammonia. It was driven by an electric motor. A second compressor, obtained later directly from Messrs. Brotherhood, was a single-crank two-stage annulus machine, of 16 in. (406 mm.) bore for the first stage and an annulus of 16–13½ in. (406–343 mm.) for the second stage, a stroke of 9 in. (229 mm.) and a speed of 300 r.p.m. This was likewise electrically driven.

Before considering the details of the plant we will first look for a moment at the general arrangement, shown in Fig. II, 3. The design of the condenser and evaporator and the conditions of working will then be considered, after which it will be desirable to return again to the general arrangement. The condenser, with its circulating pipes for the hot water system, may be seen at the bottom of the plan and on the right in the sectional view : in this, the working substance of the heat pump flows inside the tubes and the hot water flows outside the tubes. Pipes from the condenser lead the working substance, now condensed to liquid, through suitable throttling valves to the evaporator. In the evaporator, the river water flows inside the tubes and the working substance occupies the space between the tubes and the shell. The shell plates are pierced at intervals and the working substance, after

---

Reference to Fig. II—3 (opposite)

A Accumulator.
B ½-inch charge valve for charging the system with sulphur dioxide.
C 2-inch liquid pipe.
D Evaporator.
E Compressor.
F Condenser.
G Spray pipes.
H 3-inch suction pipe.
I Pump.
J Motor.
K Heat and water meter.
L Electrode boiler.
M 4-inch hot-water pipe.
N 3-inch supply pipe.
O High-pressure safety valve.
P Storage tank.
Q Electric control gear.
R Belt guard.
S High-pressure float valve.
T 4-inch bore water outlet pipe to river.
U 4-inch bore pipe from river to pump.
V 4-inch bore inlet pipe from pump.
W 4-inch suction pipe.

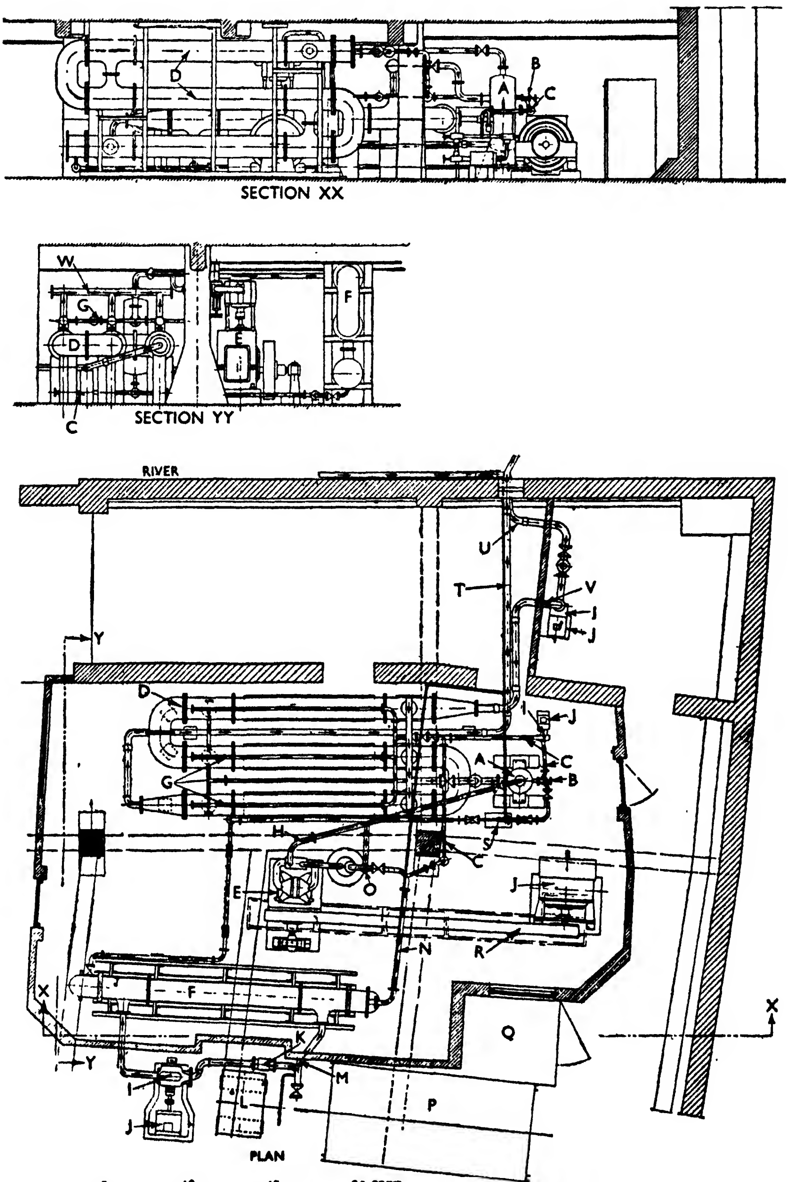

FIG. II—3.
General Arrangement of Norwich Heat Pump.

the fall of pressure in the throttling valves, and in the form of a liquid or very wet vapour, is sprayed on to the outsides of the tubes. After vaporisation, the working substance is drawn from the evaporator into the compressor, compressed and delivered to the condenser.

The working substance used was sulphur dioxide, a choice that was forced upon the designer largely by the circumstances of the mechanical design and by the fact that this substance was obtainable whereas other more desirable substances were not available. To help to understand what was involved in this choice, Table II, 1,

TABLE II, 1*

Properties of working Substances at 40°F. and 120°F.

Substance	P_1 lb. per sq. in.	P_2 lb. per sq. in.	P_1/P_2	V_{S2} cu. ft. per lb.	L_2 B.Th.U per lb.	L_2/V_{S2}
Ammonia, NH_3	286·4	73·3	3·91	3·971	536·2	135·2
Sulphur Dioxide, SO_2	120·0	27·3	4·40	2·950	159·4	54·1
Methyl Chloride, CH_3Cl.	154·2	43·2	3·57	2·286	172·0	75·2
Freon 11, $C\ Cl_3F$	33·2	7·0	4·72	5·447	81·2	14·9
Freon 12, $C\ Cl_2F_2$	171·8	51·7	3·32	0·792	65·7	83·0
Water, H_2O	1·7	0·122	13·98	2·444	107·3	0·44

*Data from Venemann : *Refrigeration Theory and Applications*, Nickerson and Collins Co.

in which are set out certain important properties of typical working substances suitable for use in heat pumps and refrigerators, will be considered. The properties are given for the two temperatures, 40°F. (4·4°C.) and 120°F. (49°C.), values which correspond, respectively, to typical saturation temperatures at which evaporation may take place, in the evaporator, and condensation may take place, in the condenser, in heat pumps used for space-heating.

The properties of the working substance that are of special importance when piston compressors are used include : the order of the necessary saturation pressures P_1 and P_2 ; the pressure ratio P_1/P_2 ; the latent heat per cu. ft. of the vapour at the lower pressure, L_2/V_{s2}. As regards the order of the pressures, the loads on the valves, pistons and the running gear increase with the pressure, as well as the tendency to valve and piston leakage. On the other hand, pressures below atmospheric are not desirable, since any leakage between the system and the atmosphere would cause air to be introduced into the system and this air, by exerting its partial pressure, would lead to increase of the pressures and, by changing adversely the pressure-volume relationships of the compression, lead to inefficiency. Further, the air may contain moisture which may cause corrosion, may freeze up in the passages, may produce explosive mixtures, or may adversely affect the lubrication of the compressor. In considering the pressure ratio, the lower the ratio, the lower are the tendency to piston leakage and the reduction in the effective suction volume of the compressor by the re-expansion of the working substance left after delivery in the clearance volume. In both piston and rotary compressors, the lower the pressure ratio, the higher the adiabatic efficiency. Considering the latent heat per cu. ft. of vapour at the lower pressure, the higher the value, in piston compressors, the smaller is the necessary effective capacity, and thus the size, cost and mechanical losses of the machine ; in rotary compressors, on the other hand, a low value is desirable, since, the higher the specific volume, the lower the blade leakage losses. Incidentally, as regards the theoretical performance ratio, the differences between all these working substances are small, so that this quantity does not enter into the choice of working substance.

Of the substances in Table II, 1, Freon 12 appears to have the best all-round properties, with the sole drawback that the order of its saturation pressures is high ;

the pressure ratio is the lowest and the necessary effective capacity is the lowest except for ammonia, a substance of which the saturation pressures are seen to be excessively high. Freon 12 has other good qualities: it is non-inflammable and odourless, is low in toxicity and is non-corrosive. Sulphur dioxide is markedly inferior to Freon 12 in every respect, except as regards its saturation pressures and the fact that it, too, is non-inflammable.

Considering the data of Table II, 1, in relation to the Norwich design, it will be recalled that this design was, perforce, based on the only available compressor. This machine had been designed for use with ammonia, but not as a component of a heat pump installation. Ammonia it will be noticed in the table, has the highest value for the latent heat per cu. ft. of vapour at the lower pressure. Unfortunately, the values of the saturation pressures at the given temperatures are high and at the upper temperature, 120°F. (49°C.), a saturation pressure of 286·4 lb. per sq. in. (20·1 kg. per sq. cm.) is reached. This pressure was much too high in relation to the permissible bearing loads, for which a maximum piston pressure of 200 lb. per sq. in. (14 kg. per sq. cm.) was the limiting value. This pressure would correspond to a saturation temperature of 96·3°F. (35·7°C.) for ammonia, which, if the compressor formed part of an ordinary refrigeration installation, would be a reasonable value. Ammonia as a working substance for this heat pump was clearly ruled out. Of the other substances in Table II, 1, sulphur dioxide was the only one obtainable at the time and the choice of this substance was thus dictated by these circumstances.

With regard to the properties of sulphur dioxide, it will be noted that, apart from Freon 11 and steam, it has the lowest value for the latent heat per cu. ft. at the lower pressure. Steam comes into question only in large installations and with rotary compressors. It is seen that, of the substances in the table, only Freon 11 has a higher value for pressure ratio. The high value of pressure ratio for sulphur dioxide is an important drawback, since the compressor was a single-stage machine, and this factor leads to an increased tendency to leakage. On the other hand, sulphur dioxide, unlike ammonia, does not attack copper alloys and it was thus possible to employ the standard brass condenser tubes of steam practice in the installation in both the evaporator and the condenser, which was a great advantage under the circumstances of the construction.

The shell of the condenser (see Fig. II, 4), was made of standard flanged cast iron water pipes, the necessary connections being made with suitable T-pieces. Three shells are mounted one above the other, and each consists of two T-pieces with straight lengths of pipe between them. Tube plates are provided, at the extreme ends, the condenser tubes being jointed to them, in the normal way, with screwed ferrules and packing forming glands. The safe internal pressure of the pipes was only 100 lb. per sq. in. (7 kg. per sq. cm.) and the brass tubes, which were capable of withstanding higher pressures, carried the sulphur dioxide, with the hot water, at approximately atmospheric pressure, outside the tubes ; the gas circuit at the ends of the tubes was completed by built-up boxes and heads.

The evaporator shown in Fig. II, 4, was of similar construction to the condenser, but the working substance, since this is now at the low pressure part of the cycle, passed on the outsides of the tubes and the river water circulated through the tubes. The pipes are placed horizontally side by side. The liquid from the condenser, after expansion through the regulator from the higher to the lower pressure, was supplied to the evaporator through spray pipes and nozzles and entered as a finely atomised spray. Details of these sprays are shown in Fig. II, 5. This arrangement was expected to facilitate the early vaporisation of the liquid and to reduce the necessary quantity of working substance in the plant. The vapour, on leaving the evaporator, passed to an accumulator, which may be seen at A in the general arrangement,

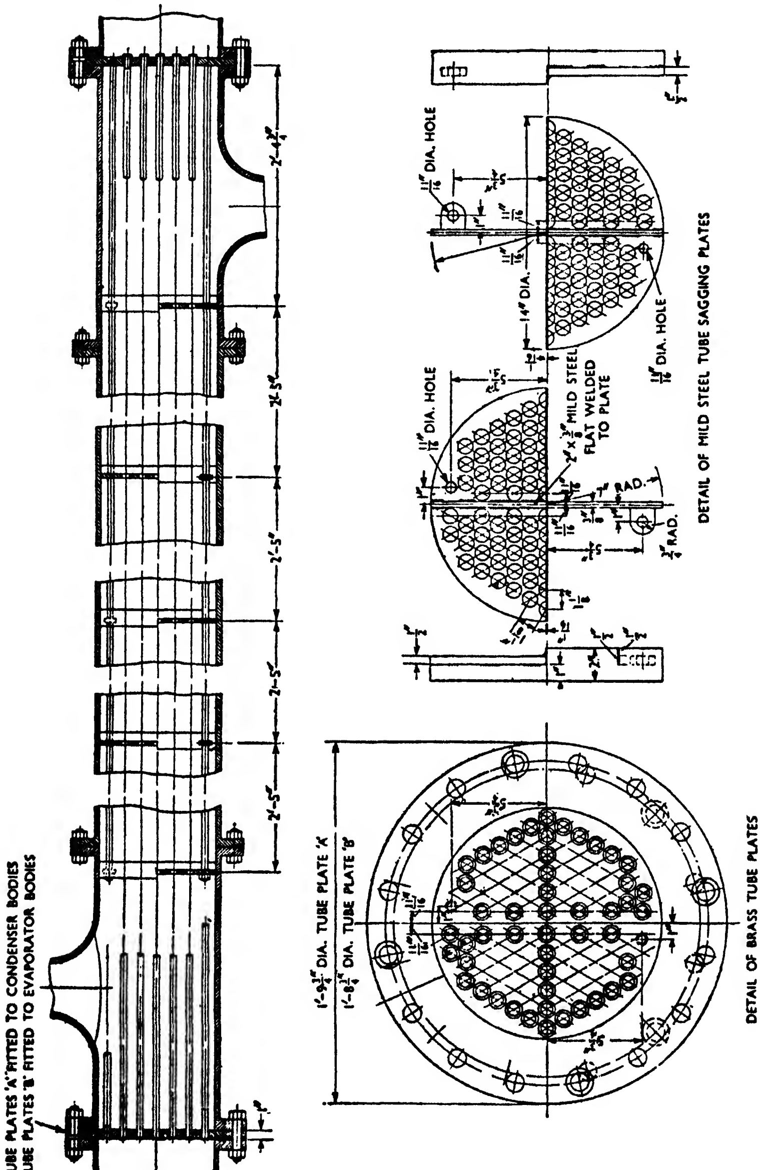

FIG. II—4.
Details of Condensers and Evaporators of Norwich Heat Pump.

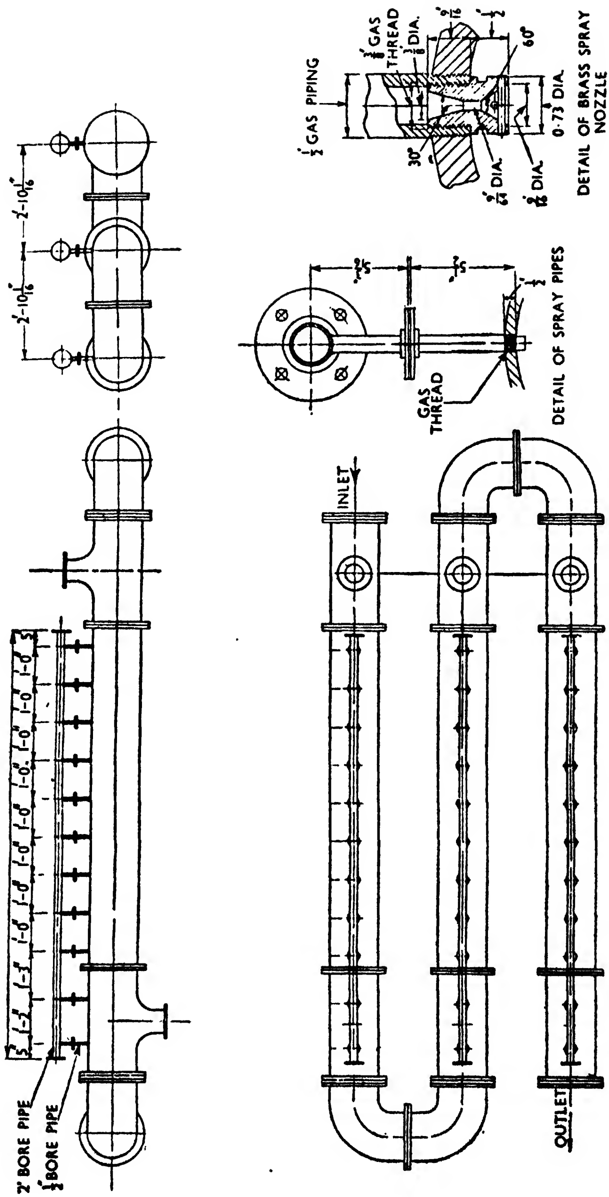

FIG. II—5.
Details of Sprays for Evaporator of Norwich Heat Pump.

Fig. II, 3, whence it was drawn to the compressor. Suitable control and safety devices complete the installation.

In the heating period of 1945–6, the mean value of the performance energy ratio, measured by the ratio of the heat delivered to the electrical energy consumed, was 3·45/1. In the opinion of the author, the results of the Norwich experiment fully justify the claim of Mr. Sumner that heating by means of heat pumps is a practicable method in this country in cases in which upwards of 8 therms or 800,000 B.Th.U. (202,000 kcal.) per hr. are required and a suitable source of heat at low temperature is available.

The two examples of heat pump installations so far considered have been simple and straightforward cases involving their application to space-heating. We will next consider the slightly more complex case shown in diagrammatic form in Fig. II, 6. This is the installation designed and constructed by Messrs. Escher Wyss for the Zurich City Hall. In the diagram, the pump, 2, pumps the water from the River Limmat through the evaporator, 1 ; after giving up heat, this water is returned to the river. The heat pump, 3, draws away the vapour from the evaporator, compresses it and delivers it to the condenser, 4. The heat is absorbed in the heating system, in which the hot water is circulated by the pump, 8, through the circulating pipes, 5, to the radiators, 6, and also to an air pre-heater, 7, and back to the condenser. The working substance, after condensation, passes through the regulator, 9, and from there to the evaporator.

The dimensions of the heat pump are so arranged that it provides the heating for the building to meet all normal demands, but to provide against extreme conditions, an electrically-heated hot water accumulator is installed which is automatically switched on when needed. This is in service for only a small part of the annual heating period, but the relative cost of this, namely, for interest on capital and electrical power, is less than those which it would be necessary to expend if the heat pump part of the installation were made large enough to cope alone with conditions of exceptional cold.

