REFRIGERATION

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A DESCRIPTION OF THE PRINCIPLES
EMBODIED IN MODERN COMMERCIAL
AND DOMESTIC REFRIGERATORS

WITH PROBLEMS AND CALCULATIONS
ON HEAT TRANSFERENCE

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LONDON

SIR ISAAC PITMAN & SONS, LTD.

1947

SIR ISAAC PITMAN & SONS, LTD.
PITMAN HOUSE, PARKER STREET, KINGSWAY, LONDON, W.C.2
THE PITMAN PRESS, BATH
PITMAN HOUSE, LITTLE COLLINS STREET, MELBOURNE
UNITEERS BUILDING, RIVER VALLEY ROAD, SINGAPORE
27 BECKETTS BUILDINGS, PRESIDENT STREET, JOHANNESBURG

ASSOCIATED COMPANIES
PITMAN PUBLISHING CORPORATION
2 WEST 45TH STREET, NEW YORK
205 WEST MONROE STREET, CHICAGO
SIR ISAAC PITMAN & SONS (CANADA), LTD.
(INCORPORATING THE COMMERCIAL TEXT BOOK COMPANY)
PITMAN HOUSE, 381-383 CHURCH STREET, TORONTO

BOOK PRODUCTION WAR ECONOMY STANDARD

THIS BOOK IS PRODUCED IN COMPLETE CONFORMITY WITH THE AUTHORIZED ECONOMY STANDARDS

MADE IN GREAT BRITAIN AT THE PITMAN PRESS, BATH
D7—(T.257)

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CHAPTER I

REFRIGERATING SYSTEMS

REFRIGERATION is the controlled removal of heat. This process of transferring heat from a body involves numerous applications of physics, particularly of the laws of heat.

There are two principal methods of removing heat by refrigeration—

1. The vapour compression system.
2. The absorption system.

1. Vapour Compression System. This method is used with large industrial plants which require the rapid removal of large quantities of heat.

With this method a vapour is compressed and then cooled to a liquid. This liquid is made to re-evaporate without gaining or losing any heat, gaining latent heat during evaporation at the expense of sensible heat. Loss of sensible heat reduces the temperature so that the vapour formed is very cold.

This cold vapour is circulated near the body to be cooled and from which it receives heat. After gaining heat in this manner, the vapour is recompressed and liquefied to repeat another cycle.

Compressor. A diagrammatic illustration of a vapour compression system is shown in Fig. 1. Vapour is compressed from a low to a high pressure in some form of compressor, which may be of either the reciprocating or rotary type. Discharge pressures may vary between 100 and 200 lb./in.^{2}, while the suction pressure is kept as low as possible, generally about 15 lb./in.^{2}

Motive power may be provided by an electric motor, heavy-oil, or steam engine developing up to 100 h.p. Reciprocating compressors can create higher pressures than rotary compressors and may be designed with two or three stages.

Condenser. The hot vapour which leaves the compressor is liquefied *at this high temperature and pressure* by removing its latent heat. As indicated in Fig. 1, cold water is circulated through the condenser and becomes warmer as it removes heat from the vapour. The temperature of the circulating water generally rises about 20° F. between inlet and outlet. Removal of heat from the vapour makes it become wet, and this wetness increases until the latent heat of vaporization is completely removed, when the vapour is liquefied.

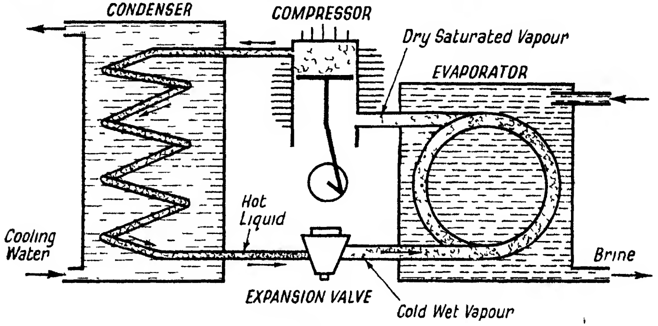

FIG. 1

Care is needed in correctly designing condensers to remove heat from the vapour. If the latent heat is not completely removed the vapour is not completely liquefied, and the subsequent cycle is unsatisfactory. On the other hand, if heat is extracted from the vapour after the removal of all the latent heat, an unnecessary pre-cooling will occur.

Expansion Valve. Although quite small, this is a very important item of the vapour-compression system. In this valve, the liquid is made to re-evaporate without gaining or losing any heat, i.e. at constant total heat. A more detailed account of this valve is given later, but it will be seen from Fig. 2 that this valve allows the liquid to pass from a small space into a larger one. On entering the large space the liquid is made to expand, and in so doing it is converted into a very wet vapour. This change of state can only occur satisfactorily if the expansion takes place at constant total heat, in that the latent heat gained in evaporating is produced at the expense of the sensible heat.

The relative dimensions of the two orifices or spaces are important in the performance of this valve, and the ratio of their areas is proportional to the ratio of the high- and low-side pressures. Precautions must always be taken to allow for wire-drawing or throttling as the vapour flows through the valve. Only by correct design, and lagging the valve with suitable insulation, is it possible to obtain conditions approaching constant total heat which are so necessary for this valve.

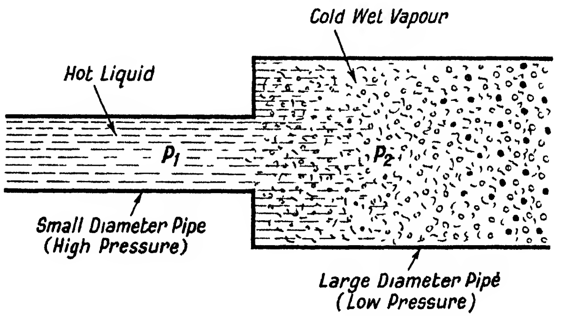

FIG. 2

EXAMPLE. Liquid ammonia at an expansion valve has a total heat of 240 B.Th.U./lb. reckoned at 0° F., and is expanded down to such a temperature that it has used 200 B.Th.U./lb. in latent heat of vaporization. Calculate the final temperature if it is initially at 145° F. If heat leakage into the valve causes a *gain* of 20 B.Th.U./lb., calculate the final temperature after expansion. Assume specific heat of liquid ammonia 1·25.

SOLUTION. The total heat before expansion is 240 B.Th.U. for 1 lb. of ammonia. This ammonia is liquid, and therefore only has sensible heat. When the ammonia expands in the valve it uses 200 B.Th.U. in evaporating, so that it then possesses latent as well as sensible heat.

Reckoning from 0° F. for 1 lb. ammonia after expansion, let T = final temperature, ° F.

$$\begin{aligned} \text{Sensible heat} &= 1{\cdot}25(T - 0)\ \text{B.Th.U.} \\ \text{Latent heat} &= 200\ \text{B.Th.U.} \\ \text{Total heat} &= 200 + 1{\cdot}25\,T\ \text{B.Th.U.} \end{aligned}$$

Since no heat is lost, this must equal the initial total heat before expansion

$$\therefore\ 200 + 1{\cdot}25\,T = 240$$
$$\text{i.e. } \underline{\underline{T}} = \underline{\underline{32^\circ\text{ F.}}}\quad \textit{Answer.}$$

If heat leakage supplies a gain of 20 B.Th.U./lb., then the total heat after expansion will be equal to $240 + 20 = 260$ B.Th.U./lb.

$$\therefore\ 200 + 1{\cdot}25\,T = 260$$
$$\text{i.e. } \underline{\underline{T}} = \underline{\underline{48^\circ\text{ F.}}}\quad \textit{Answer.}$$

This example shows that if heat leaks into the system, a 10 per cent heat leakage increases the final temperature by 50 per cent. Heat leakage to an uninsulated valve may easily occur as the lower vapour temperature is below that of the atmosphere; in this case it is at freezing point, 32° F. The care required of an expansion valve may be fully appreciated by this example, particularly as this is a vital part of the circuit.

Evaporator. The wet vapour which leaves the expansion valve is at a very low temperature. Two systems are used to apply these reduced temperatures to the removal of heat in the refrigeration of a body, namely—

1. Direct method.
2. Indirect method.

With the direct method, the wet vapour is controlled to flow through piping or grids in the cold-store. Heat is removed directly from the body within the store, through the piping to the cold vapour.

The indirect method is used with large units using several cold-stores which are required at various different temperatures. Vapour coils are immersed in brine tanks, from which pipes lead to the various cold stores. Heat is extracted from the brine by the cold vapour, and hence the brine is cooled. This brine is circulated around the grids in the cold-store so as to receive heat from the bodies it contains. A continuous brine flow is maintained, usually by electric pumps, so that circulation is continuous. The quantity of brine flowing through the grids is regulated by opening or closing the brine valve, so that it is possible to control the removal of heat and therefore the temperature within the cold-store.

"Brines" are used as the cooling media in indirect systems. Such liquids do not freeze at the low temperatures used in most refrigerators. If ordinary water was used, it would freeze at 32° F. under normal conditions and limit the operational temperature of the machine. Brine is made by adding chemicals to the water so that a "salt" solution is formed. For this purpose, calcium or sodium chloride are generally used, since they are cheap, dissolve easily, and have suitable chemical properties.

As the wet vapour receives heat from the brine, so it evaporates and becomes completely saturated. This vapour then remains at a very low temperature and pressure while it takes up latent heat. The cold vapour is finally sucked through the compressor inlet ports to be compressed once more, hence the cycle is repeated.

EXAMPLE. Calculate the necessary diameter of brine pipes for a system removing 112,000 B.Th.U./hr., assuming the brine has a specific heat of 0·7, density 66 lb./ft.^{3}, and a temperature rise of ·8° F., given that the maximum brine velocity is 140 ft./min.

SOLUTION. We must first determine the mass of brine flowing through the system.

Let W = brine flow, lb./min.

$$\therefore \text{ Heat gained by brine/hr.} = (W\,60) \times 0{\cdot}7 \times 8$$
$$= 112{,}000 \text{ B.Th.U.}$$
$$W = 333 \text{ lb./min.}$$

Since 1 ft.^{3} of brine weighs 66 lb., the brine flow expressed in ft.^{3}/min. will be

$$= \frac{333}{66}$$
$$= 5{\cdot}05 \text{ ft.}^3\text{/min.}$$

Let d = diameter of pipes, in ft.

$$\therefore \text{ Area of section} = \frac{\pi}{4} d^2 \text{ ft.}^2$$

$$\therefore \text{Brine flow} = 140 \times \frac{\pi}{4} d^2 = 5{\cdot}05 \text{ ft.}^3/\text{min.}$$

$$\therefore d^2 = 0{\cdot}0459 \text{ ft.}^2$$

$$d = 0{\cdot}214 \text{ ft.}$$

$$= \underline{\underline{2{\cdot}57}} \text{ in. } \textit{Answer.}$$

The simplicity of this calculation is typical of the method adopted in designing the various parts of a refrigerating system.

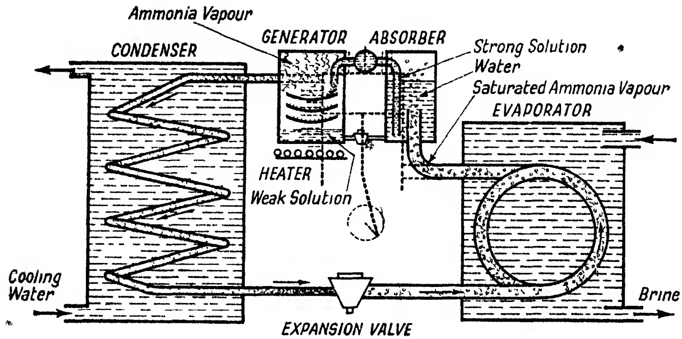

FIG. 3

2. Absorption System. The principle of the absorption system is used in many domestic units and in the so-called "gas" refrigerator.