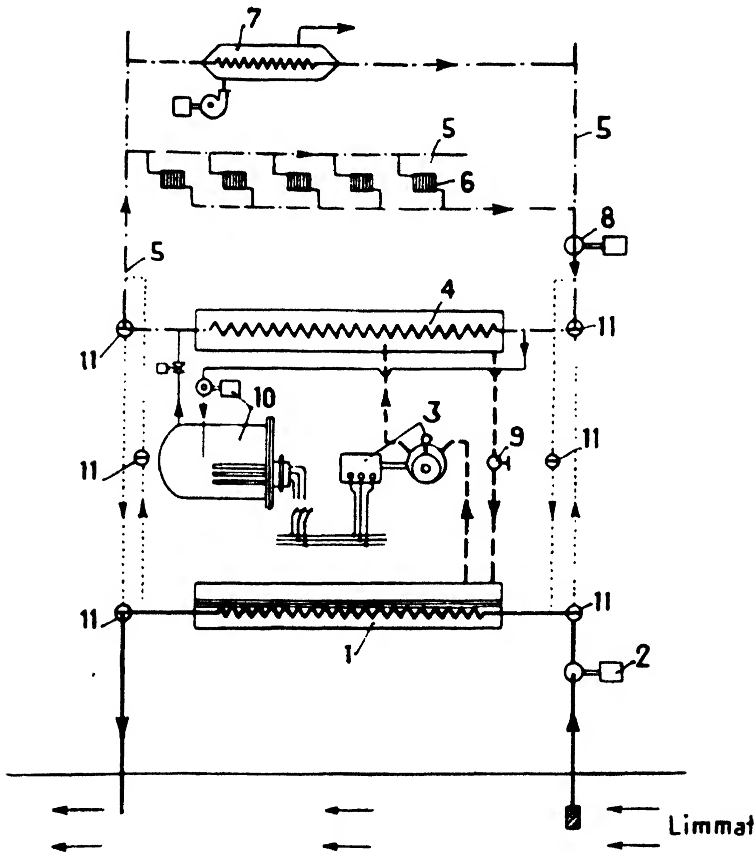

FIG. II—6.
Diagram of Heat Pump Installation of Zurich City Hall.

A further feature of the installation is that arrangements are provided in hot weather for reversing the circuits of the evaporator and the condenser in relation to the radiators, 6, and the air heat-transfer organ, 7. The water from the river now circulates by way of the pipe system 11, shown by dotted lines, through the condenser and, instead of giving up heat and suffering a fall of temperature, acts as a cooling medium and has its temperature raised before flowing back to the river. The circulating pipes, 5, are now connected to the evaporator, 1, and the water in these gives up heat to the working substance, so that the radiators, 6, act as coolers in the rooms, and the air passing through 7 has its temperature lowered instead of raised.

A still more complex system that has several points of interest is shown diagrammatically in Fig. II, 7. This was

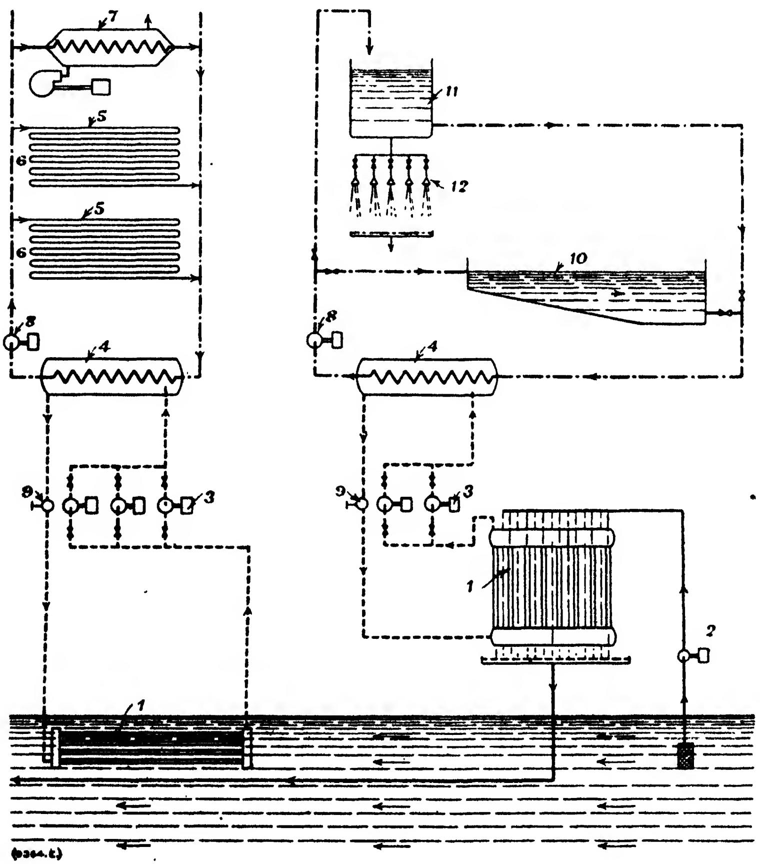

Fig. II—7.
Diagram of Heat Pump Installation of Zurich Swimming Baths.

installed by Messrs. Escher Wyss at the new Zurich Swimming Baths. In this case, the installation is arranged for two services, the space-heating of the building and the heating of the water for the swimming bath and the showers, each section calling on from two to three separate heat pumps as may be necessary to meet the greatly varying demands for heat at different seasons of the year.

In the space-heating system, to be seen on the left of the diagram, the conditions are generally similar to those to be met in the installation just described, when this is arranged for heating. The tubes conveying the working substance in the evaporator component, 1, however, are not placed in a casing, to which the water supplying the heat is pumped from and returned to the river, but are placed in a channel formed directly in the river. The working substance, after receiving heat from the river water, is drawn into the compressors, 3, in which it is compressed and from which it is delivered to the condenser, 4. The heat receiving medium in the condenser is again the hot water of the heating system and this, delivered at 122°F. (50°C.), is circulated by the pump, 8, through the pipes, 6, to the radiators, 5, of the general heating system, and to the heater, 7, for the ventilation air. The condensed working substance flows back through the regulator, 9, to the evaporator, 1.

The system on the right of the diagram caters for the swimming bath and the showers. The evaporator, 1, of the spray type, is away from the river, the water conveying the heat being circulated to the sprays and back by the pump, 2. The system is the same as that on the left as regards the arrangement of the compressors, 3, the condenser, 4, and the regulator, 9, but there are two sets of conditions for the hot water receiving heat from the condenser, 4 ; under one set of conditions the heat is supplied to the swimming bath, 10, and, under the other, to the showers, 12. When supplying the bath, the supply temperature is only 77°F. (25°C.) and the temperature of the water in the bath is 72°F. (22°C.) ; the water flowing out of the bath returns to the condenser. One compressor is normally sufficient for this duty. Heat is provided for the showers only by accumulation during the night and is stored in the accumulator 11, at 113°F. (45°C.).

When the heat pumps supply heat at 122°F. or 113°F. (50°C. or 45°C.), as when supplying space-heating or the hot water for the showers, respectively, the performance energy ratio is such that about 11,880 B.Th.U. (3,000 kcal.) per kW-hr. are delivered. Since 1 kW-hr. is equivalent to 3,412 B.Th.U. per hr., this gives a Performance Energy Ratio of $\frac{11880}{3412} = 3{\cdot}48/1$. When the heat pumps supply heat at the much lower temperature of 77°F. to the swimming bath, the performance energy ratio is approximately doubled and over 23,760 B.Th.U. (6,000 kcal.) per kW-hr. are delivered, corresponding to an actual performance energy ratio of nearly 7/1. The heating of the water for swimming baths, on account of the relatively small rise of temperature necessary, is thus a specially attractive application of the heat pump.

An electric boiler supplies the heat necessary for heating the water in the bath when this is completely refilled after its periodic cleaning. It provides also very hot water for heaters in the windows in the roof of the swimming bath hall in very cold weather and for any special requirements of very hot water, so that the heat pumps work only within the restricted ranges of temperature mentioned above. for which their performance energy ratios are reasonable, This boiler forms also a convenient reserve. The heat in the waste water from the swimming bath and from the showers is not wasted, as the water is led also to the sprays of the spray evaporator in which as much as practicable of this heat is transferred to the working substance.

If storage capacity for hot water is provided, a greatly increased flexibility in the operation of a heat pump is possible. For example, when driven electrically, the heat pumps may be driven and heat may be stored during the off-peak periods when the cost per unit is lower. Or, if the demand for heat is intermittent, a smaller heat pump than that necessary for maximum demand may be installed. Data concerning the rate of heat loss of a tank 12 in. (0·305 m.) in diameter and 60 in. (1·523 m.) long were provided by tests carried out at the Illinois Engineering Experiment Station*, the results of which are shown in Fig. II, 8. This tank is in the basement of the I.B.R.

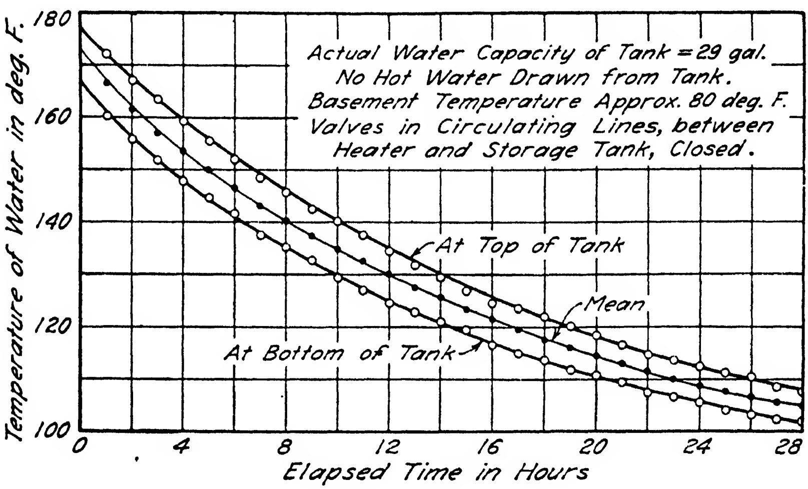

FIG. II—8.
Rate of Cooling of Water Storage Tank.

Research House, I.B.R. being an abbreviation for the Institute of Boiler and Radiator Manufacturers. The tank was lagged with 1 in. (25·4 mm.) thickness of air-cell insulation on its sides and bottom and with 1 in. (25·4 mm.) thickness of magnesia on the top. For a final temperature of 120°F. (49°C.) at the end of a period of 8 hrs., an initial temperature of 137°F. (58·4°C.) is necessary, with an ambient temperature of 80°F. (26·7°C.). The rate of heat loss even with this relatively small tank under these conditions is thus seen to be not excessive.

*University of Illinois Bulletin, No. 37, Fig. 6.

With a large tank, equally well insulated, this rate would be reduced, since loss is a function of surface area, which varies as the square of the linear dimension, and fall of temperature for a given loss varies inversely as the volume, that is, inversely as the cube of the linear dimension.

Mr. Sumner† also considered this question of thermal storage and presented the data in Table II, 2, which are based on his own records.

In considering these data, we see again that there are two points of view to be considered, namely, that of the consumer and the national one. For the consumer, there is the financial saving, to which must be added the cleanliness of his installation, following the absence of coal and ash on his premises ; there is also no need for the employment of a stoker. From the national point of view, coal is consumed under skilled direction with high efficiency at a central station, with less pollution of the atmosphere per ton burned and, further, much less coal is burned to produce the necessary heat, with the accompanying saving in transport of fuel and ashes. Finally, there is a reduction in the employment of low-grade labour, which, as we have seen, is an essential economic development for this country.

In any discussion of a conversion of a normal hot water space-heating service to heat pump operation, it is essential to take into account the radiating surface available in the existing radiators. In the Norwich case, these proved to have dimensions ample for heating with hot water supplied at 120°F., a temperature sufficiently low to bring the total range to a value consistent with a good Performance Energy Ratio for the heat pump. The water from heating boilers is normally intended to be at considerably higher temperatures and the dimensions of the radiators are determined so as to give the necessary heating effect at these temperatures. Circulation is often faulty so that the radiation surfaces are not used effectively over the whole system. Both the surfaces and the efficacy

†loc. cit.

of the circulation should be carefully considered in a possible conversion to heat pump operation.

TABLE II, 2

Comparative Annual Costs of Heating a Large Building

Seasonal Total Heat supplied : 20,000 therms ($5{\cdot}05 \times 10^9$ kcal.). Coal of heating value 12,000 B.Th.U. (3,030 kcal.) per lb. at 65/- per ton. Average combustion efficiency, 55 per cent. Cost of Electricity : (*a*) loads on peak, £4 per kVA plus 0·6d. per kW-hr. ; (*b*) loads off peak, 0·6 per kW-hr. Average Performance Energy Ratio, 4/1.

	Coal-fired boilers	Heat pump	
		Alone	With thermal storage
Capital cost, £	1,500	4,000	4,500
Annual capital charges, £	225 (15 per cent)	280 (7 per cent)	315 (7 per cent)
Cost of coal or electricity £	440	601	367
Attendance £	230 (including coal and ash handling)	—	—
Repairs and Maintenance £	150	50	50
Replenishing working substance £	—	25	25
Total annual cost £	1,045	956	765
Cost per therm (per 25,300 kcal.)	12·5d.	11·5d.	9·1d.

While it is clear from Mr. Sumner's data and from the other evidence that, under given conditions, the application of the heat pump to the space-heating of buildings is fully justified, it has, necessarily, the common drawback of all methods of space-heating in temperate climates, namely, that the utilisation is only seasonal.

The low value of the load factor, given by the ratio, $\frac{\text{actual heat per year}}{\text{potential heat per year}}$, or the small number of hours with full-load operation, expressed by the ratio, $\frac{\text{actual hours per year}}{\text{possible hours per year}}$, give a measure of this drawback. These low values are, of course, improved when the installation also provides cooling in hot weather, but, even then, the values in temperate climates remain only moderate. Industrial applications, which work throughout the year, can usually yield much higher values of the load factor, and are, as a result, much more favourable, in this respect, in relation to the financial advantage of the heat pump.

Those industries which require large quantities of hot water are particularly suited to operation with heating by a heat pump : of these, installations in which the hot water is used for washing and similar processes and, after use, is still at a fairly high temperature are naturally the best, since the heat in the used water may be utilised in providing some of the heat for the evaporator.

One industrial installation of considerable interest is that of the Steckborn Artificial Silk Company, of Steckborn, Switzerland, built in 1943, by Messrs. Brown Boveri and Company, Baden, Switzerland, and shown in diagrammatic form in Fig. II, 9. In rayon manufacture, considerable quantities of hot water at low and medium temperatures are necessary for heating the spinning baths, the spinning machines, the drying ducts and vats, as well as for the spun material itself. Conditions at Steckborn are particularly favourable for an installation embodying heat pumps, for the following reasons :

1, the works adjoin Lake Constance, from which practically unlimited supplies of low temperature heat are available ;

2, the demands for heat are considerable and are reasonably constant ;

3, the plant is in operation throughout the year and thus offers the possibility of a good load factor.

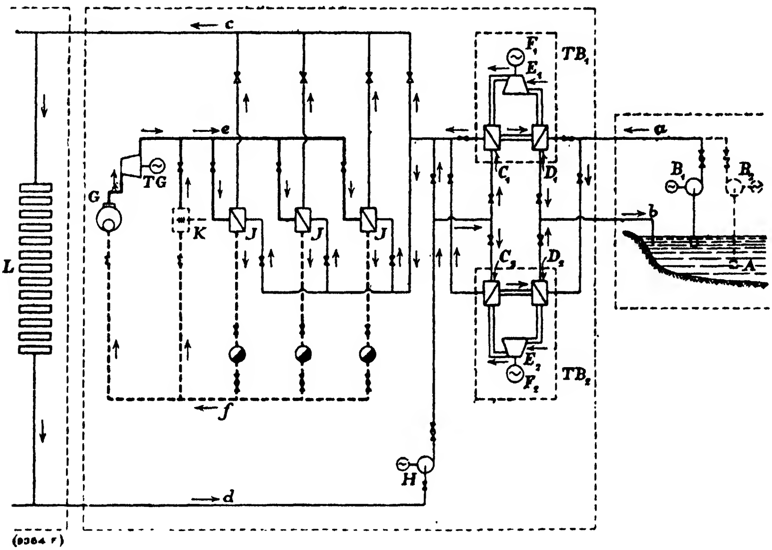

FIG. II—9.
Diagram of Industrial Heat Pump Installation at Steckborn, Switzerland.

Heat is required in two forms, namely, as hot water for heating and in the form of steam for processing. Steam was formerly also used for heating and the heating system was first converted to hot water heating ; this change was found advantageous because of improved cleanliness and the elimination of leakage of steam and condensate. High grade heat in the form of process steam is still supplied by boilers, but the necessary low grade heat is supplied by heat pumps, supplemented by the low grade heat available after the steam has been used in processing. The installation thus provides, from industry, an example in which a combination of high and low grade heat is involved.

Considering first the heat supplied by the heat pumps to the hot water system, it is convenient to begin at the right of the diagram, with the circulating systems of the lake water to the low temperature side of the heat pumps. A indicates the lake and there are two alternative pumps, B_1 and B_2, each arranged to take in the lake water at the highest practicable seasonal temperatures. In summer, B_1 is in use and draws water from near the surface of the lake at temperatures of from 55°F. (13°C.) to 68°F. (20°C.). In winter, B_2 draws water from near the bottom of the lake at about 39°F. (3°C.). The supply water is led through the pipeline *a* to the evaporators, D_1 and D_2, of the heat pump sets, TB_1 and TB_2, and, after suffering a fall of temperature ranging roughly from 2°F. (1°C.) in winter to 4°F. (2°C.) in summer, is returned through the pipeline *b* to the lake, the rate of circulation being adjusted to accord with operating and climatic conditions. Each heat pump set consists of D, the evaporator, E, the compressor, and C, the condenser, the suffixes 1 and 2 referring to the individual sets, TB_1 and TB_2. The working substance used is methyl chloride. The turbo-compressors E are driven by motors through three-speed gearing, so that the compressor speed can also be regulated to meet seasonal changes of temperature. Either or both of the sets can be in operation according to demand. The water circuits are shown in the diagram by thin firm lines, pipeline *c* showing the delivery to the elements, such as L, requiring hot water, and pipeline *d* showing the return of the cooled water to the condensers, C_1 and C_2, to have its temperature raised. Circulation is provided by the electrically-driven pump, H. The water is delivered from the condensers at 158°F. (70°C.) and is returned at 136°F. (58°C.).

Coming now to the steam circuits, these may be followed on the diagram by the thick firm and broken lines, the firm lines showing the process steam and the broken lines, the condensed steam. Two sources of steam are shown in the diagram : G is a coal-fired boiler from which the steam passes first to a turbo-generator set, TG, after which the steam passes, through the pipeline *e*, to those elements that require heat at relatively high temperatures, and is then led to the heat exchangers, J, in which heat may be given by the steam to the water in the hot water circuit ; from the heat exchangers the condensed steam is led, through the pipeline *f*, back to the boiler G. An alternative source of steam, in a condition suitable for the high-temperature elements, and thus not including the turbo-generator, TG, may be supplied by the electric boiler K, the circuits being otherwise as just described.