The difference between this system and the vapour-compression system is that the vapour compressor of the latter is replaced by an absorber in which the ammonia vapour is absorbed by the water. This water is then heated and vapour liberated at a much higher temperature and pressure. By this means, heat energy replaces mechanical energy in transferring the ammonia from the low-pressure to the high-pressure side.

As shown in Fig. 3, the absorption system is similar to the vapour-compression system in certain respects. Saturated vapour from the evaporator is mixed with water in the absorber. Owing to the affinity of water for ammonia, a strong solution is formed, which passes on into the generator. The solution is heated and vapour passes off at a high pressure and temperature. Thus, as in the former vapour-compression system, the vapour may be condensed to a liquid in the condenser, expanded to a very cold, wet, vapour in the expansion valve and, finally, acquire heat from the evaporator. After the absorber, this system is identical with that of Fig. 1.

In an actual small domestic refrigerator, the elements of the system are combined in a single unit in such a manner that there are no separate parts with inter-connecting pipes as shown in Fig. 3.

Absorber. In this, the saturated vapour is pumped into a pool of water. The system must be very cold in order that the ammonia may be sufficiently absorbed, for which reason the absorber is usually cooled by air or circulating water. Water absorbs ammonia best at low temperatures.

After leaving the absorber, the strong ammonia solution may be circulated to the generator, either by a mechanically-operated centrifugal pump or by an injector using a small, high-pressure, ammonia jet. Very small domestic units use the injector-pumping method, since it does not introduce the complications of electric motors.

EXAMPLE. It is possible to absorb 0·698 lb. ammonia in 1 lb. of water at a pressure of 30 lb./in.^{2} and a temperature of 80° F. Assuming specific heat of liquid ammonia is 1·10 and latent heat of vaporization 545 B.Th.U./lb. and that vapour is given off at 100 F., what heat must be added to evaporate 1 lb. ammonia?

SOLUTION. The 1 lb. of ammonia will be contained in $\frac{1}{0{\cdot}698} = 1{\cdot}43$ lb. of water. Notice the solution will weigh $1 + 1{\cdot}43 = 2{\cdot}43$ lb.

To evaporate off the ammonia we must supply sufficient heat to—

1. Raise the temperature of 1 lb. ammonia from 80° F. to 100° F.

2. Evaporate 1 lb. ammonia at 100° F.

3. Raise the temperature of 1·43 lb. of water from 80° F. to 100° F. Thus for each stage we calculate—

Heat required to raise temperature of 1 lb. ammonia

$$= 1 \times 1{\cdot}1 \times (100 - 80)$$

$$= 22 \text{ B.Th.U.}$$

Heat required to evaporate 1 lb. ammonia

$$= 545 \text{ B.Th.U.}$$

Heat required to raise temperature of 1·43 lb. water

$$= 1{\cdot}43 \times 1 \times (100 - 80)$$
$$= 28{\cdot}6 \text{ B.Th.U.}$$

∴ Total heat required to evaporate 1 lb. ammonia from solution

$$= 22 + 545 + 28{\cdot}6$$
$$= \underline{595{\cdot}6} \text{ B.Th.U. } \textit{Answer.}$$

This heat may be supplied to the solution either by the products of combustion from a gas burner or by electrical heater elements within the generator, but usually by the former method.

Generator. This is a modified form of boiler or water heater. With a "gas" refrigerator, the generator is heated by gas jets. Ammonia vapour which is given off passes to the condenser and proceeds to remove heat in the normal way. The solution from which the ammonia has evaporated is either drained off or re-used in the absorber. This latter method is best, because the ammonia which is initially absorbed cannot be completely removed, so that by passing the weak solution back to the absorber through an auxiliary reducing valve it may be used again in the plant.

3. **"Gas" Refrigerator.** Modifications embodied in the elementary absorption units have led to the introduction of the modern "gas" refrigerator, such as is shown in Fig. 4.

Ammonia gas in the water solution of the generator is given off when heat is applied. The air condenser is used to liquefy the vapour and liquid ammonia drops into the hydrogen vessel. This vessel, which is filled with hydrogen under pressure, enables the total pressure, due to the hydrogen and ammonia, to equalize the pressure in the remainder of the apparatus. This ensures pressure-equilibrium throughout. A high pressure is exerted on the hydrogen in the vessel, while the partial pressure due to the ammonia is very small. This is rather similar to injecting liquid ammonia into a vacuum, the effect of which is to make the liquid evaporate. Hydrogen is a very light gas; gaseous ammonia is heavy, and therefore it sinks to the bottom of the hydrogen-vessel. At the bottom of the vessel, ammonia gas is re-absorbed into a strong solution and siphons back into the absorber. The evaporating process at constant pressure in the hydrogen vessel requires the addition of heat, which is obtained directly from the cold room.

The "gas" refrigerator, then, is merely a complication of the

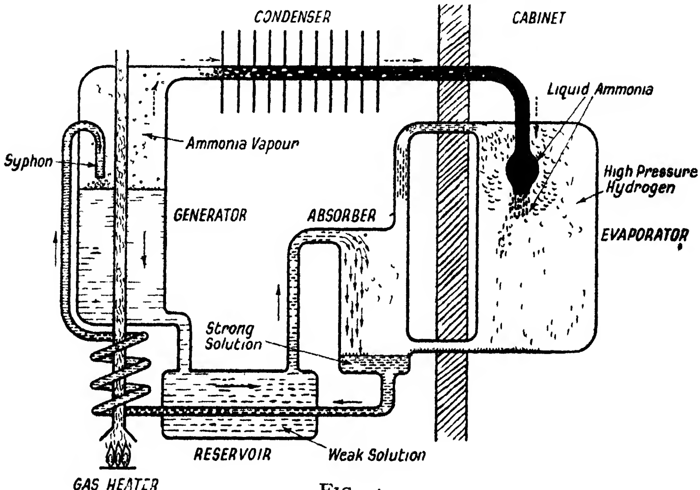

FIG. 4

absorption system. Use of an inert hydrogen pressure in the vessel ensures that every part of the system is at the same constant total pressure, but that the partial pressure exerted by the ammonia varies according to the part of the cycle which is under consideration. Determination of the actual pressure exerted by either hydrogen or ammonia in any part of the refrigerator involves an application of Dalton's Laws for the mixture of vapours.

4. **Heat Removal.** Within the cold store, heat must be removed from two sources—

1. Chemical reactions within the food, resulting in respiration and the generation of heat.
2. Leakage of heat from the warm exterior to the cold interior of the store.

The respirational effects of various foods vary and the amount of their heating properties differ according to their storage conditions, thus apples at 32° F. liberate about 800 B.Th.U./ton/day, while at 85° F. they give off between 6000 and 15,000 B.Th.U./ton/day. The removal of this chemically-formed heat is a fundamental in the storage of foodstuffs.

Leakage of heat into the cold store is governed by various factors involving consideration of the various forms of heat transference and is affected by the efficiency of the insulation. Radiation, conduction, and convection are the principal factors involved in the transference of heat, the total heat leakage from all these sources being a load which seriously impedes the economic operation of the refrigerator.

Calculations

1. Frictional effects in a compressor add 152,500 B.Th.U./hr. to a vapour. Calculate the horse-power of the compressor. 778 ft.-lb. = 1 B.Th.U. 33,000 ft.-lb./min. = 1 h.p. [60 h.p.]

2. Tests on a refrigerator show that 99·5 B.Th.U./lb. vapour are added by the compressor. If the compressor adds a total of 11,940 B.Th.U./hr., calculate the mass flow of vapour in lb./min. [2 lb./min.]

3. Calculate the water flow, in gal./min. required by a condenser to remove 480,000 B.Th.U./hr., assuming the temperature of the condensing water may rise from 45 to 65° F. [40 gal./min.]

4. In an expansion valve, the liquid has an initial temperature of 200° F. Assuming a specific heat of 1·7 for the liquid determine the temperature of the cold vapour if it acquires 302 B.Th.U./lb. latent heat from the liquid heat during the expansion. [22·4° F.]

5. With an indirect-system evaporator, it is required to remove 210,000 B.Th.U./hr. Assuming the brine has a specific heat of 0·70 and a temperature rise of 5° F., calculate the mass flow in gal./min. [100 gal./min.]

6. Calculate the quantity of heat removed from a cold store in B.Th.U./hr. by a "gas" refrigerator using $\frac{1}{10}$ lb. of ammonia per hour. Assume a latent heat of vaporization of 510 B.Th.U./lb. in the hydrogen vessel. [51 B.Th.U./lb.]

7. How many lb. of meat can be stored in a "gas" refrigerator extracting 720 B.Th.U./day, assuming that meat generates heat at the rate of 0·5 B.Th.U./lb./hr., and that there is a heat leakage into the cabinet of 26 B.Th.U./hr.? [8 lb.]

Solutions to Calculations

1.

$$\text{Work added by compressor} = 152{,}500 \times 778 \text{ ft.-lb./hr.}$$
$$= 118{,}900{,}000 \text{ ft.-lb./hr.}$$
$$= 1{,}980{,}000 \text{ ft.-lb./min.}$$
$$\therefore \text{h.p.} = \frac{1{,}980{,}000}{33{,}000}$$
$$= \underline{\underline{60}} \quad \textit{Answer.}$$

2.

Let W = mass flow of vapour, lb./min.

∴ Heat added by compressor per hour

$$= (W \,.\, 60) \times 99{\cdot}5$$
$$= 5970\, W = 11{,}940$$
$$\therefore W = \frac{11{,}940}{5970}$$
$$= \underline{\underline{2{\cdot}0 \text{ lb./min.}}} \quad \textit{Answer.}$$

3.

Let G = mass flow of water, gal./min.

$$= 10\, G \text{ lb./min.}$$

Temperature rise of water

$$= 65^\circ \text{ F.} - 45^\circ \text{ F.}$$
$$= 20^\circ \text{ F.}$$

∴ Heat removed by circulating water per hour

$$= (10 \,.\, G \,.\, 60) \times 20$$
$$= 12{,}000\, G = 480{,}000$$
$$\therefore G = \frac{480{,}000}{12{,}000}$$
$$= \underline{\underline{40 \text{ gal./min.}}} \quad \textit{Answer.}$$

4.

Let T = final temperature, ° F.

Total heat of liquid, from 0° F.

$$= 1 \times 1{\cdot}7 \times 200$$
$$= 340 \text{ B.Th.U./lb.} \quad . \quad . \quad (1)$$

Latent heat of vapour = 302 B.Th.U./lb.

Sensible heat of vapour = $1 \times 1{\cdot}7 \times T$ B.Th.U./lb.

∴ Total heat of vapour, from 0° F.

$$= 302 + 1{\cdot}7\,T \text{ B.Th.U./lb.} \quad . \quad (2)$$

Total heat is constant in an expansion valve,

$$\therefore 302 + 1{\cdot}7\,T = 340$$
$$T = \frac{38}{1{\cdot}7}$$
$$= \underline{22{\cdot}4^\circ \text{ F.}} \quad \textit{Answer.}$$

5.

Let W = mass flow of brine, gal./min. = $10\,W$ lb./min.

Heat removed by brine per hour

$$= (10\,W \,.\, 60) \times 0{\cdot}70 \times 5$$
$$= 2100\,W = 210{,}000$$
$$\underline{W} = \underline{100 \text{ gal./min.}} \quad \textit{Answer.}$$

6.