The combined systems offer a great range of flexibility to meet variation in demand and in climatic conditions. Automatic governing devices maintain constant the temperature of supply of the hot water and, at the same time, ensure that as much heat as possible is supplied by the heat pumps, both when the heat is derived from these alone and also when they are running in combination with the steam-to-water heat exchangers.

It is interesting to make a comparison, in relation to capacity, between this installation and that at Norwich : in the latter, the mean rate of heat delivery was 5 therms ($5 \cdot 10^5$ B.Th.U. or $1 \cdot 26\ 10^5$ kcal.) per hour with a maximum capacity of 8 therms ($8 \cdot 10^5$ B.Th.U. or $2 \cdot 02.\ 10^5$ kcal.) per hour, while the maximum capacity of *each* of the two heat pump units at Steckborn is 67·4 therms ($6 \cdot 74.\ 10^6$ B.Th.U. or $1 \cdot 7\ 10^6$ kcal.) per hour, over eight times as great.

A case in which the same compressor may serve simultaneously both a refrigerator and a heat pump installation is provided by a cold storage installation and any adjoining premises, such as offices, which require heating. In cold weather, with a sufficiently high pressure ratio for the compressor, the circulating water from the condenser of the refrigerator can be passed through the hot water radiators of these premises. A suitable relationship must, of course, exist between the dimensions of the cold storage room and of the adjoining premises, but, when this is the case, the heat for the heating system is derived from the heat at low temperature leaking into the cold stores together with the work done in the compressor.

The possible combinations embodying heat pumps in industrial processes are clearly both numerous and diverse, but the position has certainly been reached that all cases in which heat is supplied directly by the combustion of fuel, particularly those in which live steam or hot water may be used and allowed afterwards to convey considerable quantities of heat to waste, should be carefully examined in relation to the possible application of heat pumps, alone or in combination. Installations in which large quantities of electrical energy are applied directly for heat at relatively low temperatures should similarly be examined.

There is one especial circumstance in this country that renders the possibilities of the applications of heat pumps particularly favourable, namely, the long coast line and the numerous towns situated near to tidal waters. The sea provides an inexhaustible source of low temperature heat and, in our latitudes, is only extremely rarely liable to freezing.

One last point, in concluding this lecture, is concerned with the difficulty that may at present be experienced in obtaining the actual components of a heat pump installation, even if the resultant saving in fuel consumption is clearly demonstrable. Apart from the other advantages of cleanliness and the saving of manual labour, it is clear that, from the present national standpoint, the provision of mechanical equipment, such as heat pumps, which would reduce fuel consumption, either directly or as a result of saving in electric energy, should have a priority equal to that of new power station plant.

LECTURE III

In this lecture, we shall first consider heat pumps in which the working substances are exclusively, or mainly, in gaseous form ; we shall then go on to deal with the principles upon which air conditioning is based, and shall conclude by considering two installations in which air conditioning is carried out with the aid of heat pumps.

In Lecture I, in studying the principles of heat pumps with gaseous working substances, we limited ourselves to those cases in which dry air was the working substance and in which the cycle of operations followed was the Joule or constant pressure cycle. Two principal conclusions were drawn :

(*a*) that the Performance Energy Ratio is higher, the smaller the pressure ratio, R ;

(*b*) that, for a given value of R, the Performance Energy Ratio is rapidly reduced as the component efficiencies of compression and expansion, η_c and η_t, respectively, become smaller.

In connection with Fig. I, 6, which, for convenience, is reproduced here, with a scale of Total Heat added,* as Fig. III, 1, the following expression was deduced :

$$\text{Performance Energy Ratio} = \frac{T_C - T_A + \frac{1}{\eta_c}(T_E - T_C)}{\frac{1}{\eta_c}(T_E - T_C) - \eta_t(T_A - T_F)}.$$

Satisfactory as this expression is, it does not indicate all the limitations that may be imposed, by external conditions, on the cycle when it is applied to the operation of a heat

*With C_p assumed to be constant, as is reasonable in this restricted range of temperature.

pump. It will be noticed that the temperatures all relate to the ideal cycle, CEAF, and that the only other quantities present are η_c and η_t. In this cycle, it will be remembered, the ratios of the upper and lower temperatures of the adiabatic stages, CE and AF, are given by:

$$\frac{T_E}{T_C} = \frac{T_A}{T_F} = (R)^{\frac{\gamma-1}{\gamma}}.$$

For convenience in writing, let $(R)^{\frac{\gamma-1}{\gamma}}$ be denoted by k.

Then $$T_C = \frac{T_E}{k} \text{ and } T_F = \frac{T_A}{k}.$$

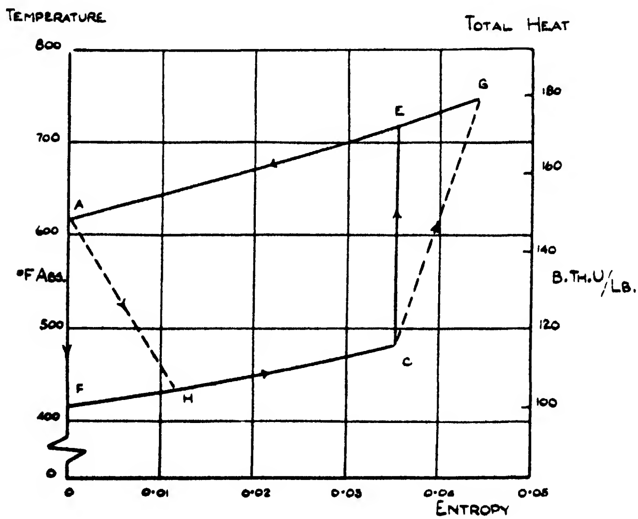

FIG. III—1.
Temperature-Entropy Diagram for Joule Cycle.

The above expression, with the order of the terms in the numerator changed, may thus be re-written:

$$\text{Performance Energy Ratio} = \frac{\frac{1}{\eta_c}\left(T_E - \frac{T_E}{k}\right) + \frac{T_E}{k} - T_A}{\frac{1}{\eta_c}\left(T_E - \frac{T_E}{k}\right) - \eta_t\left(T_A - \frac{T_A}{k}\right)}.$$

Multiplying through by k and η_c and simplifying, the right-hand side becomes :

$$\frac{T_E(k-1)+\eta_c(T_F-T_A k)}{T_F(k-1)-\eta_c\,\eta_t\,T_A(k-1)}=\frac{T_E+\eta_c\left(\dfrac{T_F-T_A k}{k-1}\right)}{T_E-\eta_c\,\eta_t\,T_A}.$$

Dividing through by T_A, the expression can be re-written :

$$\text{P.E.R.}=\frac{\dfrac{T_E}{T_A}+\eta_c\left(\dfrac{\dfrac{T_F}{T_A}-k}{k-1}\right)}{\dfrac{T_E}{T_A}-\eta_c\,\eta_t}=\frac{m+\eta_c\left(\dfrac{m-k}{k-1}\right)}{m-\eta_c\,\eta_t},$$

in which $$m=\frac{T_E}{T_A}=\frac{T_C}{T_F}.$$

Since EA and FC, the stages during which heat transfer takes place, are at constant pressure,

$$m=\frac{T_E}{T_A}=\frac{T_C}{T_F}=\frac{V_E}{V_A}=\frac{V_C}{V_F}.$$

Consideration of the expression in its final form shows that this is not dependent upon the actual values of these temperatures, but is concerned only with the value of the ratio m ; it is, further, concerned with the values of η_c and η_t, which are ratios, and with the value of k, which, denoting $(R)^{\frac{\gamma-1}{\gamma}}$, is a ratio raised to a power. This expression may thus be used to find the value of the Performance Energy Ratio for any case in which these four ratios are known.

Reference has already been made to the importance of R, and thus k, and of the component efficiencies, η_c and η_t, in connection with their influence upon Performance Energy Ratio. The examples taken, however, were limited in their scope and it is of interest to extend the ranges of values of these quantities and to include also a suitable range of possible values of the ratio m. Calculations have accordingly been made and some of the results are set out in Table III, 1.

TABLE III, 1.

Performance Energy Ratio in Relation to R, m, η_c and η_t.

Line	R	k	m	Values of $\eta_c = \eta_t$					
				1·0	0·95	0·9	0·85	0·8	0·75
1	4	1·486	1·161	3·06	2·04	1·59	1·35	1·20	1·10
2	3	1·369	1·161	3·71	2·42	1·86	1·55	1·36	1·23
3	2	1·219	1·161	5·56	3·52	2·63	2·14	1·82	1·60
4	1	1·122	1·161	9·20	5·68	4·13	3·27	2·72	2·34
5	2·0	1·219	1·046	5·56	2·06	1·42	1·16	1·02	0·94
6	2·0	1·219	1·078	5·56	2·66	1·86	1·50	1·29	1·15
7	2·0	1·219	1·161	5·56	3·52	2·63	2·14	1·82	1·60
8	2·0	1·219	1·219	5·56	3·86	2·98	2·46	2·10	1·85
9	1·442	1·110	1·046	10·08	3·45	2·21	1·71	1·43	1·26

These results show that, in all cases, the values of Performance Energy Ratio suffer a considerable reduction as the component efficiencies, η_c and η_t, are reduced ; lines 1 to 4 show that decreases in R, and thus in k, lead to improved performance ; lines 5 to 8 show that decreases in m, on the other hand, lead to worsened performance.*

Important as these results are, they must naturally be brought into a suitable relationship with the conditions of application of the cycle in practice. Referring again to Fig. III, 1, it is seen that, in the ideal cycle CEAF, heat is received during the stage FC and rejected during the

* Such conclusions follow, of course, from differentiating the expression,

$$\text{P.E.R.} = \frac{m + \eta c \left(\frac{m - k}{k - 1}\right)}{m - \eta c\, \eta t},$$

with respect to m, k, ηc and ηt, separately taking for each differentiation, the other variables as constant. But, for present purposes, the setting-out of actual results is particularly convenient.

stage EA. T_C is thus the lowest possible temperature of the source of heat and T_A is the highest possible temperature of the medium to which the heat is delivered. It is possible to express the overall range of temperature, $(T_E - T_F)$, of this cycle in terms of k and m, η_c and η_t in the ideal case having the value, 1·0 :

$$\begin{aligned} T_E - T_F &= (T_E - T_A) + (T_A - T_F) \\ &= T_A(m - 1) + T_A\left(1 - \frac{1}{k}\right) \\ &= T_A\left(\frac{km - 1}{k}\right) = T_F(km - 1) \\ &= T_C\left(\frac{km - 1}{m}\right) = T_E\left(\frac{km - 1}{mk}\right). \end{aligned}$$

In a cycle in which η_c and η_t are less than unity, such as CGAH in the same figure, the range of temperature is $(T_G - T_H)$. The working substance receives heat during the stage HC and rejects it during the stage GA. T_C thus remains, as before, the lowest possible temperature of the source and T_A, the highest possible temperature of the receiver of heat.

The efficiency of compression $\eta_c = \dfrac{T_E - T_C}{T_G - T_C}$, so that :

$$T_G = T_C + \frac{1}{\eta_c}(T_E - T_C).$$

Substituting k T_C for T_E, and remembering that T_C k $= T_A$ m :

$$T_G = T_C\left[1 + \frac{k - 1}{\eta_c}\right] = T_A\frac{m}{k}\left[1 + \frac{k - 1}{\eta_c}\right].$$

Similarly, the efficiency of expansion, $\eta_t = \dfrac{T_A - T_H}{T_A - T_F}$,

from which : $\quad T_H = T_A - \eta_t(T_A - T_F).$

But $T_A = T_F$ k, and :

$$T_H = T_A\left[1 - \eta_t\left(1 - \frac{1}{k}\right)\right] = \frac{T_A}{k}[k - \eta_t(k - 1)]$$

$$= \frac{T_C}{m} [k - \eta_t (k - 1)]$$

Thus,

$$(T_G - T_H) = T_A \frac{m}{k} \left[1 + \frac{k-1}{\eta_C} - \frac{k}{m} + \frac{\eta_C (k-1)}{m} \right]$$

$$= \frac{T_A}{k} \left[\left(\frac{m}{\eta_C} + \eta_t \right)(k-1) + m - k \right]$$

$$= \frac{T_G}{m} \left[\left(\frac{m}{\eta_C} + \eta_t \right)(k-1) + m - k \right]$$

in terms of the limiting values, T_A and T_C, respectively.

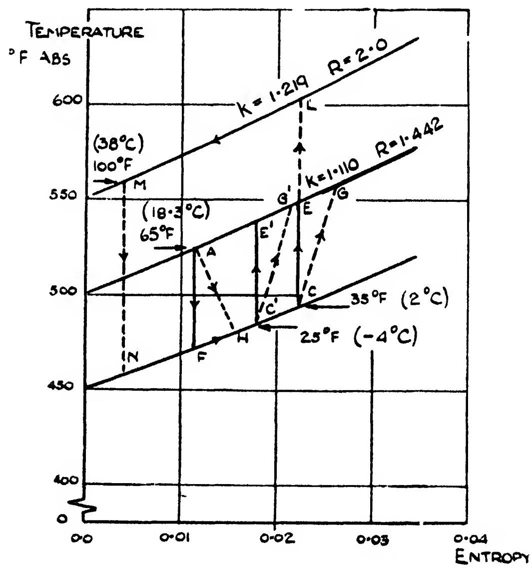

Fig. III—2.
Temperature-Entropy Diagrams for Three Joule Cycles.

Fig. III, 2 is a temperature-entropy diagram for three ideal cycles : CEAF, C′E′AF and CLMN. For the first two cycles, R = 1·442 and k = 1·110 ; for CLMN, R = 2 and k = 1·219. Consider first the cycle CEAF : for this cycle, T_c = 495°F. abs. (275°C. abs.) or 35°F. (2°C.) ; that is to say, the highest temperature at which heat is received is 35°F. (2°C.), a temperature which is of the order of that of a river or other external liquid source of low temperature heat. T_A, the lowest temperature at which heat may be delivered, is shown as 525°F. abs. (292°C. abs.) or 65°F. (18·3°C.), ordinary room temperature. Under these conditions, m is fixed and equal to

$$\frac{T_E}{T_A} = \frac{T_C\,k}{T_A} = \frac{495.\ 1{\cdot}110}{525} = \frac{549}{525} = 1{\cdot}046.$$

The values of the Performance Energy Ratio corresponding to these values of k and m are given, both for the ideal cycle and for those with certain equal values of η_c and η_t, in line 9 of Table III, 1.

Some consideration should be given at this point to the possible values in practice of η_c and η_t. It has been convenient, in the calculations, to take these as equal but normally it is possible, with corresponding designs, to obtain somewhat higher values for η_t, for expansion, than for η_c, for compression. Incidentally, this applies also to the expansion and compression components of gas turbine installations. It follows from the fact that, whereas losses in the early part of an expansion can be recouped, to some extent, in the later part of the expansion, losses in a compression process, are cumulative. Actually, with modern design and with limited pressure ratios, efficiencies of the order of 0·85 should readily be realised in machines of moderate and large capacity, and this value will be assumed in the discussion that follows.

With $\eta_c = \eta_t = 0{\cdot}85$, applied to the cycle CEAF, the modified cycle becomes CGAH.

$$T_G = T_A\frac{m}{k}\left[1 + \frac{k-1}{\eta_c}\right] = 525\cdot\frac{1{\cdot}046}{1{\cdot}110}\left[1 + \frac{1{\cdot}110 - 1}{0{\cdot}85}\right]$$

$$= 525\,\frac{1{\cdot}046}{1{\cdot}110}\,1{\cdot}129 = 556°\text{F. abs. } (309°\text{C. abs.})$$

Taking C_p as 0·24, the heat delivered per lb. of air = $C_p(T_G - T_A) = 0{\cdot}24\,(556 - 525) = 7{\cdot}44$ B.Th.U. per lb. (4·13 kcal. per kg.)

$$\text{With } \eta_t = 0{\cdot}85,\ T_H = \frac{T_A}{k}\,[k - \eta_t\,(k - 1)]$$

$$= \frac{525}{1{\cdot}110}\,[1{\cdot}110 - 0{\cdot}85\,(1{\cdot}110 - 1)]$$

$$= \frac{525}{1{\cdot}110}\,(1{\cdot}110 - 0{\cdot}093) = 525\,\frac{1{\cdot}017}{1{\cdot}110}$$

$$= 481\,^\circ\text{F. abs. } (267\,^\circ\text{C. abs.}).$$

The heat received per lb. from the low temperature source is thus $0{\cdot}24\,(495 - 481) = 3{\cdot}36$ B.Th.U. per lb. (1·87 kcal. per kg.). The Performance Energy Ratio for CGAH was calculated earlier and, in Table III, 1, line 9, is given as 1·71, compared with 10·08 for the ideal cycle.

It is of interest at this stage of our discussions to compare certain of these results with those of the Norwich experiment studied in Lecture II. The average heating demand in that installation was 5 therms ($5{\cdot}10^5$ B.Th.U. or $1{\cdot}26{\cdot}10^5$ kcal.) per hour. To supply this quantity of heat by the cycle just described would involve a mass flow of $\frac{500000}{7{\cdot}44} = 67{\cdot}200$ lb. (30,550 kg.) of air per hr. This is to be compared with a mass flow of sulphur dioxide in the Norwich case of the order of $\frac{500000}{150} \cdot \frac{3{\cdot}5}{4{\cdot}5} =$ 2,590 lb. (1,294 kg.) per hr. The necessary capacities of the compressors in the two cases are thus in the proportion of over 100 to 1, assuming the air at suction to the compressor in the former case to be at atmospheric pressure, whereas vaporised sulphur-dioxide is the substance handled by the compressor in the latter case. Taking the Performance Energy Ratio at Norwich as 3·5, against 1·71 in the present case, the powers necessary would be, roughly, 42 and 86 kW, that is, in the ratio of 1 : 2·05 although, as will be seen later, other losses in the air heat pump have yet to be taken into account.

In the cycle CEAF on Fig. III, 2, the highest temperature at which heat was received was 35°F. (2°C.).

If heat is received from the external atmosphere, the temperature may, under severe winter conditions, be reduced to 25°F. (− 4°C.). The cycle C′E′AF is arranged to meet these conditions, the lowest temperature at which heat may be delivered, T_A, being as before, 65°F. (18·3°C.). $T_{E'}$ is now 538°F. abs. (291°C. abs.), so that $m = \frac{538}{525} = 1{\cdot}023$.