The heat from the store is used to evaporate ammonia in the hydrogen vessel.

∴ Heat removed from cold store per hour

$$= \tfrac{1}{10} \times 510$$
$$= \underline{51 \text{ B.Th.U./hr.}} \quad \textit{Answer.}$$

7.

Let M = weight of meat stored, lb.

The heat generated by this meat, together with the heat leakage, constitute the heat load to be removed by the refrigerator.

$$\text{Heat generated by meat} = 0{\cdot}5\,M \text{ B.Th.U./hr.}$$

$$\text{Heat leakage into cabinet} = 26 \text{ B.Th.U./hr.}$$

$$\therefore \text{Heat load} = 26 + 0{\cdot}5\,M = \frac{720}{24} = 30$$

$$\therefore \underline{\underline{M}} = \frac{4}{0{\cdot}5}$$

$$= \underline{\underline{8 \text{ lb.}}} \quad \textit{Answer.}$$

CHAPTER II

BRINES AND REFRIGERANTS

THE indirect system of evaporation is better than the direct system for large refrigerators because it can "store" cold. With this system, if the refrigerator is shut off the brine still remains cold and can be circulated to cool a cold store, even though the compressor itself may be shut off. Heat which the brine receives is not immediately transferred to the refrigerant, but is used to raise the brine temperature until the brine ultimately becomes warm again. The time taken to raise the brine temperature is quite long, although it necessarily depends on the store heat, capacity of the brine tanks and circuits, specific heat of the brine, etc., so that it is possible to stop the compressor for an appreciable interval of time.

EXAMPLE. Assuming that a store supplies heat to the brine at a constant rate of 15,000 B.Th.U./hr., calculate the time taken to raise the temperature of a brine circuit containing 1·6 tons of brine from 26 to 47° F. Specific heat of brine = 0·68.

SOLUTION. The brine temperature must be raised through 47° F. – 26° F. = 21° F.

∴ Heat required by brine to raise its temperature

$$= (1{\cdot}6 \times 2240) \times 0{\cdot}68 \times 21$$
$$= 51{,}000 \text{ B.Th.U.}$$

In 1 hr. the store adds 15,000 B.Th.U. to the brine.

∴ Time taken to raise brine temperature

$$= \frac{51{,}000}{15{,}000}$$
$$= \underline{3{\cdot}4} \text{ hr. } \textit{Answer.}$$

This example goes to show that, in practice, it is possible to close down the machinery for a fairly long period and yet stll continue to remove heat from the cold store at the desired rate. Such an arrangement provides time for repairing machinery and reduces the working-hours of the staff. Both of these circumstances are of great value to large industrial installations.

It might also be added that, in practice, the heat transference to the brine is not constant. The rate of heat transmission varies according to the temperature difference between the brine and the store, so that as the brine becomes warmer the temperature difference falls and the heat transference rate decreases. The time taken to warm the brine is therefore longer in practice than under conditions of constant heat transference assumed in the foregoing example.

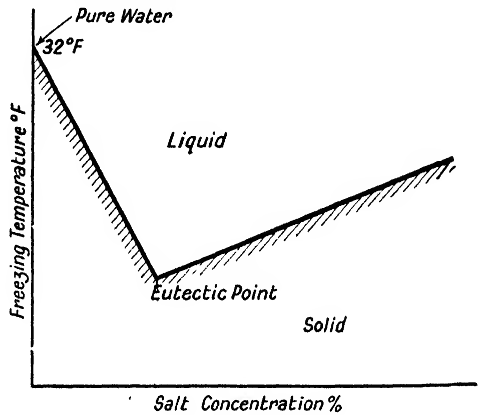

FIG. 5

1. **Brines.** Various concentrations of brine are obtained by adding different proportions of chemicals. Such varying proportions have properties which differ, particularly the freezing point.

2. **Freezing Point.** This is the most important property of any brine concentration. A curve showing the freezing point of a brine at various concentrations is given in Fig. 5. It will be seen that as salts are added and the brine concentration is increased, so the freezing point falls to a minimum value. Further additions of salt beyond this critical concentration cause the freezing point to rise. This minimum freezing point occurs with the eutectic mixture. For a sodium chloride brine the eutectic mixture occurs with a 24 per cent concentration.

Freezing point is not the only criterion of brine concentration, since other factors, such as its liability to corrode metals, its specific gravity, specific heat and viscosity, must also be duly considered.

3. **Specific Gravity.** Ordinary laws of mixtures calculated by arithmetical addition do not always hold for chemical solutions owing to crystalline phenomena within the chemicals themselves. Thus, if the specific gravity of a chemical salt in the solid state is 2·5, then if a solution is formed with 12 per cent salt by weight, the arithmetical law of mixtures would allow calculation of the specific gravity as follows—

Consider 1 lb. of solution

$$\text{Weight of salt in solution} = 0{\cdot}12 \text{ lb.}$$

$$\text{Weight of water in solution} = 0{\cdot}88 \text{ lb.}$$

$$\text{Volume of salt} = \frac{0{\cdot}12}{2{\cdot}5 \,.\, 62{\cdot}5} \text{ ft.}^3$$

$$= \frac{\cdot 048}{62{\cdot}5} \text{ ft.}^3$$

$$\text{Volume of water} = \frac{0{\cdot}88}{62{\cdot}5} \text{ ft.}^3$$

$$\text{Volume of 1 lb. water} = \frac{1}{62{\cdot}5} \text{ ft.}^3$$

$$\text{Specific gravity of solution} = \frac{1/62{\cdot}5}{0{\cdot}928/62{\cdot}5}$$

$$= \underline{\underline{1{\cdot}078.}}$$

In actual practice this result does not hold, in fact the specific gravity of a 12 per cent brine solution is 1·10. The relationship between specific gravity and concentration for calcium chloride brines is given in Fig. 6.