With other conditions remaining constant, the value of m depends on the temperature range during one or other of the constant pressure operations of the cycle. Taking $\eta_c = \eta_t = 0{\cdot}85$, $T_{G'}$ is 547°F. abs. (304°C. abs.), and the heat delivered is 0·24 (547 − 525) = 5·28 B.Th.U. per lb. (2·93 kcal. per kg.). For the same heating load as in the last paragraph, the necessary mass flow would be 94,700 lb. (43,050 kg.) per hr., that is, 40 per cent greater. The Performance Energy Ratio of this cycle, C′G′AH, is :

$$\frac{m + \eta_c\left(\dfrac{m - k}{k - 1}\right)}{m - \eta_c\,\eta_t} = \frac{1{\cdot}023 + 0{\cdot}85\left(\dfrac{1{\cdot}023 - 1{\cdot}110}{0{\cdot}110}\right)}{1{\cdot}023 - 0{\cdot}85\;0{\cdot}85}$$

$$= \frac{1{\cdot}023 - 0{\cdot}672}{1{\cdot}023 - 0{\cdot}723} = \frac{0{\cdot}351}{0{\cdot}300} = 1{\cdot}17,$$

compared with 1·71 for the cycle CGAH ; so that, for the same heating load, the necessary power is increased in the proportion 1·71/1·17 or by 46 per cent. This increase, found necessary in order to extend the temperature range ($T_A - T_C$) by only 10°F. (5·6°C.), gives emphasis to the importance of limiting this range as much as practicable.

The two cycles considered, CEAF and C′E′AF (or CGAH and C′G′AH, respectively, when modified for $\eta_c = \eta_t = 0{\cdot}85$) have the same value of k, but the former has a greater value of m ; it shows both a higher Performance Energy Ratio and a lower mass flow for a given heating load than the latter. Inspection of Fig. III, 2 shows, however, that for a given value of k, m is limited by the desired temperature range: since T_C cannot exceed T_A, $m = \frac{T_C}{T_F}$ cannot exceed $k = \frac{T_A}{T_F}$. In order to increase the temperature range and, at the same time, to have a practicable value of m, an increase of k is essential. The third ideal cycle shown, CLMN, has a temperature range $(T_M - T_C) = (100°–35°) = 65°F. (36°C.)$. The value of $m = \frac{604}{560} = 1{\cdot}078$. The data for this case are given in line 6 of Table III, 1, from which, with $\eta_c = \eta_t = 0{\cdot}85$, the Performance Energy Ratio is 1·50. With T_M constant, the range, $(T_M - T_C)$, may be reduced by increasing T_C: the limiting value of m is $\frac{560}{459} = 1{\cdot}219$ at which $T_C = T_M$, for which the data are given in line 8 of the table. Fig. III, 3, gives, as a matter of interest, the pressure-volume diagrams of the cycles in Fig. III, 2.

These examples demonstrate that while, as was shown earlier, for a high value of the Performance Energy Ratio, m should be as great as possible and k should be as small as possible, the practicable values of m and k are not chosen at will but are determined by the external conditions of a particular design.

It is desirable next to consider possible arrangements of heat pumps to work on the typical cycles given in the figure. The chief provision to be made is for the delivery of heat from the working substance, at the upper pressure, and for the transfer of heat to the working substance at the lower pressure; these functions correspond to those of the condenser and the evaporator, respectively, in the vapour-compression cycle. For supplying space-heating to buildings, the three arrangements shown diagrammatically in Fig. III, 4 (*a*), (*b*) and (*c*) may be considered. In all the diagrams, M is the machine, electrical or otherwise, giving the mechanical power necessary to meet the difference between the work done *on* the air by the compressor component, C, and the work done *by* the air in the expansion component, E ; X_a, X_b, X_c and X_d show heat exchangers ; T_C, T_G, T_A and T_H are the temperatures at the points in the circuits corresponding to points such as C, G, A and H on Fig. III, 1.

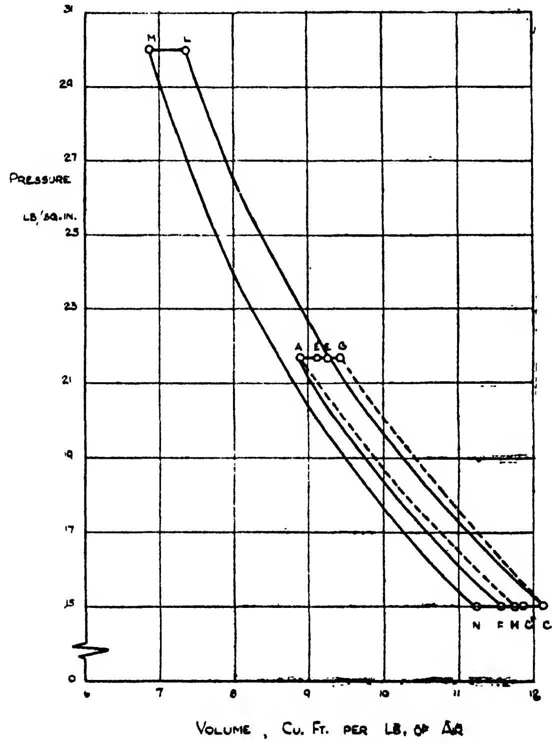

Fig. III—3.

Pressure-Volume Diagrams for the Cycles of Fig. III—2.

In (*a*) the upper pressure, which prevails at points G, E and A, is atmospheric and the lower pressure, prevailing at points H and C, is less than atmospheric. The compressor, C, delivers air at T_G directly into the building, while the expansion component, E, draws an equal mass of air from the building at T_A, the heat delivery stage thus taking place in the air in the building. The air expands in E to the lower pressure, 1/R atmosphere, and to the temperature T_H, and is then rejected through the heat exchanger X_a. On the other side of the heat transfer unit, forming the source of low temperature heat, atmospheric air, for example, is circulated by means of the pump P and, by giving up its heat to the air leaving the expansion component at T_H, raises the temperature of this air to the value T_C, at which it is drawn into the compressing component. SF show the ventilation system, with U as an alternative point of supply.

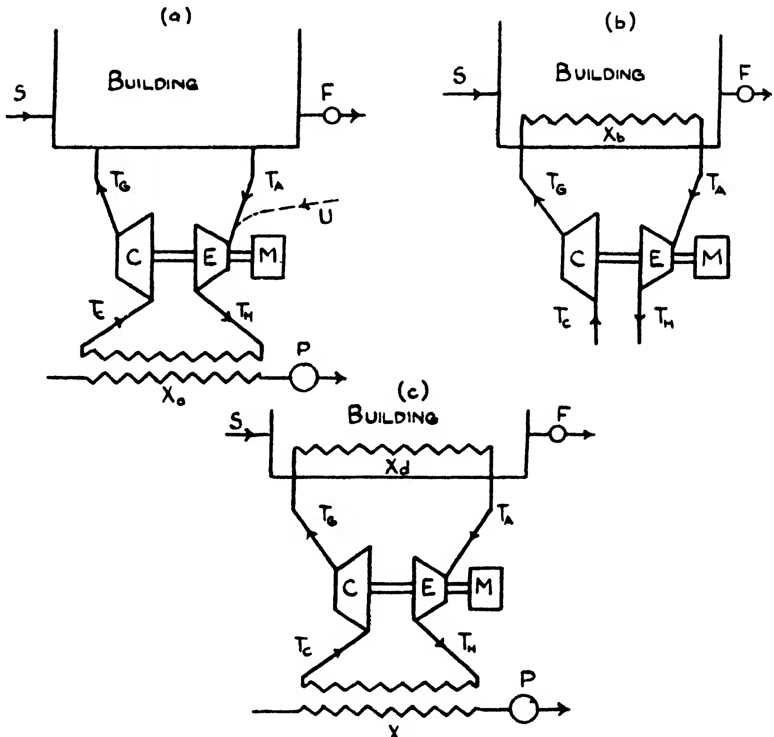

FIG. III—4.
Three Arrangements for Space-Heating.

In (*b*) the lower pressure, at points H and C on Fig. III, 1, is atmospheric ; air is drawn from the atmosphere at T_C, compressed in the compressor C and delivered, at

T_G, to the heat exchanger, X_b, in which the pressure is R atmospheres. The compressed air gives up heat in X_b to the air in the building, suffering a fall of temperature from T_G to T_A, and then passes to the expansion component, in which it expands to atmospheric pressure and to T_H, and is then discharged to the atmosphere. The heat reception stage HC takes place in the external atmosphere.

In (*c*) there is no restriction on the order of the pressures. Heat is received between H and C, at the lower pressure, in the heat exchanger X_c. The air at T_C is compressed to the upper pressure, R times the lower value, and delivered at T_G to the heat exchanger X_d, in which heat is delivered by transfer to the air in the building, being cooled to T_A ; it passes at this temperature to the expansion component E, does work, and returns at T_H to X_c.

Comparing the three arrangements, it is seen that, whereas (*a*) and (*b*) each involve only one heat exchanger, (*c*) shares the same drawback as the vapour-compression heat pump and requires two heat exchangers. In X_a, heat is transferred from air or water at atmospheric pressure to air at 1/R atmosphere ; in X_b, the pressure within is R atmospheres while the pressure external to the exchanger is atmospheric ; in X_c and X_d, the pressure on the outside is atmospheric but the internal pressures can be chosen at will, providing the ratio of that in X_d is R-times that in X_c, and will be normally above atmospheric in order to increase the density and thus to reduce the dimensions of the installation for a given load. As regards the sizes of the heat exchangers necessary for the same heating load, and for the same temperature differences in the exchangers, the order will then be, from smallest upwards : X_d, X_c, X_b, X_a.

Considering the necessary ranges of temperature of the working substance in the three cases, it is seen that : in (*a*), T_A is room temperature, but T_C must lie somewhat below that of the external source of heat, for heat to be transferred in X_a ; in (*b*), T_A should still be above room temperature to facilitate heat transfer in X_b to the air in the building, while T_C is that of the external atmosphere ; similarly, in (*c*), T_A must be above room temperature and T_C lower than that of the external source of heat, which would lead to the temperature range in this case being higher than in the arrangements (*a*) and (*b*).

In considering the efficacy of a heat exchanger, the performance must be regarded from the points of view of heat transfer and of the mechanics of fluids. With the ideal conditions that would be associated with an infinitely large exchanger, the temperature of the heat-receiving fluid leaving the exchanger would be equal to temperature of the heat-delivery fluid at entry ; the fall of pressure necessary to pass the fluids would be zero and no mechanical losses would occur from this source. In actual heat exchangers, conditions of cost and space limit the sizes and, on the one hand, temperature differences between the entering heat-delivery fluid and the leaving heat-receiving fluid are unavoidable, and, on the other hand, the differences of pressure necessary to overcome friction in one or both fluids lead to mechanical losses. The former lead to an increase in the necessary range of temperature of the working substance of the heat pump and thus to a reduction in the performance energy ratio ; the latter lead to an increase in the amount of work to be done to circulate the fluids and thus also bring about a reduction of the performance energy ratio. In a particular installation, therefore, a suitable compromise must be made between the reduction in capital cost, space occupied, etc., that results from a decrease in the size of a heat exchanger, and the increase in running costs, owing to the decrease in the Performance Energy Ratio caused by the increase of the range of working temperatures and the increase in mechanical work. More detailed consideration of heat exchangers is not practicable here.*

* A valuable review of existing knowledge in this subject will be found in the late Professor C. H. Lander's paper : *Proc. I. Mech. E.*, 1942, Vol. 148, p. 81 and 1943, Vol. 149, p. 147.

Summarising the total mechanical work to be provided in a heat pump installation of this kind, we have :

1, the net work done in compression and expansion, as determined by the actual values of η_c and η_t ;

2, the work done in passing the working fluid through the heat exchanger or exchangers ;

3, the work done in overcoming the fluid friction in the pipes connecting the exchanger components to the compression and expansion components ;

4, the work that may be done in circulating the secondary fluids concerned with the heat-delivery to the working substance at the lower pressure and the heat reception from the working substance at the higher pressure.

The ratio of the heat delivered to the total work done gives the overall performance energy ratio of the heat pump. In order to obtain the overall performance energy ratio of the complete installation, the losses in the driving machine, electrical or otherwise, must be added to the total work done in the heat pump proper.

Although, in relation to the Joule or constant pressure cycles so far considered, the space-heating of buildings has been mentioned, this has been limited to the supply of heat and to the assumption that the working substance is in the form of a single gas. Everything said previously, therefore, can be applied to industrial installations in which any suitable gas may be employed, and in which the ranges of pressure and temperature permit values of R, k, m and the component efficiencies, η_c and η_t, consistent with the attainment of reasonably high values of performance energy ratio.

In many cases, such as those already considered in connection with Figs. I, 2 ; II, 1 ; II, 6 and II, 7, heat, or its opposite, cold, as in Fig. II, 6, is all that it is necessary to supply to the respective buildings. This is a question partly of climate and partly of the purposes for which the buildings are used. For many buildings, however, and particularly those used for certain industrial processes and for accommodating large assemblies of people in climates with great differences between outdoor and desirable room temperatures, heating, cooling and air-conditioning are all necessary.

As regards cooling, all three of the arrangements shown in Fig. III, 3 may, with suitable changes, operate as means for cooling the buildings. The resulting cycle of operations is shown, as a temperature-entropy diagram (or total heat-entropy diagram) in Fig. III, 5, and the re-arranged components and circuits are given in Fig. III, 6.

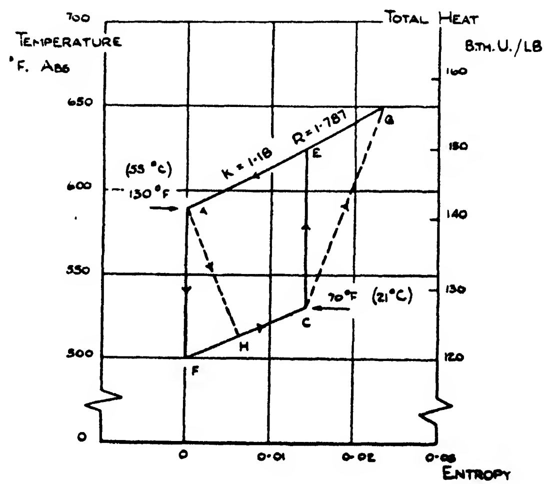

FIG. III—5.

Temperature-Entropy Diagram for Cooling Circuit.

The cycle shown in Fig. III, 5, is arranged with ideal heat exchange, for a room temperature, T_C, of 70°F. (21°C.) and for an outside temperature, T_A, of 130°F. (55°C.). With T_F taken as 500°F. abs. (278°C. abs.), $k = 1{\cdot}18$ and $R = 1{\cdot}787$. With $k = 1{\cdot}18$, $T_E = 1{\cdot}18 \times 530 = 625$°F. abs. (347°C. abs.). Taking, as before, $\eta_c = \eta_t = 0{\cdot}85$, $T_H = 513{\cdot}5$°F. abs. (285°C. abs.), and $T_G = 650$°F. abs. (361°C. abs.). Heat is received by the working substance in the building from H to C and is rejected to the external cooling medium from G to A. Although the cycle remains the same, the function of the installation, since it "brings about the removal of heat from a body at a temperature lower than that of its ambient," is now that of a refrigerator, rather than that of a heat pump, and the criterion of performance, $\frac{\text{Heat Received}}{\text{Work Done}}$, is the Coefficient of Performance.

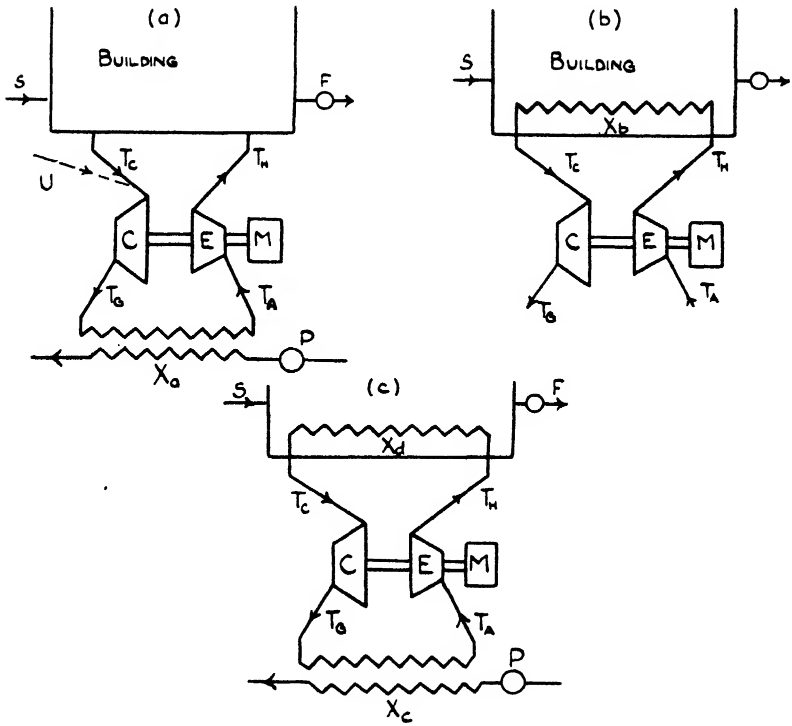

FIG. III—6.
Three Arrangements for Space-Cooling.

In (*a*) air at room temperature, T_C, and at atmospheric pressure, is now drawn into C, compressed to R atmospheres and delivered at T_G into X_a, in which heat is given up during the stage GA ; it now passes at T_A to E, in which it is expanded again to atmospheric pressure and to T_H and then discharged into the building, in which heat is received during the stage HC, and the temperature raised again to T_C. The pressure of the air in X_a is now R atmospheres against 1/R atmosphere in Fig. III, 4, and this must be provided for in the design. Additional piping is necessary for the new cooling circuit and the direction of flow of the external fluid, now receiving heat in X_a, should be in the opposite direction to that desirable in Fig. III, 4, in which the external fluid gave up heat to the working substance. In order that C and E, after the change, may continue to work under favourable conditions, it may also be necessary to arrange that their relative and actual speeds of revolution should be changed.

In (*b*) and (*c*) similar re-arrangement will be necessary. In (*b*) the pressure in X_b is now 1/R atmosphere instead of R atmospheres, as previously ; the stage in which heat is received by the working substance, from H to C, now takes place in X_b, and the stage for the rejection of heat, from G to H, in the atmosphere. Similarly, in (*c*), the pressure in X is now R times that in X_d instead of, as previously, 1/R of the pressure in X_d. The heat exchangers, X_b and X_c, must be designed to accommodate the new conditions of pressure. The direction of flow of the external fluid in X_c must, similarly with that in X_a, be reversed.

Owing to the effects of the deficiency of the compression and expansion components, η_c and η_t being less than 1·0, the rise of temperature of the working substance in the building ($T_C - T_H$) in Fig. III, 5, will probably be much smaller than the corresponding fall in the case of Fig. III, 2, namely ($T_G - T_A$). The mass flow of the working substance around the circuit must, therefore, be adjusted accordingly.

In passing, it will be noticed that the temperatures assumed in Fig. III, 5 are extreme, since 130°F. (55°C.) is very high for that of the external source and 70°F. (21°C.) for the room temperature is lower than would be thought comfortable in such a hot climate. The range $(130° - 70°) = 60°F. (33°C.)$ conveniently gave $k = 1{\cdot}18$ and $R = 1{\cdot}787$; this pressure ratio, while still small, is higher than those found desirable to meet the conditions of the diagrams in Fig. III, 2.

As a matter of interest, the value of the Coefficient of Performance of the refrigerator cycle of Fig. III, 5

$$= \frac{T_C - T_H}{(T_G - T_C) - (T_A - T_H)}$$

$$= \frac{530 - 513{\cdot}5}{(650 - 530) - (590 - 513{\cdot}5)}$$

$$= \frac{16{\cdot}5}{120 - 76{\cdot}5} = \frac{16{\cdot}5}{43{\cdot}5} = 0{\cdot}38.$$

From the point of view of the heat receiving external medium, the installation is still a heat pump, of which the Performance Energy Ratio $= \frac{T_G - T_A}{43{\cdot}5} = \frac{650 - 590}{43{\cdot}5}$

$= \frac{60}{43{\cdot}5} = 1{\cdot}38$, as must be the case from the definitions of these two quantities.

These calculations for employing the reversed Joule or constant pressure cycle for cooling, like those given earlier for heating, relate to dry air and the reasoning employed may similarly be applied to other cases in which single gases are present. As was stated earlier, however, many cases of space-heating, or space-cooling, also involve air conditioning and this will now be studied. Air conditioning means, for a building designed primarily to accommodate people, keeping the air in a state consistent with health and comfort, or, for an industrial building, keeping the air in the state desired for the industrial operations to be carried on there. It involves, therefore, not only control of the air temperature, for which heating or cooling may be necessary, but also control of the humidity of the air, by humidifying or by drying, and of its purity in respect of dust, bacteria and objectionable gases.

As regards human comfort, it is now recognised that this is determined much more by the relative humidity than by the temperature as measured by the ordinary or dry-bulb thermometer : for instance, 80°F. (27°C.) with high humidity may be more trying than 110°F. (43°C.) with dry air. Atmospheric air is a mixture of dry air with different proportions of water vapour, the air and the water vapour each contributing its partial pressure to the total atmospheric pressure. The wet bulb thermometer gives the temperature at which the water vapour present will exert its saturation pressure. The Relative Humidity of the air is thus given as the ratio of the pressure of the vapour present, when exerting its saturation pressure at its dew point, to the saturation pressure corresponding to the dry bulb temperature. The Specific Humidity of the air is the weight of water vapour, w lb., present in the atmosphere, per lb. of dry air. The Total Heat of air $= H_a + wH_s$, in which H_a is the Total Heat of 1 lb. of dry air and H_s is the Total Heat of 1 lb. of vapour at the dry bulb temperature.

In air conditioning, the following processes may be carried out :

1, Humidifying, by passing the air through the sprays of a washer, thus increasing its content of water vapour. Since latent heat is necessary to vaporise the water, unless sufficient heat is supplied, a fall in temperature will take place ; this will both yield heat and will also bring about a more rapid approach to a saturated atmosphere.

2, Heating and Cooling : if air, saturated or not, is heated, its relative humidity will decrease, but its specific humidity will not change ; if the air is cooled, the relative humidity will increase until it becomes saturated, the heat abstracted lowering the temperature of both dry air and vapour until the saturation point is passed, beyond which it will also include the latent heat liberated in the condensation of the vapour. If the condensate is removed in a separator or de-humidifier, and the air is re-heated, it will have a lower moisture content, or will have become "dried."

3, Mixing: If two masses of air at the same pressure but of different relative humidities are mixed, without gain or loss of heat externally, the total heat will not change and the relative humidity of the mixture will depend on the initial specific humidities of the respective masses.

In a building, heat may be lost or gained externally by heat transmitted through walls, windows, ceilings and floors ; heat may be received or emitted by persons ; heat may be given off by machines, lights, or by materials under treatment ; the content of water vapour may be increased by the presence of people or by industrial operations. Air conditioning thus demands, from the necessary apparatus, re-circulation, both of the air in the room and of the necessary fresh air supplied, and the heating or cooling, drying or humidifying of this air, in order to retain all the air in the room at any time at the desired temperature and humidity. The quantity of fresh air to be supplied must be sufficient to give good air to breathe and to prevent the possibility of offensive odours. Broadly speaking, restaurants require no re-circulation but only fresh air ; in theatres and public buildings, one-fourth of the air delivered by the apparatus should be fresh ; for office buildings the proportion may be from one-fifth to one-ninth, depending on the number of the occupants, etc.

Calculations relating to air conditioning may conveniently be made with the aid of Psychrometric Charts. A psychrometer is merely a wet and dry thermometer, the derivation of this word showing it to be concerned with the measurement of cold ; it is an easy step for this to be related to human comfort. We will, however, consider an actual installation from first principles. This is the office building of the Ohio Power Company at

Coshocton, Ohio.* The heat pump in this case does not use air as the working substance but is of the vapour-compression type, similar to those considered in the earlier lectures.

The building has two storeys and a basement. It is a reinforced concrete structure with a total volume of 170,000 cu. ft. (4,820 cu. m.), of which 115,000 cu. ft. (3,260 cu. m.) are included in the two storeys. Exterior walls have 1 in. thickness of insulation and the roof 2 in. of insulation. The windows are double and do not open. The design is estimated to involve, in winter, with a temperature indoors of 72°F. (22·2°C.) and an outdoor temperature of − 5°F. (− 20·6°C.) a heat loss of 409,700 B.Th.U. (103,400 kcal.) per hr., and in summer, with a temperature indoors of 78°F. (25·6°C.) and 50 per cent relative humidity, and outdoor temperatures of 95°F. (35°C.) dry bulb and 75°F. (23·9°C.) wet bulb, a gain of heat of 404,700 B.Th.U. (102,200 kcal.) per hr. The fresh air entering the building is 150,000 cu. ft. (4,250 cu. m.) per hr., while a total volume of 660,000 cu. ft. (18,700 cu. m.) per hr. is circulated and treated in the air-conditioning plant.

In Fig. III, 7, the diagrams (*a*) and (*b*) give the circuits arranged for heating and cooling, respectively. In both diagrams, the broken lines show the circuit passed through by the working substance, Freon 12, of the heat pump, while the firm lines show the circuits of the two heat-conveying media, both water in this installation. In (*a*), the medium conveying heat to the evaporator is water from a deep well ; the water is pumped from the well and delivered to the evaporator by the Pump P_1, and, after giving up its heat there, is passed to waste to a second well 60 ft. distant from the supply well. The medium receiving heat in the condenser is circulated by pump P_2 around a circuit which includes the conditioner, in which the heat received in the condenser is transferred to the air under treatment. C is the compressor, driven by

* P. Sporn and E. R. Ambrose. Transactions A.S.H.V.E., Vol. 50, 1944.

the motor M, and X is a heat exchanger, in which heat is given, by the working substance passing from the condenser to the regulator, to the working substance on its way from the evaporator to the compressor. In (*b*), the

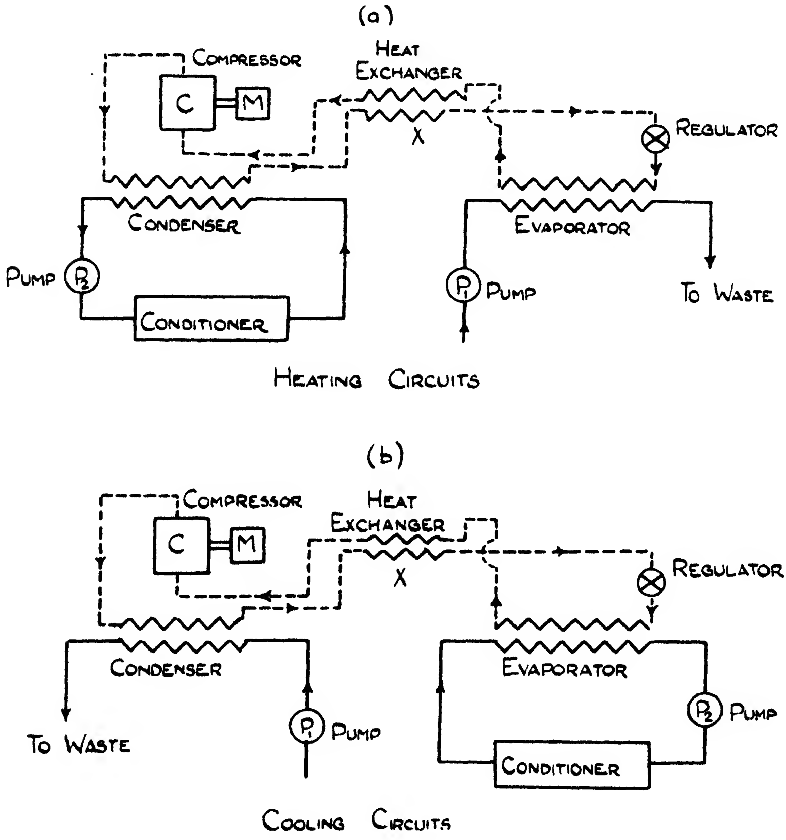

FIG. III—7.

Heating and Cooling Circuits for an Office Building using Heat Pumps Combined with Air Conditioning.

heat pump circuits are unchanged but the heat is now delivered to the evaporator by the circuit which includes the pump P_2 and the conditioner, while the condenser is cooled by the well water delivered by the pump P_1.

In this arrangement, heat is extracted from the building by the conditioner and transferred to the working substance from which it is delivered to the well water.

It is interesting to calculate the external load on the installation under extreme winter and summer conditions, respectively. This external load consists, in winter, of the heat necessary to replace the heat transfer loss, together with the heat necessary to change the conditions of temperature and humidity of the fresh air supplied from those outside to those prevailing inside the building. The quantity of fresh air supplied is equivalent at − 5°F. (− 20·6°C.) and at normal atmospheric pressure to 0·088, 150,000 = 13,200 lb. (6,000 kg.) per hr. At − 5°F. (20·6°C.) it is seen, from the extended Air and Steam Tables, given in skeleton form in Table III, 2, that saturated air contains 0·0006 lb. of water vapour per lb. of dry air.

TABLE III, 2

Properties of Air and Water Vapour.

Temperature		Pressure		Specific Humidity of Saturated Air $\times 10^{-2}$	Total Heat of Dry Saturated Steam relative to water at 32°F. (0°C.)	
°F.	°C.	lb./in.²	kg./cm.²	lb./lb. or kg./kg.	B.Th.U./lb.	kcal./kg.
95	35	0·815	0·057	3·66	1103·1	612·3
78	25·6	0·476	0·033	2·09	1095·8	608·2
75	23·9	0·430	0·030	1·88	1094·5	607·5
72	22·2	0·390	0·027	1·70	1093·2	606·8
58	14·5	0·239	0·017	1·03	1087·1	604·0
39	3·9	0·117	0·008	0·50	1078·9	599·4
− 5	− 20·6	0·014	0·001	0·06	1059·6	588·7

The actual condition is not given, but in any case the error involved in neglecting the water vapour at this temperature is not large. At 72°F. (22·2°C.) dry saturated, the water vapour present is 0·017 lb. per lb. of dry air and the saturation pressure is 0·39 lb. per sq. in. (0·027 kg. per sq. cm.). With the specified relative humidity of

30 per cent for the air in the building, the saturation pressure at the dew point is thus : $\frac{30}{100} \cdot 0{\cdot}39 = 0{\cdot}117$ lb. per sq. in. (0·008 kg. per sq. cm.), which corresponds to a wet bulb temperature of 39°F. (3·9°C.) and 0·005 lb. of water vapour per lb. of dry air.

The total heat of the mixture, above 32°F. (0°C.) is

$$H_a + wH_s = 0{\cdot}24\,(72 - 32) + 0.005\,[1078{\cdot}9 + 0{\cdot}5\,(72 - 39)]$$
$$= 9{\cdot}6 + 5{\cdot}5 = 15{\cdot}1 \text{ B.Th.U. } (8{\cdot}4 \text{ kcal.}),$$

the specific heat of superheated steam being taken as 0·5. The sensible heat for the rise in temperature of the air from its original assumed condition, dry at − 5°F., to 32°F. (− 20·6°C. to 0°C.), namely, $0{\cdot}24\,[32 - (-5)] = 0{\cdot}24 \times 37 = 8{\cdot}9$ B.Th.U. per lb. (4·9 kcal. per kg.), the heat to be added is $15{\cdot}1 + 8{\cdot}9 = 24{\cdot}0$ B.Th.U. per lb. of dry air with its associated water vapour, 0·005 lb. (13·3 kcal. per kg. of dry air with 0·005 kg. of water vapour). The external heat load relative to the fresh air is thus $13{,}200 \times 24{\cdot}0 = 316{,}800$ B.Th.U. (80,000 kcal.) per hr. The total external heat load is thus $409{,}700 + 316{,}800 = 726{,}500$ B.Th.U. (183,600 kcal.) per hr. The actual heat load on the air conditioning plant will be less than this by the heat generated inside the building by persons, lights, lifts, etc. The water necessary to humidify the fresh air would be $13{,}200 \times 0{\cdot}005 = 66$ lb. (30 kg.), but here again the actual humidification necessary in the conditioner will be affected by the moisture in the air entering for treatment from the building.

In tests upon this installation, the following are some of the data recorded : the outside air temperature was 35°F. (2°C.) and the conditioned air was delivered to the building at 94·3°F. (34·6°C.) ; the quantities of fresh and conditioned air were as given above, namely, 13,200 and 47,000 lb. (6,000 and 21,400 kg.) per hr., respectively ; the heat output was 366,600 B.Th.U. (92,600 kcal.) per hr. ; the power consumption comprised that of the compressors, 27·5 kW, and of the auxiliaries, not including the conditioner fan, 5·2 kW. Based on the input to the compressor motors, the Performance Energy Ratio = 3·9 ; taking the total input, 27·5 + 5·2 kW, the Performance Energy Ratio = 3·3. The heat derived from pumping losses and from losses in the electrical machines helps, of course, to heat the building.

Considering the external cooling load : this consists of the heat transfer load, estimated at 404,700 B.Th.U. (102,200 kcal.) per hr., together with the heat which must be removed from the fresh air supplied in order to bring it to the desired indoor conditions. The fresh air is taken as supplied at 95°F. (35°C.) dry bulb, 75°F. (23·9°C.) wet bulb, and, in the building, must be changed to conditions in which its temperature is 78°F. (25·6°C.) dry bulb and its relative humidity is 50 per cent. From the table, it is seen that saturated air at the wet bulb temperature, 75°F. (23·9°C.) contains a weight of water vapour = 0·0188 lb. per lb. (0·0188 kg. per kg.) of dry air, and this is the actual weight of vapour at 95°F. (35°C.). The corresponding weight at 95°F. (35°C.) when saturated is 0·0366 lb. per lb. (0·0366 kg. per kg.) of dry air. The saturation pressures at 75°F. (23·9°C.) and 95°F. (35°C.) are, respectively, 0·430 and 0·815 lb. per sq. in. (0·195 and 0·371 kg. per sq. cm.) ; so that the relative humidity is $\frac{0{\cdot}430}{0{\cdot}815}$ = 53 per cent. Under the conditions in the building, the saturation pressure at 78°F. (25·6°C.) is 0·476 lb. per sq. in. (0·0335 kg. per sq. cm.) ; so that, with a relative humidity of 50 per cent, the actual pressure is 0·238 lb. per sq. in. (0·0167 kg. per sq. cm.) which corresponds to a wet bulb temperature of 58°F. (14·5°C.). The weight of water vapour for saturation at this temperature is 0·0103 lb. per lb. (0·0103 kg. per kg.) of dry air, and this is the value for the indoor conditions. Thus (0·0188 − 0·0103) = 0·0085 lb. of water vapour per lb. (0·0085 kg. per kg.) of dry air supplied from outside must be removed by drying in the conditioner.

The total volume of fresh air supplied is, as before, 150,000 cu. ft. (4,250 cu. m.) per hr. but, because of the new temperature, and, assuming normal atmospheric pressure, this corresponds to :

$$\frac{P_a V}{R.T} = \frac{(14{\cdot}7 - 0{\cdot}43)\ 144.150{,}000}{53{\cdot}2.555}$$
$$= 10{,}530 \text{ lb. } (4{,}795 \text{ kg.}) \text{ per hr.}$$

The external cooling load due to the dry air supplied is thus : 10,530 × 0·24 (95 − 78) = 43,000 B.Th.U. (10,860 kcal.) per hr. With each 1 lb. of dry air at 95°F. (35°C.) is associated 0·0188 lb. of water vapour, of which the Total Heat = 0·0188. 1094·5 = 20·6 B.Th.U. (or 0·0188 kg. of water vapour, per kg. of dry air, with a total heat of 11·43 kcal.). Similarly, after de-humidifying, with each 1 lb. of dry air in the building at 78°F. (25·6°C.) is associated 0·0103 lb. of water vapour, of which the Total Heat = 0·0103 × 1087·1 = 11·2 B.Th.U. (or, 0·0103 kg. of water vapour, per kg. of dry air, with a total heat of 6·2 kcal.). The heat to be abstracted by cooling and de-humidifying is thus : 20·6 − 11·2 = 9·4 B.Th.U. per lb. of dry air (or, 11·2 − 6·2 = 5·0 kcal. per kg. of dry air). The cooling load on this account is thus, 10,530 × 9·4 = 99,000 B.Th.U. (or, 4,795 × 5·0 = 23,975 kcal.). This is the net result of the combined cooling and de-humidifying. Actually, the air would be cooled well below the dew point in the building, 58°F. (14·5°C.), passed through a separator to remove the liquid, and then re-heated to 78°F. (25·6°C.).

The total external cooling load is thus : (404,700 + 43,000 + 99,000) = 546,700 B.Th.U. per hr. (or, 102,200 + 10,860 + 23,975 = 137,035 kcal.). This value may be compared with that for the extreme heating load found above : 726,500 B.Th.U. (183,500 kcal.) per hr. In the case of the heating load, however, this value was reduced by the heat generated by persons, lights, lifts, etc. ; for cooling, on the other hand, the heat generated in the building will have to be added to the external cooling load and will thus increase the load on the air-conditioning system.

It is appropriate that this example should have been taken from American practice since all the installations in that country recently reported on by Professor E. B. Penrod* have been concerned with human comfort. It should again be pointed out that installations which provide both space-heating and space-cooling have the great advantage, over those for space-heating only, of a better load factor. The data in Table III, 3, in which load

* Paper No. 47—SA—10, *Mechanical Engineering*, Vol. 69, No. 8, August, 1947, and *Engineering Experimental Station Bulletin*, University of Kentucky, Vol. I, No. 4, June, 1947.

TABLE III, 3

Heat Pump Air-Conditioning Plants in District Offices of Southern California. Edison Company Limited.