Deviations from the law of mixtures are due to combinations which occur when salts are dissolved. Salts are always somewhat porous internally, and although they may have a given outside volume the true value may be less. In addition there is a limit to the quantity of salt which water will contain in solution, a precipitate being formed on the addition of further chemicals to a saturated solution.

4. **Specific Heat.** The specific heat of a brine does not remain constant. Two factors influence change of specific heat, temperature, and density. Variation of specific heat for

different calcium chloride brines are shown in Fig. 7. As the salt concentration increases, so the specific heat decreases.

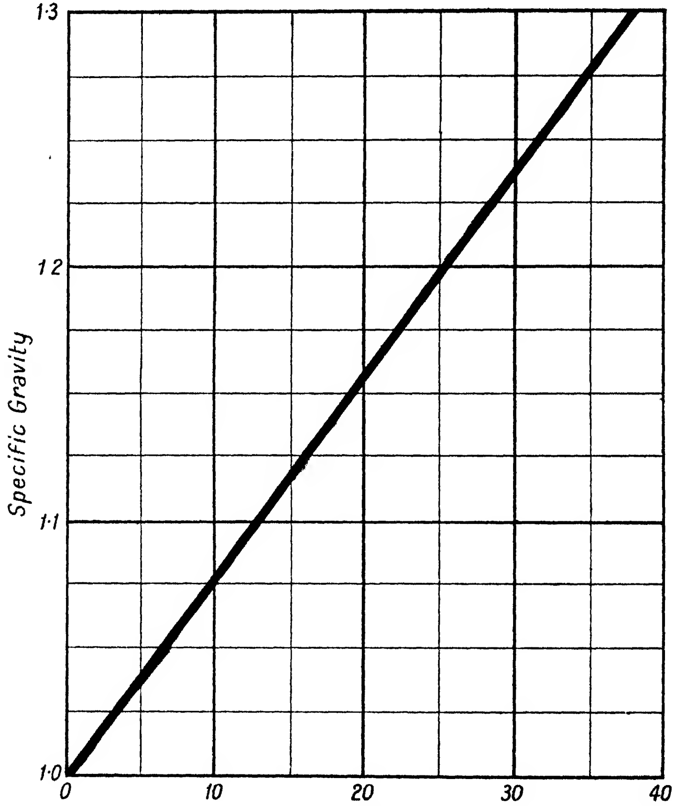

FIG. 6

The effect of variations of specific heat is to complicate calculations of tank capacity and evaporate sizes. Certain allowances for variation in specific heat must accordingly be made in the design. The following example shows that an appreciable variation in the brine flow is calculated by taking different values for the specific heat.

EXAMPLE. An evaporator coil transmits 3·0 B.Th.U./

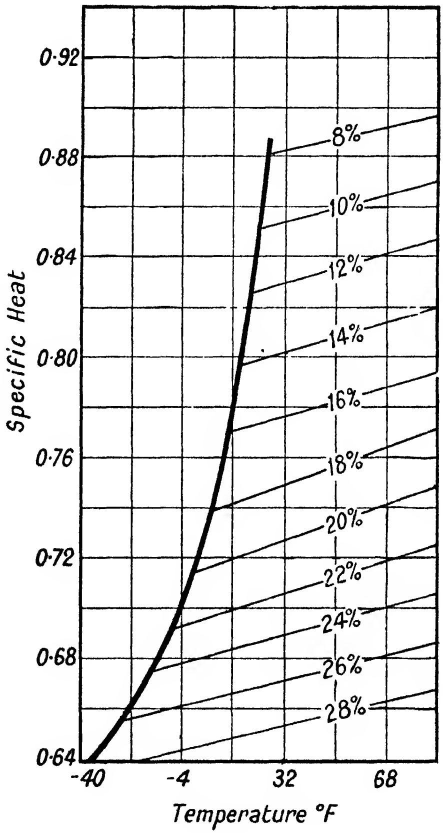

FIG. 7

ft.^{2}/hr./° F. for 800 ft.^{2} with a temperature difference of 12° F. Calculate the brine flows in lb./min. for calcium chloride brines of 10 per cent and 20 per cent salt concentrations, mean brine temperature 23° F. Temperature rise of brine 2° F.

SOLUTION. From Fig. 7 it will be seen that brine at 23° F. and at a concentration of 10 per cent has a specific heat of 0·850, while a concentration of 20 per cent has a specific heat of 0·728.

For a temperature difference of 1° F.,

$$\text{Heat transferred} = 3{\cdot}0 \times 800$$
$$= 2400 \text{ B.Th.U./hr.}$$

For a temperature difference of 12° F.,

$$\text{Heat transferred} = 2400 \times 12$$
$$= 28{,}800 \text{ B.Th.U./hr.}$$

Let W_1 = brine flow for 10 per cent concentration, lb./min.

$$\text{Heat transferred/hr.} = (W_1 \,.\, 60) \times 0{\cdot}850 \times 2 = 28{,}800$$

$$\underline{\underline{W_1}} = \frac{28{,}800}{60 \times 0{\cdot}850 \times 2}$$
$$= \underline{\underline{282{\cdot}5}} \text{ lb./min.}$$

Let W_2 = brine flow for 20 per cent concentration, lb./min.

$$\text{Heat transferred/hr.} = (W_2 \,.\, 60) \times 0{\cdot}728 \times 2 = 28{,}800$$

$$\underline{\underline{W_2}} = \frac{28{,}800}{60 \times 0{\cdot}728 \times 2}$$
$$= \underline{\underline{329{\cdot}5}} \text{ lb./min.}$$

It will therefore be seen that the decrease in specific heat caused by increasing the brine concentration causes an increase of 47 lb./min. in the brine flow. This amounts to an increase of 16·65% of the original brine flow.