District Office and Year of Installing	Whittier, 1937	San Bernardino, 1937	Montebello 1938	Santa Ana, 1940
Volume air-conditioned, cu. ft.	75,300	73,800	50,100	116,700
Floors	1	1 plus Mezzanine	1	2
Approx. installation costs, dols.	5,500	5,400	6,500	8,900
Average annual values :				
Heating, kWh.	7,996	11,440	8,556	21,800
Cooling, kWh.	16,100	27,460	17,111	30,526
Total air-conditioning, kWh.	24,096	38,900	25,667	52,326
Load factor, heating, per cent	7·94	9·42	6·75	11·35
Load factor, cooling, per cent	15·88	22·63	13·5	15·95
Load factor, total, per cent	23·82	32·05	20·25	27·30
kWh. per cu. ft.	0·318	0·527	0·513	0·499
Cost of air-conditioning, dols.	531	672	530	901

factor is defined as $\frac{\text{Average annual power consumption}}{\text{Maximum annual power consumption}}$, bring out this point and also give some interesting information concerning costs. The excess of cooling load over heating load is not due entirely to the difference between the heat transferred to the building in hot weather and that lost in cold weather ; as explained above, the heating load is reduced by the heat generated by people, lights, and apparatus in the building, whereas the cooling load is increased. It should also be mentioned that the buildings, being only one- or two-storied, are not very economical as regards their external heating and cooling loads.

An interesting example of the application of a heat pump, as a part of an air-conditioning installation, to industrial purposes is provided by that at the Landquart Paper Mills, at Landquart, Switzerland. The installation was designed and built by Messrs. Brown Boveri and Company, Baden, Switzerland, and embodies the Lèbre design of heat pump which uses air as the working substance. Information concerning the details of the Lèbre design has not yet been published for a full description to be given here, but Fig. III, 8, shows the installation in diagrammatic form. In the ventilation system of a paper works there are three necessary operations : (1) the supply of fresh air to the machine shops ; (2) the supply of air at reasonably high temperature and low humidity to the drying felts ; (3) the extraction or withdrawal of moist heated air from above the paper machines. The first, except in hot weather, requires heat ; the second requires considerable heat to effect the drying operation, while in the third the air receives considerable heat, sensible and latent. In the system shown, the surplus heat in (3) is conveyed, both through direct heat exchange, and with the aid of the heat pump, to (1) and (2).

Fresh air enters at A, at temperatures ranging, with the seasons, from 32°F. to 68°F. (0°C. to 20°C.), the quantity supplied normally being 116,600 lb. (53,000 kg.) per hr.

This air first receives heat in the heat exchanger X_1, its temperature being raised, if necessary, to about 68°F. (20°C.) and then continues to flow, but in two directions. Of the total quantity, about 33,000 lb. (15,000 kg.) per hr. is induced by the fan F_1, to be seen at the top of the diagram, and passed through a heater, which is used, if necessary, to adjust the temperature to 68°F. (20°C.), to the machine shops at B. The destination of the remaining air will be considered later.

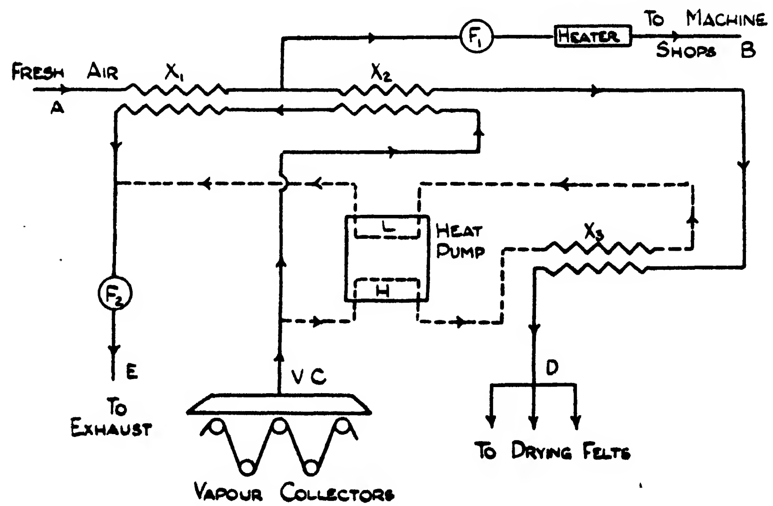

FIG. III—8.
Diagram of Heat Pump Installation at a Paper Mills.

VC, at the bottom of the diagram, indicates the collector of vapour and moist air from the paper machines. This moist air is withdrawn at 104°F. (40°C.) and of relative humidity 70 per cent, and, depending on the conditions, may range in quantity from 57,200 to 116,600 lb. (26,000 to 53,000 kg.) per hr. A constant quantity of 24,200 lb. (11,000 kg.) per hr. of this vapour is drawn into H, the high temperature side of the heat pump, in which its temperature is raised to about 140°F. (60°C.), and afterwards passes to the heat exchanger X_3, in which it gives up heat and its temperature is lowered. From X_3 it passes to L, the low temperature side of the heat pump, in which it gives up further heat, and is withdrawn by the fan F_2 and exhausted to the atmosphere at E.

The remainder of the moist air withdrawn at VC, in quantity which ranges from 33,000 to 92,400 lb. (15,000 to 42,000 kg.) per hr., is extracted by the action of the fan F_2 and passes, by way of the two heat exchangers, first through X_2 and then through X_1, and, joining the fluid mentioned above flowing from L, is exhausted by F_2 to the atmosphere.

It will be recalled that only a part of the fresh air which passed through X_1, where it received heat, was delivered to the Machine Shops. The remainder, 83,600 lb. (38,000 kg.) per hr., continues through the second heat exchanger X_2, in which it receives sufficient heat to raise its temperature to the level 83°F. to 86°F. (28°C. to 30°C.), and then passes to the heat exchanger X_3, in which it receives further heat, to raise its temperature to 104°F. (40°C.) and more, and thence to the felts being dried at D.

The primary source of heat is the warm moist air at 104°F. (40°C.), containing considerable latent heat, from the paper machine, but, since the essential need of the installation is to dry the felts, this air, with its relative humidity of 70 per cent, would be ineffective. By means of the heat pump, which causes low temperature heat at L to be transformed into higher temperature heat at H, together with the heat exchangers, the heat is transferred to fresh dry air, which is delivered at the same or at a higher temperature. Under the worst conceivable conditions, that is, with the fresh air fully saturated at its highest entering temperature, 68°F. (20°C.), the relative humidity of this air, after being heated for drying the felts, would be only 32 per cent.

The makers claim that, with a designed heat output of 455,000 B.Th.U. (115,000 kcal.) per hr., a performance energy ratio of 2·78 was realised. Further, without taking into account the pre-heating of the fresh air in X_1 and X_2, about 180 tons of coal are saved annually. The higher temperature of the drying air brings the additional advantage that the life of the felt strips is lengthened.

The Swiss, as was explained in the first lecture, were compelled during the recent war to economise to the extreme in fuel and electrical power, and they have thus given considerable attention to their industrial installations. The climatic and other conditions in the United States, in comparison, have favoured the development of heat pumps for air-conditioning. In Great Britain, although the home climate is temperate, many of our connections overseas are in regions in which the extension of air-conditioning, with or without the aid of heat pumps, would greatly improve everyday life. On the other hand, our need at home to economise fuel and power needs no emphasis, and a close examination of all industrial installation, large and small, in relation to the possibility of improvements to thermal efficiency by the application of heat pumps, would well repay those who take the trouble.

LECTURE IV

In this, the last of the lectures, we shall begin by considering thermal compressors in some detail and shall study two industrial installations embodying these, one fairly simple and the other more complex. Following this, we shall investigate the question of hot water heating derived from the waste heat from internal combustion engines, especially when this heat is combined with further heat delivered by heat pumps. An example is then considered in which the quantity of fuel consumed in heating an existing building is compared with the estimated consumptions using heat pumps driven electrically and by oil engines. The next section will be concerned with the method of operation of thermo-compressors, or thermal compressors of the jet type, in which the compression of vapours of low density is effected by convergent-divergent nozzles similar to those of the air ejectors used in the condensers of steam power plants. Lastly a description will be given of the Philips hot air engine which works on the Stirling cycle and, which, when run reversed, forms a heat pump, or a refrigerator, of considerable promise.

In the last section of my first lecture I explained the principles upon which thermal compressors operate, and used, as illustration, the simple installation for the distillation of water shown in Fig. I, 9 and 10, and for which the temperature-entropy diagram was given in Fig. I, 11.

Further scope for applications of thermal compressors will be found in the chemical industry, for the manufacture of pharmaceutical products, for concentrating dyes and for evaporating solutions and mixtures of two or more substances; the foodstuffs industry similarly needs evaporation plants for concentrating fruit juices, for producing condensed or powdered milk and in the preparation of sugar and salt. Such industrial applications, where they are practicable, usually work independently of the seasons and have, therefore, the great advantage of a high load factor; in many cases, too, the ranges of temperature involved are small and are thus favourable to the attainment of high performance energy ratios.

The range of temperature is determined, firstly, by the lower temperature, namely, that chosen for the evaporation of the substance to be treated, and, secondly, by the upper temperature to which the vapour, driven off in the evaporation process, must be raised by increase of pressure in the compressor in order that the heat transfer, from the high pressure and high temperature vapour to the substance being evaporated, may be satisfactory under the conditions prevailing in the installation.

The orders of the pressures, and the corresponding saturation temperatures involved, will naturally depend on the substances to be treated. Where possible, evaporation should take place at atmospheric pressure, so as to avoid the provision of special pressure-sealing devices for the glands of the compressor and of a special pump to evacuate the condensate and to discharge it to the atmosphere. Since the Performance Energy Ratio depends closely on the ratio $\frac{T_1}{T_1 - T_2}$, this ratio, given the same range of temperature $(T_1 - T_2)$, is higher, the higher the levels of the operating temperatures. There are, however, cases in which it is essential to carry out concentration or evaporation at low temperatures, although the saturation pressures may be much lower than atmospheric; this applies particularly to foodstuffs, such as milk and fruit juices, in which high temperatures would affect their vitamin content.

In a thermal compressor installation, the energy supplied by the compressor must be sufficient, not only to meet the direct demands of the evaporation, but also to cover all heat quantities passing, in any way, from the installations, whether as parts of the processes or as losses, thermal or mechanical. With efficient heat transfer in the evaporation process, the difference in the saturation temperatures of the high pressure steam and of the boiling solution would determine the necessary pressure ratio of the compressor. To this, however, an allowance must be added to take account of heat losses, unavoidable leakages, the formation of deposits on the heat transfer surfaces and the lag of the boiling point of the solution. This lag is the difference between the boiling point of the solution and that of pure water and tends to increase as the concentration proceeds. Incrustation of the surfaces can be reduced by suitable separators, dryers and washers. The possible corrosion of the materials of the compressor must also be taken into account. There is little more to be said in general terms on this subject, except that each individual application will naturally have its own practical design problems.

Fig. IV, 1, shows a view of a thermal compressor installed by Messrs. Brown Boveri in a foodstuffs factory in Switzerland. It has an evaporative capacity of 6,600 lb. (3,000 kg.) of water per hr. A is the evaporator, from which the vapour is drawn off, by way of the separator D, into the compressor C, which is driven through gearing by the motor B. This vapour is compressed and delivered to A, to the heating tubes, through the walls of which its latent heat is transferred to the solution undergoing evaporation. Inspection windows, as at E, facilitate observation of the evaporation processes. The installation is used, according to the seasons, for concentrating milk products and unfermented fruit juices, both of which require low temperatures, and thus evaporation pressures are well below atmospheric. Up to 6,700 gals. of fruit juices may be treated daily. It is claimed that the consumption of electrical energy with this installation is only one-ninth of that necessary with direct electrical heating.

Fig. IV—1.
Brown Boveri Thermal Compressor Installation for a Foodstuffs Factory.

A second and more complex installation is one used for the production of salt, and was installed for the United Swiss Rhine Salt Works Company, at Ryburg, Switzerland, by the Escher Wyss Company. This is shown, in diagrammatic form, in Fig. IV, 2. In this diagram, the brine purifiers, 1, receive the brine from the drilling tower, 2. From the purifiers the clear brine solution is pumped into the compensation tank, 3, from which it is drawn into the pre-heaters, 4, after which it passes to the evaporators, 5, of which there are six, in which most of the water is evaporated, leaving the salt crystals. The vapour driven off is led to the wash tower or separator, 7, to remove the moisture and brine in suspension, and thence to the suction side of the turbo compressors, 8, two in number, in which its pressure and temperature are raised to the desired levels. This high pressure steam is led back to the evaporators, 5, from which, after giving

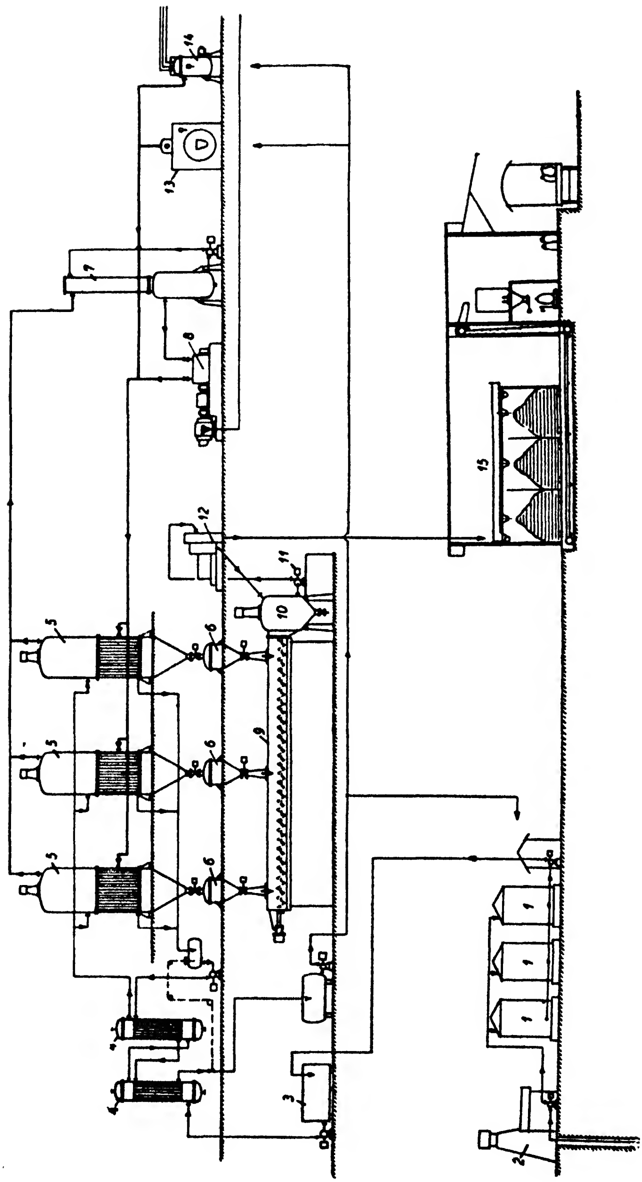

FIG. IV—2.
Escher Wyss Thermal Compressor Installation for a Salt-Works.

up its latent heat and being condensed, it is led, as condensate, to the heating side of the pre-heaters, 4. For supplying the heat necessary in starting the operation of the plant, and, when the plant is in operation, the small quantity of heat required to retain a balance, two small coal-fired boilers, 13, are provided for winter use, while the small electrical boiler, 14, is used in summer. The condensate, after giving up its heat in the pre-heaters, 4, is either returned as feed water to one or other of these boilers or flows to waste.

The salt crystals remaining after the evaporation has taken place accumulate in the conical bottoms of the evaporators, 5, from which they pass as salt pulp to the hoppers, 6, and thence to the conveyor trough, 9. Here a conveyor screw transfers the pulp to a mixer, 10, in which a stirrer, in the form of a propeller, maintains a mixture of brine and salt crystals capable of being delivered by the pump, 11, to the centrifugal separator, 12, which serves as a water extractor. The water is returned to the mixer, 10, and the separated salt crystals are delivered to the salt bunkers, 15, and are now ready for dispatch.

The output of this installation is 40,000 tons of salt per annum. Before conversion to operation with thermal compressors, the installation used the now obsolescent method of direct-heating for the evaporator pans. In 1941*, when the order was given for the conversion, a triple-effect vacuum plant was also under consideration and, with the price of coal then ruling, 85 Swiss francs per ton, was expected to give competitive results with those from thermal compressors. But, whereas in the former the saving in coal expected was 66 per cent, with the latter a saving of 90 per cent was thought reasonable. The plant was put into operation in 1942 and in the first year of working showed a saving of 14,000 tons of coal. Each kilowatt-hour taken by the motors of the thermal compressors led to a saving of 49,400 B.Th.U. (12,500 kcal.)

* Bulletin, *Association Suisse des Electriciens*, No. 16, 1943.

per hr. Using this value as a basis, the Performance Energy Ratio of the thermal compressors is $\frac{49400}{3412}$ or $\frac{12500}{860} = 14{\cdot}5$. Encouraged by the results with this installation, the owners have ordered a similar installation for their works at Schweizerhalle.

This brief study of these two industrial applications of thermal compressors will suffice to indicate the possibilities of such equipment in relation to economy of fuel and power.

We will next consider the very interesting case of the oil engine in combination with space heating by hot water. Such a combination has been made in simple form in certain applications in Denmark*, in which waste heat in the engine cooling water and in the exhaust gases has been applied to hot water systems. In one small installation, an oil engine was used primarily for the generation of electricity and a brake thermal efficiency of 31 per cent was realised ; a further 46 per cent of the heat in the fuel was recovered from the cooling water and exhaust gases and used for heating, making the overall efficiency 77 per cent. The power developed was 1,060 b.h.p. (1,076 metric h.p.) and approximately 40 therms, or 4,000,000 B.Th.U. (1,010,000 kcal.) per hr. were delivered to the heating systems. This is five times the maximum rate of heat delivery in the Norwich installation, which provided heat for a building the volume of which was about 500,000 cu. ft. (14,160 cu. m.). Incidentally, the volume of a modern family flat would hardly exceed 8,000 cu. ft. (214 cu. m.).

Suppose that, instead of being used to generate electricity, the whole of the useful engine output of 1,060 b.h.p. (1,076 metric h.p.) had been applied to drive a heat pump under conditions similar to those at Norwich. Taking the same overall performance energy

* A. K. Bak and N. C. Geertsen : Fuel Economy Conference of the World Power Conference. The Hague, 1947, Section C5, Paper No. 4.

ratio of $3{\cdot}5 : 1$, the heat delivered by the heat pump would be :

$\frac{3{\cdot}5}{1} . \frac{31}{46} . 4{\cdot}0 \times 10^6$ B.Th.U. $\left(\frac{3{\cdot}5}{1} . \frac{31}{46}\, 1{\cdot}01 \times 10^6 \text{ kcal.}\right)$ per hr. But the waste heat from the engine already provides $40{\cdot}10^5$ B.Th.U. ($10{\cdot}1.\ 10^5$. kcal.) per hr. ; so that the total heat available for heating would be :

$$\left(1 + \frac{3{\cdot}5}{1} . \frac{31}{46}\right)(4{\cdot}0 \times 10^6)$$

$= 3{\cdot}36\,(4{\cdot}0 \times 10^6)$ B.Th.U. ($3{\cdot}36\; 1{\cdot}01 \times 10^6$ kcal.) per hr. The fuel supplied to the engine has a heat equivalent of:

$$\frac{100}{46} .\ 4{\cdot}0 \times 10^6$$

$= 2{\cdot}17 .\ 4{\cdot}0 \times 10^6$ B.Th.U. ($2{\cdot}17 .\ 1{\cdot}01 \times 10^6$ kcal.) per hr. The overall ratio of the heat delivered to the heat supplied in the fuel is thus $\frac{3{\cdot}36}{2{\cdot}17} = 1{\cdot}55$. An oil-fired boiler used directly for heating the water could hardly be expected to have an efficiency greater than 75 per cent : so that, the combination of engine and heat pump is, in relation to fuel quantity, more than twice as efficient. To complete such a comparison, the quality and cost of the fuels for the engine and boiler, and the respective capital and maintenance costs would, of course, also require consideration.

There are cases in which oil engines are running at less than their rated outputs, so that further power is available, or in which, by adding a supercharger to an unsupercharged engine, it would be possible to produce further power. Under these conditions, the addition of a heat pump driven by this extra power can be very attractive. For example, in the installation considered, assuming the same brake thermal efficiency, an increase in the output from 1,060 to 1,315 b.h.p. (1,076 to 1,335 metric h.p.), that is, by 28·6 per cent, would make it possible, while generating the same output of electrical power, to double the heat available for the hot water system.

If P is the basic brake horse power and p the surplus horse power available to drive a heat pump, both expressed in heat units, H is the rate with which the waste heat from the engine is utilised when developing its basic power and h is the rate at which heat is delivered by the heat pump, a simple expression may be derived to show the relationship between these quantities. Using the same values for brake thermal efficiency and for the percentage of waste heat utilised for heating, 31 and 46 per cent, respectively, and the same value, 3·5 : 1, for the performance energy ratio of the heat pump, then $H = \frac{46}{31}$ P as before, but the extra heat h due to the additional power p $= \frac{46}{31} p + \frac{3\cdot5}{1} p = 1\cdot482\ p + 3\cdot5\ p$ $= 4\cdot982$ p.

The total heat delivered is thus given by :

$H + h = 1\cdot482\ P + 4\cdot982\ p$, and $h = 4\cdot982\ p$.

Or, if the percentage of heat utilised can be raised slightly from 46 to 46·5 per cent, this gives the rounded values :.

$H + h = 1\cdot5\ P + 5\ p$; $h = 5p$, a handsome return in heat for the extra power p.

If it is not possible for practical reasons to obtain power additional to that normally required for the primary purpose, such as the generation of electricity or for pumping, it may be possible, by providing suitable capacity for heat storage, to drive the heat pump during off-peak periods of power generation and to store the heat in heat accumulators. Such a combination would lead both to an improvement of the load factor of the engine and to a utilisation of available low grade heat.

In space-heating installations in Great Britain, simplicity and reliability have been the main considerations in the past, with fuel consumption a minor matter, and thermal efficiencies of 60 per cent at full load, and much less at reduced loads, have been accepted. With electrically-driven heat pumps, as compared with boilers, for buildings heated for long periods, Mr. Sumner's figures indicated a reduction per therm from 16·6 lb. of coal at the building being heated to 9·1 lb. at the central station. But a further point in favour of the heat pump is that, whereas the fuel consumption with banked fires in boilers can be considerable, its stand-by consumption is nil : it can be switched rapidly from no load to full load when desired. At any load, assuming an overall efficiency of electricity production and distribution of 25 per cent, a Performance Energy Ratio of 4 means that the heat, delivered where it is required, is equal to 100 per cent of that supplied in the fuel at the power station. In special applications, such as for heat pumps in industrial applications in which, owing to smaller ranges of temperature, higher values of the Performance Energy Ratio may be attained, or in heating the water of swimming baths, in which this ratio may be double that for space-heating, this relationship will be considerably higher than 100 per cent.

To illustrate this, a type of building with a relatively good load factor will be considered, namely, one which serves as a swimming baths in summer and as an assembly or dance hall in winter. An actual building of this kind in the southern counties is roughly of block form, 200 ft. × 80 ft. × 40 ft. high, giving a volume of 650,000 cu. ft. (18,100 cu. m.), a wall area of 22,400 sq. ft. (2,080 sq. m.) and roof and floor areas of 16,000 sq. ft. (1,488 sq. m.), respectively. Of the wall area, 15·2 per cent is taken by windows ; of the roof area, 15·0 per cent is taken by double skylights. The building is modern, with brick walls and a concrete roof, asphalted ; heating is by coke-fired boilers ; panel heating is installed, the supply temperature being 110°F. (43·3°C.) and the return temperature, 90°F. (32·2°C.). During the 30-week winter season, as an assembly hall, 50 tons of coke are consumed, giving an average consumption of 1·67 tons per week ; during the 22-week summer season, as a swimming baths, 40 tons are consumed, with an average consumption of 1·82 tons per week.

In winter, the building is occupied for, say, 36 hrs. per week, with a maximum load estimated at 950,000 B.Th.U. (240,000 kcal.) per hr. Allowing 30 per cent for exceptional conditions, this requires a boiler capacity of 1,250,000 B.Th.U. (316,000 kcal.) or 12·5 therms per hr. Taking the liberal value for the mean efficiency of the boiler as 50 per cent, with coke of calorific value 12,000 B.Th.U. per lb. (6,667 kcal. per kg.), the mean total load during the winter period is 133,000 B.Th.U. (33,600 kcal.) per hr. : if a lower mean value is taken for the efficiency, the estimated mean load is reduced in proportion. In summer, heat must be supplied, 1, for the heating of the water of the swimming bath when re-filled, 2, to keep the baths hall and water at 72°F. Maximum boiler capacity will be demanded for 1, and will determine the time of re-heating. The mean total load during the summer period is $133{,}000 \times \frac{1 \cdot 82}{1 \cdot 67} = 145{,}000$ B.Th.U. (36,600 kcal.) per hr.

For comparison, three cases will be examined : (*a*) as recorded with the coke-fired boilers ; (*b*) with one or more heat pumps, electrically driven from a central station, burning coal of calorific value, 13,000 B.Th.U. per lb. (7,225 kcal. per kg.) with an overall thermal efficiency of 25 per cent ; (*c*) with one or more heat pumps, driven at the building by one or more oil engines, of brake thermal efficiency, 30 per cent, burning oil of calorific value, 18,000 B.Th.U. per lb. (10,000 kcal. per kg.), 46 per cent of the heat of the fuel being recovered from the waste heat of the engines. In winter, the range of working temperature of the heat pumps is such that a Performance Energy Ratio of 3·5 is assumed ; in summer, the range is much smaller and a Performance Energy Ratio of 6 is taken. The data from calculations on this basis are set out in Table IV, 1.

TABLE IV, 1

Comparison Between Coke-Fired Boilers and Heat Pumps, Electrically and Oil-Engine Driven.

Installation	Capacity	Mean Fuel Consumption, Tons per week	Thermal Efficiency = $\frac{\text{Heat delivered}}{\text{Heat in fuel}}$	Annual Fuel Consumption
(*a*) Boilers, coke-fired	12·5 therms per hr.	(Coke)		(Coke)
Winter		1·67	50 per cent	90 tons
Summer		1·82	50 per cent	
(*b*) Heat pumps, driven by electric motors	h.p. of motors, 141	(Coal)		(Coal at Station)
P.E.R., 3·5 Winter		0·88	87 per cent	39 tons
P.E.R., 6 Summer		0·56	150 per cent	
(*c*) Heat pumps, driven by oil engines	h.p. of engines, 99	(Diesel oil)		(Diesel oil)
P.E.R., 3·5 Winter		0·36	155 per cent	17 tons
P.E.R., 6 Summer		0·27	228 per cent	

The results show the estimated saving in fuel over (*a*), at the central power station when the heat pumps are electrically-driven (*b*) and when the heat pumps are driven by oil engines (*c*). The oil-engine heat pump combination in particular, is most attractive, especially when space is a major consideration.*

In a recent paper to the Institute of Marine Engineers, Mr. W. Sampson† discussed Steam Jet Refrigeration for Marine Purposes and certain of the matters he put forward are relevant to our subject. Fig. IV, 3, reproduced from his paper, shows a steam jet refrigeration plant, the function of which is to convey low temperature heat by means of steam, as a working substance, to the ship's

* See also: S. J. Davies and F. G. Watts, *Engineering*, 17th and 24th September, 1948; F. G. Watts, *Engineering*, 10th December, 1948.

†Trans. *Institute of Marine Engineers*, Vol. LVIII, No. 11, 1946.

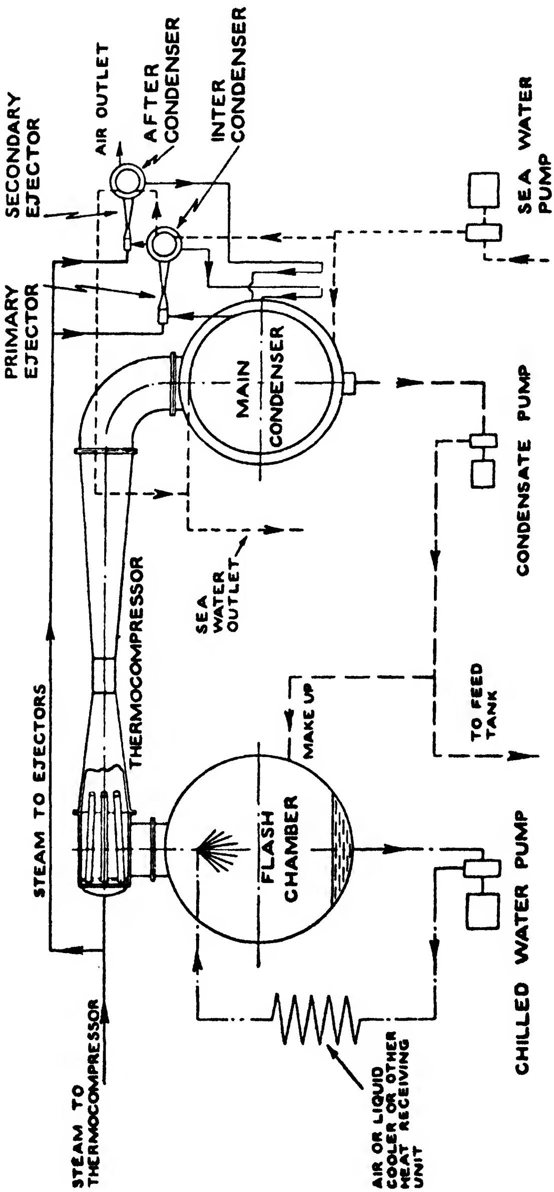

Fig. IV—3.
Diagram of Steam Jet Compressor Refrigeration Plant.

main condenser. In the condenser, this steam joins with the exhaust steam from the main engines and is condensed, so transferring the refrigerated heat, but at a higher temperature, to the circulating water of the condenser.

The regions of low and high pressure of the cycle are, respectively, the flash chamber and the condenser. The rise of pressure accompanying the transfer of the working substance from the flash chamber to the condenser is brought about by means of a thermal jet compressor, in which the motive energy is derived from a steam jet. The principle underlying this operation is, of course, the same as that employed in the steam jet air ejectors of the condensing plants of steam installations, and to be seen on the right of the figure. The return to the flash chamber of the working substance, which is in the liquid form during this part of the cycle, is by means of the condensate pump. Since the pressure after the condensate pump must be above atmospheric, this delivery involves a throttling action as the liquid enters the flash chamber.

In the figure, the steam from the jet thermal compressor combines with the main exhaust steam. This is, of course, not essential and the cycle can be carried out independently ; in that case, the condensate pump could be dispensed with, and the return to the flash chamber effected by simple throttling.

Heat is given to the working substance in the flash chamber to cause evaporation and is abstracted from it again in the condenser. The secondary heat-conveying circuit in the flash chamber is provided by the water itself ; this is drawn at its lowest temperature by a pump from the bottom of the flash chamber, passed through a heat exchanger, or heat receiving unit, in which it receives heat, and returned to the top of the chamber, into which it is discharged in the form of a spray. On shipboard, this heat may be received from air coolers for conditioning plants, from water coolers, or from liquid CO_2 coolers.

It is convenient first to consider the operations in the flash chamber. Suppose the water from the heat receiving unit, being delivered as a spray at the top, is at 55°F. (14°C.) and that an equal quantity leaves at the bottom at 45°F. (7·2°C.). The saturation pressure corresponding to 45°F. (7·2°C.) is 0·1475 lb. per sq. in. (0·0104 kg. per sq. cm.) or a vacuum of 29·7 in. The latent heat at this pressure is 1068·4 B.Th.U. per lb. (594 kcal. per kg.). For each pound of water admitted, let it be necessary to evaporate x lb. Thus :

$$\begin{aligned} (1 - x)(55 - 45) &= x[1068{\cdot}4 - (55 - 45)] \\ &= x\,1058{\cdot}4 \\ \text{Or } 10 &= 10x + 1058{\cdot}4\,x\,; \\ \text{and } x &= 0{\cdot}00938 \text{ lb.} \end{aligned}$$

So that somewhat less than 1 per cent of the water must be evaporated to cause this drop in temperature of 10°F. (5·6°C.).

The jet thermal compressor has one or more steam jets in which the actuating steam expands down to a pressure slightly below that in the flash chamber. This slight difference of pressure causes the vapour in the chamber to pass out to the jet in which it is entrained by the actuating steam. The actuating steam and its entrained vapour pass through a convergent pipe to a parallel throat, after which a divergent pipe causes a fall of velocity and a corresponding rise of pressure to a value sufficiently above that of the condenser pressure for the combined steam to enter the main condenser.

The operations in the jet thermal compressor may be followed from the sketch in Fig. IV, 4, and from the total heat-entropy diagram in Fig. IV, 5 ; the letters used correspond in the two figures. A shows the high pressure steam assumed to be dry saturated at pressure p_A. It expands in the convergent-divergent nozzle with relative efficiency, $\dfrac{H_A - H_B}{H_A - H_{B'}}$, point B on the diagram showing its condition on leaving the nozzle. At B, it has the velocity

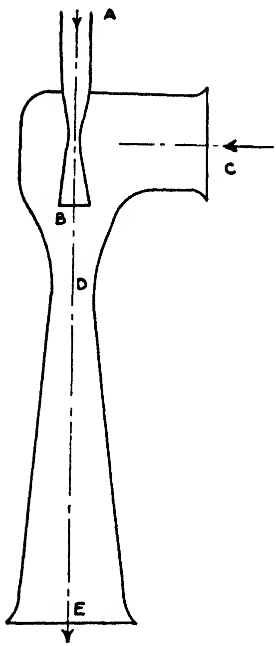

FIG. IV—4.
Diagram of Jet Thermal Compressor.

v_B, due to the effective heat drop $(H_A - H_B)$, and its dryness fraction is q_B. Steam evaporated in the flash chamber enters at C; its pressure is p_C which is equal to p_B and $p_{B'}$; its velocity is v_C, which is very small compared with v_B. This steam mixes with the steam in condition B and the point D shows the state of the mixture in the parallel throat. If W is the weight entering C per second and w is the weight of steam leaving the nozzle at B, then, by the principle of the Conservation of Momentum: $(W + w)v_D = W v_C + w v_B$; neglecting losses, v_D is thus: $\dfrac{W v_C + w v_B}{W + w}$. The specific volume of the mixture at D is obtained from the heat per lb. which, neglecting losses, is given by the relationship: $\dfrac{W H_C + w H_B}{W + w}$. Knowing this value, the dryness fraction of the mixture may be found from the total heat-entropy diagram. Compression of the mixture, from p_C to p_E in the divergent pipe, DE, is shown by the line DE on the diagram, the efficiency of compression relative to adiabatic conditions being: $\dfrac{H_{E'} - H_D}{H_E - H_D}$. Neglecting friction between steam and nozzle walls $(H_E - H_{E'})$ is the heat energy generated through the dissipation of the kinetic energy of unit mass, following the reduction of velocity in the divergent pipe, DE.

Mr. Sampson describes a case in which a refrigerating effect of **600,000** B.Th.U. (**151,500** kcal.) per hr. is brought about by **2,725** lb. (**1,239** kg.) per hr. of dry saturated supply steam at **74·7** lb. per sq. in. (**5·25** kg. per sq. cm.). The cooling medium is water at 40°F.(4·4°C.), which is the temperature at which evaporation takes place in the flash chamber. The pressure in the flash chamber corresponding to this temperature is 0·1217 lb. per sq. in. (0·0085 kg. per sq. cm.). The condenser pressure is 0·74 lb. per sq. in. (0·052 kg. per sq. cm.). The main steam supply is at 464·7 lb. per sq. in. (32·6 kg. per sq. cm.) and 750°F. (399°C.), and the steam is bled from the main supply at 74·7 lb. per sq. in. (5·25 kg. per sq. cm.) for the jet thermal compressor. The main engines developed 2,500 h.p. (2,536 metric h.p.).

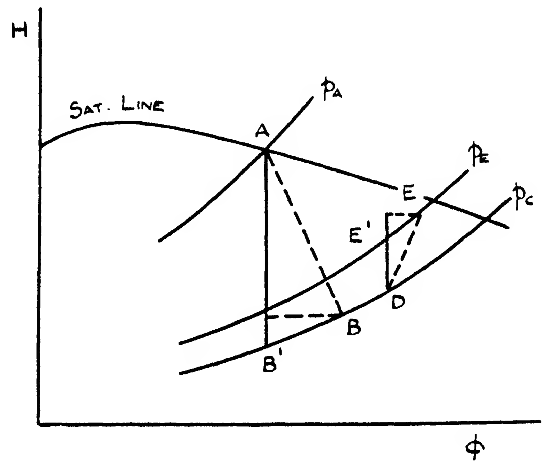

FIG. IV—5.

Total Heat-Entropy Diagram Corresponding to FIG. IV—4.