It will also be seen from this example that variation in brine concentration has an important influence upon the pump capacity. Periodical tests should always be made to ascertain the correct brine concentration. A density method is most convenient for determining this. By using a hygrometer and so obtaining the brine density application to Fig. 6 will enable the concentration to be estimated.

5. Viscosity and Friction. Salt concentration influences pump capacity by controlling brine flow. Pump power is influenced by the frictional resistance of the brine in pipes. Experiments have shown that fluid resistance depends mainly upon the physical properties of the brine and upon the viscosity.

This abstruse physical property is best explained by stating that pitch is very viscous, while water and free-running liquids have not such a high viscosity. Viscosity is therefore a force which controls the free movement of a liquid. At

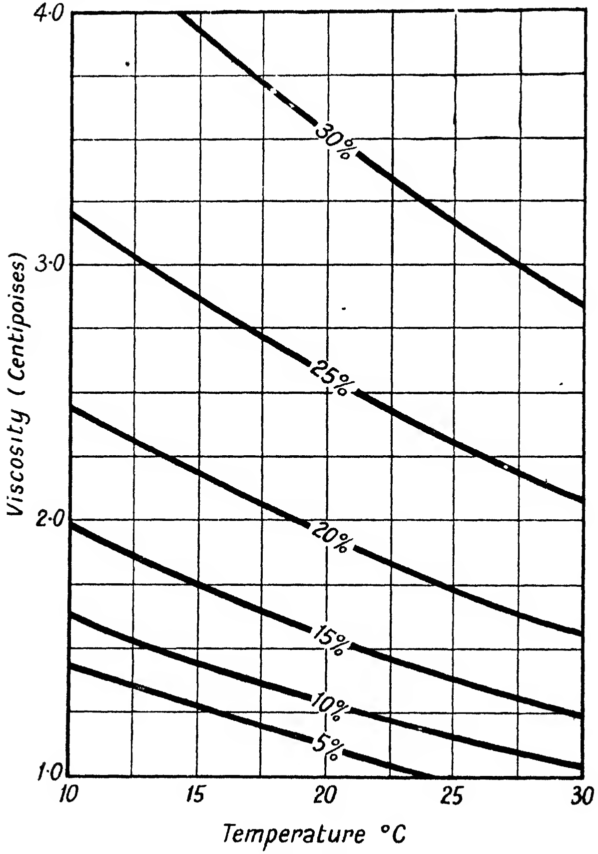

FIG. 8

constant temperature large salt concentrations are more viscous than small concentrations of salt. For a constant salt concentration, increase of temperature decreases the viscosity. The curve shown in Fig. 8 gives some indication of the brine viscosity at different concentrations for various temperatures.

Hydraulically, two types of flow exist within a pipe, viscous and turbulent flow. For viscous flow, the pressure-drop in lb./in.^{2} is given by the equation

$$p = 0.000668 \frac{\mu \, . \, l \, . \, V}{D^2} \text{ lb./in.}^2$$

where μ = coefficient of viscosity, centipoises
l = length of piping, *ft.*
V = brine velocity, ft./sec.
D = pipe diameter, in.

This expression is for velocities from zero to a critical velocity which varies for different materials and pipe diameters. Above this velocity the equation giving the pressure drop for turbulent flow must be used, namely,

$$p = 0.33 \, . \, f \, . \, \frac{l}{D} \, . \, V^2 \text{ lb./in.}^2$$

where f = friction factor
l = length of piping, ft.
V = brine velocity, ft./sec.
D = pipe diameter, ins.

The friction factor, f, is reasonably constant at about 0·075. It does, however, vary according to the condition inside the pipe, whether it is rough or smooth, or whether solid matter is deposited on its sides.

Example. Calculate the pressure drop through an evaporator consisting of 23 pipes, each 2 in. diameter, 4 ft. long, including bends, assuming a friction factor of 0·06 and brine is turbulent with a velocity of 5 ft./sec. Estimate the power of the brine pump, assuming brine density of 67 lb./ft.^{3}

Solution. The overall pipe length of the whole evaporator $= 23 \times 4 = 92$ ft.

$$\therefore \text{ Pressure drop, } p = 0.33 f \, . \, \frac{l}{D} \, . \, V^2$$

$$= 0.33 \times 0.06 \times \frac{92}{2} \times (5)^2$$

$$= 22.75 \text{ lb./in.}^2$$

To obtain the equivalent head of the brine pressure it must first be reduced to lb./ft.^{2} and then divided by the brine density.

$$\begin{aligned}
\text{Brine pressure drop} &= 22{\cdot}75 \times 144 \\
&= 3280 \text{ lb./ft.}^2 \\
\therefore \text{ Equivalent head} &= \frac{3280}{67} \\
&= 49{\cdot}0 \text{ ft.} \\
\text{Sectional area of brine pipe} &= \frac{\pi}{4}(2)^2 \\
&= 3{\cdot}142 \text{ in.}^2 \\
&= 0{\cdot}0218 \text{ ft.}^2 \\
\therefore \text{ Brine flow} &= 0{\cdot}0218 \times 5 \\
&= 0{\cdot}109 \text{ ft.}^3\text{/sec.} \\
&= 6{\cdot}54 \text{ ft.}^3\text{/min.} \\
&= 6{\cdot}54 \times 67 \text{ lb./min.} \\
&= 438 \text{ lb./min.}
\end{aligned}$$

To determine the power it is assumed that the pump must raise this 438 lb. of brine through a height of 49·5 ft. in 1 min.

$$\begin{aligned}
\therefore \text{ Work done by pump in 1 min} &= 438 \times 49{\cdot}5 \\
&= 21{,}700 \text{ ft.-lb.} \\
\text{i.e. pump power} &= \frac{21{,}700}{33{,}000} \\
&= \underline{\underline{0{\cdot}658}} \text{ h.p.} \quad \textit{Answer.}
\end{aligned}$$

This example gives an indication of the method adopted in determining the power required by the pump. No allowance has been made for losses within the pump, so that the answer gives only the output power. If the pump efficiency were assumed to be 80 per cent, then the input power to the pump, from which most calculations of the design are made, would be

$$\frac{100}{80} \times 0{\cdot}658 = 0{\cdot}823 \text{ h.p.}$$

6. Corrosion. Brine is a very corrosive agent. It tends to corrode all metals which it contacts. Precautions must accordingly be taken both in the design and during the operation of a refrigerating plant to eliminate any corrosive activity as far as possible.