The effective heat drop of the main supply steam to condenser pressure is given as 372 B.Th.U. per lb. (206·6 kcal. per kg.) and the effective heat drop of the bled steam to the bled point is 147 B.Th.U. per lb. (81·6 kcal. per kg.). If all the steam went to the turbines, the quantity required would be :

$$\frac{2500 \times 33000 \times 60}{778 \times 372} = 17100 \text{ lb. } (7775 \text{ kg.}) \text{ per hr.}$$

With A lb. per hr. passing through and 2,725 lb. (1,240 kg.) per hr. bled, the relationship is :

$$A\ 372 + 2725 \times 147 = \frac{2500 \times 33000 \times 60}{778}\text{, which}$$

gives A = 16,000 lb. (7,270 kg.) per hr. and the total steam now raised as 16,000 + 2,725 = 18,725 lb. (8,510 kg.) per hr. Thus the extra steam raised for the refrigerator is 18,725 − 17,100 = 1,625 lb. (739 kg.) per hr. The total heat of the steam as raised is 1,388 B.Th.U. per lb. (771 kcal. per kg.). With a hot-well temperature of 80°F. (26·7°C.), the heat received in the boiler is 1,388 − 48 = 1,340 B.Th.U. per lb. (745 kcal. per kg.). With fuel of calorific value 18,500 B.Th.U. per lb. (10,280 kcal. per kg.), and a boiler efficiency of 88 per cent, this requires $\frac{1625 \times 1340}{18500 \times 0{\cdot}88}$ = 134 lb. (61 kg.) of fuel. The overall coefficient of performance is thus $\frac{600000}{134 \times 18500}$ = 0·242/1, a value which includes the combined deficiencies of boiler, engine and jet thermal compressor system. This value supplies the practical criterion.

The real coefficient of performance of the refrigerator considered separately is $\frac{600000}{2725(372 - 147)}$ = 0·98/1. If, however, such an installation were regarded as a heat pump, the performance energy ratio would be 1 + 0·98 = 1·98. This is certainly not a high value, but the installation has compensating advantages : it is simple ; it involves only one working substance, steam ; it has no mechanical parts and thus requires no lubrication ; it is noiseless and control can be arranged to be automatic. Cases can thus occur in which these compensating advantages can outweigh the drawback of a low performance energy ratio. In its characteristics, however, it is even more sensitive than the mechanical vapour-compression type of heat pump to range of temperature, and thus range of pressure, and, similarly, offers its best performance when these ranges are restricted. The system offers, however, a further means of raising the grade of low temperature heat to a useful level, and its possibilities of application in suitable cases should certainly be examined.

I referred in Lecture III to the importance of heat exchangers, an importance which has received special emphasis from the recent success of gas turbines. The efficiencies of heat exchangers in relation to their hydraulic losses, as measured by the pressure drops, their dimensions, weights and costs, form a vast subject for study, which must be left out of the present course. A recent development of engine design, however, which involves an efficient heat regenerator, is of such importance, not only on account of its inherent possibilities, in its direct form, as a power unit, and, in its reversed form, as a refrigerator and heat pump, but also as regards the regenerator as a possible method of heat exchange, that it is appropriate to study it here.

The engine in question is the Philips hot air engine,* which operates on the Stirling cycle, and which has been developed by the engineers of the Philips Company at Eindhoven in the Netherlands, and I shall conclude these lectures with a brief description of its method of operation.

The cycle upon which it operates was proposed by Robert Stirling early in the nineteenth century and it was this cycle upon which the Stirling hot air engine worked. In Fig. IV, 6, the diagrams at the left show the pressure-volume relationship in the working cylinder of this engine and those at the right, the pressure-volume and temperature-entropy diagrams of the ideal cycle. In the cylinder are two pistons or displacers which contain between them a constant mass of air. The region on the left is kept at a high temperature, T_h, and that on the right at a low temperature, T_c. R is a heat regenerator. There are four stages, I, II, III and IV, on the diagrams,

* *The Philips Technical Review*, Vol. 8, May, 1946 ; Vol. 9, April, 1947.

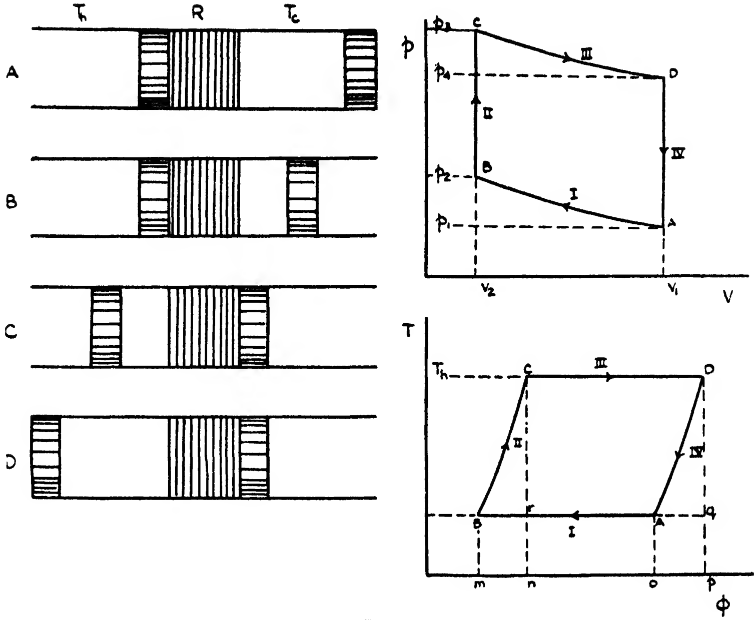

FIG. IV—6.
Stages of Philips Hot-Air Engine, working on a Stirling Cycle.

Stage I.—A–B Isothermal Compression

$$\text{W.D. on air} = RT_c \log_e \frac{V_1}{V_2} = \text{Heat Rejected}$$

Stage II.—B–C Constant Volume : Air Passed Through R

$$\text{W.D.} = 0. \quad \text{Heat Gained from R} = c_v\,(T_h - T_c)$$

Stage III.—C–D Isothermal Expansion

$$\text{W.D. by Air} = RT_h \log_e \frac{V_1}{V_2} = \text{Heat Received}$$

Stage IV.—D–A Constant Volume : Air Returned through R

$$\text{W.D.} = 0. \quad \text{Heat Lost to R} = c_v\,(T_h - T_c)$$

while diagrams A, B, C and D, at the left, show the positions of the pistons at the corresponding points of the right-hand diagrams. Between A and B, the right-hand piston moves to the left, compressing the charge, the temperature in the ideal case remaining constant ; in other words, I is an isothermal compression and work equal to the heat rejected is done *on* the air. During II, from B to C, both pistons move together, the volume between them remaining constant ; the air is passed from the cold region to the hot region through the regenerator R and receives heat at constant volume, raising its temperature from T_c to T_h. In III, from C to D, the right-hand piston is stationary and the left-hand piston moves farther to the left. The charge expands isothermally at temperature T_h and work is done *by* the air. In IV, from D to A, the charge is displaced at constant volume back through the regenerator, R, losing heat and suffering a fall of temperature from T_h to T_c.

The net work done in the cycle

$$= R\left(T_h \log_e \frac{V_1}{V_2} - T_c \log_e \frac{V_1}{V_2}\right).$$

The heat received is $RT_h \log_e \frac{V_1}{V_2}$, since the heat interchanges in stages II and IV are internal to the engine ; also, ideally, the heat received from the regenerator in II is equal to the heat given to the regenerator in IV. The efficiency is thus :

$$\frac{R \log_e \frac{V_1}{V_2} (T_h - T_c)}{R \log_e \frac{V_1}{V_2} (T_h)} = \frac{T_h - T_c}{T_h}.$$

This value is equal to that of a Carnot cycle working between the same temperatures, T_h and T_c. This may be seen on the temperature-entropy diagram, in which nrCDqp is the Carnot diagram, nCDp, the area representing the heat received at T_h, being common to both the Stirling and the Carnot diagrams. The area mBCn represents the heat received back in stage IV at constant volume from the regenerator, and is equal to the heat lost in stage II at constant volume to the regenerator. Br or mn is thus equal to Aq or op, and the heat rejected in the Stirling cycle, mBAo, is equal to that rejected in the Carnot cycle, nrqp. Clearly, the area of net work done in the Stirling cycle, ABCD, is equal to the work done in the Carnot cycle, qrCD.

Since a heat engine working on the Stirling cycle, in its ideal form, has the same efficiency as one working on the corresponding Carnot cycle, namely, the highest theoretically possible, so will a refrigerator or a heat pump working, respectively, on a reversed Stirling cycle, have the highest possible theoretical coefficient of performance or performance energy ratio, as the case may be.

If the efficiency of the regenerator is f, then additional heat, equal to $(1 - f)\, C_v\, (T_h - T_c)$, must be added to the air to complete the cycle. The efficiency of the cycle of reference is then :

$$\frac{R\,(T_h - T_c)\log_e \dfrac{V_1}{V_2}}{RT_h \log_e \dfrac{V_1}{V_2} + (1 - f)\, C_v\, (T_h - T_c)}.$$

In carrying out this cycle in practice two kinds of difference will be introduced : the first of these follows from the practical difficulty of arranging a driving mechanism to give the volume relationship of the ideal cycle ; the other follows from the practical difficulties, first, of arranging for truly isothermal expansion and compression and, second, of producing a regenerator with ideal efficiency, that is, one in which the whole of the heat given to the regenerator in IV is recovered in II.

Fig. IV, 7, shows a diagrammatic section through one of the engines already produced. The link drive is intended to produce the volume relationship as closely as possible. K represents the cooler in which heat is abstracted and H represents the Heater. R is the regenerator. The volume is enclosed between the upper piston P and the lower piston Z, above P and in S, H, R, K, and O. V_w is the space containing air at high temperature and V_k the space at low temperature. The piston

P is provided with a heat-insulating crown. S shows the passage from the Heater H into the space V_W and O shows the ports between the cooler K and the cold space V_K. The crank case Q contains air at a mean pressure which can be controlled by the pressure pump C. Passages K connect the crank case to the cylinder.

It will be noticed that the Heater, Regenerator and Cooler are placed in an annular passage surrounding the cylinder. As the ports O are uncovered by the piston it becomes possible, when the volumes enclosed between the pistons are considered, to transfer the working substance from the cold space, V_K, through the cooler, K, the regenerator R, and the heater, H, to the hot space, V_W, and vice versa.

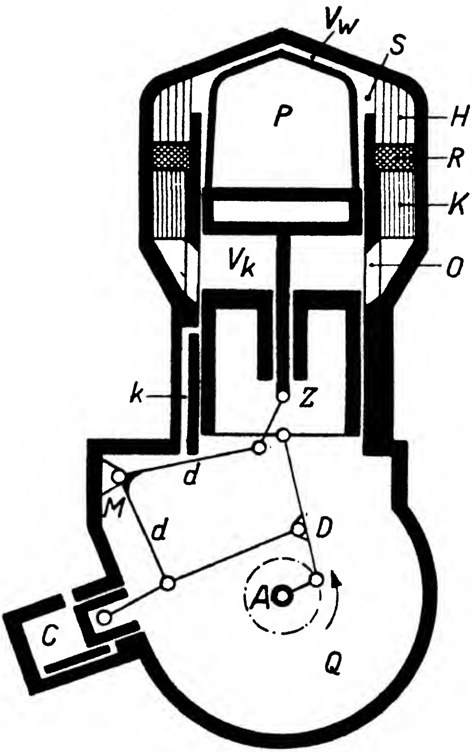

FIG. IV—7.
Diagrammatic Section through Philips Hot-Air Engine.

The heater consists of a thin walled hollow-cylindrical vessel of heat resisting material in connection with the hot space, V_W. The heater receives heat externally from a burner. The cooler is in permanent connection with the cold space through the ports in the wall of the space.

It is possible, while retaining the same orders of the pressure ratios, to raise the complete level of the pressures in the engine by the pump C which delivers air from the outside into the crank case, and in this way the power output may be controlled. This gives a range of minimum pressure of from 8–20 atmospheres. The only possibility of leakage from the enclosed engine is along the main shaft A.

The arrangement enables the cycle shown in the diagram of Fig. IV, 6, to be carried out as follows :

During stage I, the volume between the pistons is reduced and the pressure raised but piston P is practically at rest and the compression is as far as possible carried out by the movement of Z. Work is done *on* the air. In this way the air during this stage is in contact with the cooler and loses heat.

During stage II, both pistons are in motion, Z upwards and P slightly downwards, and, as a result, the air is forced upwards through the regenerator R into V_W.

During stage III, the pistons both move downwards, although the motion of Z is small, so that the air in the cooler and regenerator remains at rest. The air in V_W is in contact with H, from which it receives heat and expands, forcing the piston P downwards and thus doing positive work.

In stage IV, piston Z is practically at rest and piston P moves upwards, displacing the air in V_W through the regenerator R downwards into V_K.

Fig. IV, 8, shows an outside view of this engine. The heat exchanger, which conveys heat from the burner to the heater, is shown at the top. The pump for regulating the level of pressure may be seen at bottom left. The connections to the cooler may also be seen.

Fig. IV, 9, shows a multi-cylinder arrangement, in which the hot space of one cylinder and the cold space of the next cylinder together form a complete volume. For example, if Z_1 is at rest at the top and Z_2 moves downwards, the compression stage in cylinder 2 is carried out. Simultaneously with this, the expansion stage is

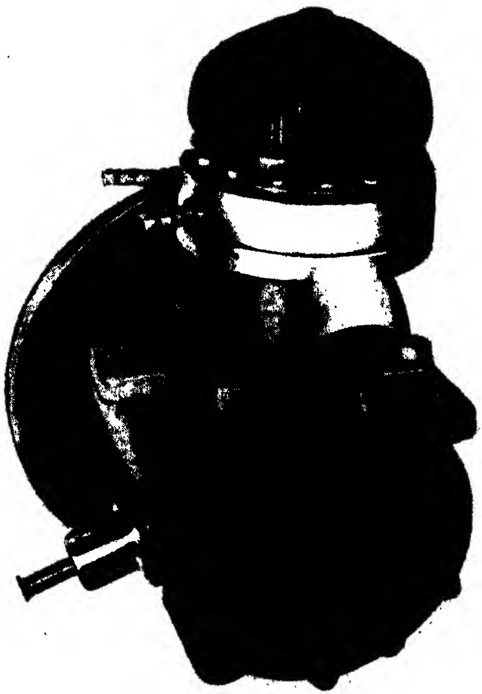

FIG. IV—8.
Outside View of the Engine of FIG. IV—7.

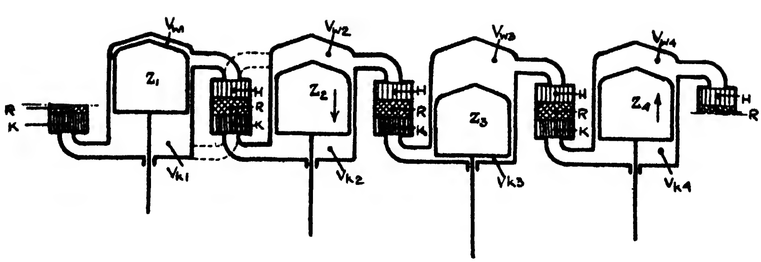

FIG. IV—9.
Diagram of Multi-Cylinder Philips Hot-Air Engine.

proceeding at the top of cylinder 2, the piston Z_3 being at the bottom of its stroke, and so on.

A comparison of the indicator diagram with one from a typical 2-stroke oil engine is given in Fig. IV, 10. It will be noticed that the pressure range in the former is much smaller and that the rate of rise of pressure very much lower, but the mean effective pressure is much greater. All this leads to higher mean effective pressures and also, as I myself can testify, to very much smoother running of the engine.

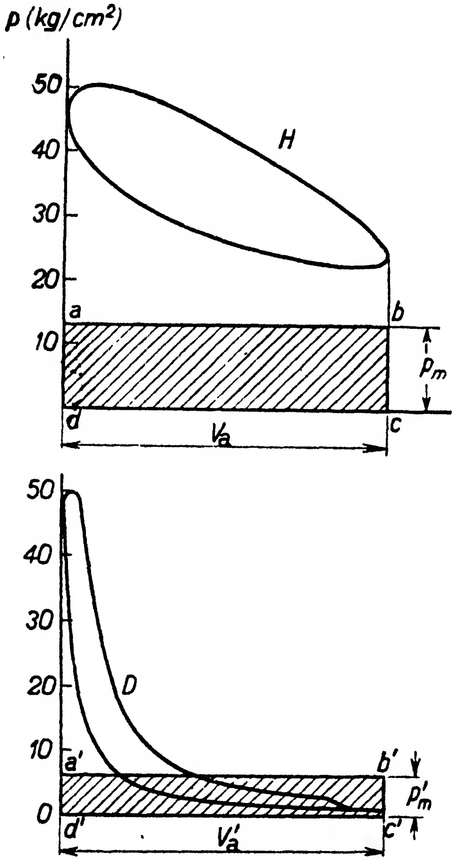

Fig. IV—10.
Indicator Diagrams from Philips Hot-Air Engine and from Oil Engine.

Fig. IV, 11 shows the regenerators of two single-cylinder engines, one for $3\frac{1}{2}$ h.p. and one for 10 h.p. The sizes can be judged from the centimetre scale. The heat is taken up and given out by fine wire of heat-resisting material, this wire being wound in such a way as to reduce, as far as possible, the resistances to the flow of the air through the regenerator.

The regenerator must allow an extremely rapid heat interchange from air to metal and from metal to air. At the same time, the pressure losses in the passages must be as low as possible and the clearance volume introduced must also be kept as small as possible. The heat capacity of the regenerator must also be sufficiently high. For high efficiency in the regenerator, the difference, at any point, between the temperature of the air and that of the regenerator, must at all times be as small as possible, in order to approach as closely as possible to ideal reversible operation.

FIG. IV—11.
Heat Regenerators for Two Philips Hot-Air Engines.

The temperature range through the regenerator can be from 212° to 1112°F. (100° to 600°C.) and vice versa, corresponding, when the air is passing, to an extremely high temperature gradient ; this change may take place within 0·01 second. Values exceeding 95 per cent, for the efficiency of the regenerator, are stated to have been realised.

The heat received by the air in the regenerator, in stage II, a quantity which determines the mean effective pressure, is about three times the quantity received from the heater, in stage III. The fact that the lowest pressure of the cycle is much greater than that of the atmosphere, and the air thus of correspondingly higher density, leads to a high development of power in relation to the size of the engine.

There are so many practical advantages in this design that engineers will look forward, with great interest, to further news of its development and application.

It has not been possible in this short course of lectures to illustrate more than a few of the installations of heat pumps and thermal compressors actually built. These installations have already become numerous and varied. In the United States, as we have seen, emphasis has been placed on the utility of the heat pump in connection with heating, cooling and air conditioning. In Switzerland, most installations derive their low temperature heat from lakes and rivers : in the United States, air and well water are mostly used as the source of the low temperature heat'ng, the smaller sets often using air. In this country, in view of the need to economise in fuel and energy, waste of low grade heat in space heating and in industrial processes, without full consideration of the possibility of its utilisation, is inexcusable. In addition to the sources of low grade heat mentioned, the sea and tidal rivers are particularly favourable. The utilisation of this heat, with the aid of heat pumps and thermal compressors, no longer involves pioneer work, but merely the application of experience already in existence. Thermal installations of all kinds should now be regarded, therefore, not only from the point of view of the high standard of reliability that has long been accepted in our engineering practice, but also with a proper practical appreciation of the relationship of heat to the temperature scale, and of the possibility of the real economies to be made.

NAME INDEX

SUBJECT INDEX