Corrosion is most commonly an oxidization of the metal surface set up by oxygen in the air, or within the brine, being chemically associated with the surface under favourable circumstances. To prevent corrosion by oxygen in the air, all brine tanks should be sealed off with non-return vent valves. By this means of hermetically sealing the brine tanks, air is prevented from entering so that the oxygen which it contains cannot corrode the interior of the tank. To expel the air entrained in the brine, two precautions must be taken. Firstly, falling jets of brine must be avoided as they collect air. To do this, the returning brine must be made to re-enter the tank at some point below the brine surface. Secondly, air already entrained within the brine must be removed, for which purpose powdered zinc is added as it galvanizes the sides and the zinc solutes assist in removing oxygen by the formation of inert zinc oxides.

Electrolytic corrosion is produced by using two different metals in the brine circuit. When immersed in brine, two metals act in the manner of a simple cell and produce a small current. This current, small as it may be, electrolyzes one metal, which gradually wears away. To prevent such corrosion, the use of dissimilar metals should be avoided. Stray electric currents produce the same effect, for which reason it is important that all electrical equipment should be well insulated and the whole system earthed.

Salt concentration of the brine influences the extent of the corrosion, both by chemical corrosion and electrolysis. Weak brines corrode more than strong brines. It is also desirable to maintain the brine slightly alkaline. To maintain alkalinity of the brine, a mixture of caustic soda with the solution at the rate of 1 lb. per 1000 gal. is recommended. Litmus paper tests are preferable as a simple test of alkalinity; an alkaline solution turns violet litmus paper to blue and an acid solution turns it to red.

Other chemical additions to brines, in order to prevent corrosion, include the use of sodium chromates. The Corrosion Committee of the American Society of Refrigerating Engineers recommends the addition of 125 lb. of sodium dichromate to every 1000 ft.^{3} of calcium brine, or 200 lb. to every 1000 ft.^{3} of sodium chloride brine. Sodium chloride forms a protective film over metals which prevents contact between the metal and brine and effectively reduces corrosion.

7. Refrigerants. A refrigerant is the vapour which is alternatively liquefied and evaporated during the process of removing heat in the refrigerator.

It is an essential property of the refrigerant that it can be liquefied. Refrigerants must always remain as vapours under the conditions within the refrigerator; they should never become gases. The essential difference between a vapour and a gas is that whereas the former can be liquefied by the application of pressure, a gas cannot be liquefied without the application of pressure and removal of heat. It is necessary therefore that a refrigerant must have a high critical pressure and temperature which are not reached in any part of the cycle.

Since the refrigerant must remain liquid at low temperatures, it is further desirable for it to have a low freezing point. Other essential properties of a refrigerant include high latent and specific heat of the vapour and low specific heat of the liquid. These properties ensure the extraction of the utmost heat for the application of the minimum power.

Among the vapours which possess most of these properties are ammonia, freon, methyl chloride, sulphur dioxide, and carbon dioxide.

8. Ammonia. The critical pressure for ammonia is 1657 lb./in.^{2}, which is far in excess of any pressure used in refrigeration, while it has a freezing point at $-108°$ F. Ammonia is produced cheaply and has been used in refrigerating practice for many years, since it possesses fewer of the disadvantages associated with other vapours. Operating costs of this refrigerant are lower than for any other, chiefly on account of its high latent heat. Vapour at 5° F. has a latent heat of 565 B.Th.U./lb. as compared with that of carbon dioxide at 117 B.Th.U./lb. or sulphur dioxide at 169 B.Th.U./lb

Ammonia vapour is almost non-corrosive to iron or steel, although there have been adverse reports regarding its effect on galvanized or tinned surfaces. Liquid ammonia rapidly destroys galvanized surfaces. Stainless steels appear to be unaffected by ammonia in either the vapour or liquid state. Laboratory tests show that ammonia is detrimental to copper, but it appears that oil films cover cuprous surfaces in actual systems and retard any deterioration. Copper parts in actual refrigerators have remained uninfluenced by ammonia after years of service. When air or moisture are present with the ammonia, all metal deteriorates rapidly.

Refrigerants tend to wash oil from lubricated surfaces with two results--

1. Surface lubrication is impeded.
2. The refrigerant is adulterated with oil.

For the loss of oil some alternative arrangement is made to ensure an ample supply of lubricant. Lubricating oils may also be doped so as to maintain some sort of film over the surface and also to assist lubrication. Colloidal graphite serves as an excellent dope for ammonia compressors.

Adulteration of ammonia with lubricating oil has very little effect on its physical properties, although it does tend to reduce the viscosity and lower the alkalinity a little. Both of these effects have not been fully studied, although there is no doubt that they are negligible with this refrigerant.

A further advantage of ammonia is its strong smell. Ammonia leaks may be rapidly detected by their smell and may therefore be easily repaired. Slight ammonia leaks have little or no effect upon fruits or vegetables, although excessive exposure will produce rot, scald, and burns.

Ammonia is used extensively as a refrigerant in large industrial organizations, where expert attention enables its excellent thermal properties to be fully employed.

9. Freon. This is one of the most modern of refrigerants. Freon 12 is used for air-conditioning and small refrigerator units, while Freon 11 is used in large centrifugal compressors.

Freons have suitable critical pressure and freezing points, but their real advantages lie in their freedom from undesirable toxity and the formation of explosive mixtures. Freon is the most suitable refrigerant for domestic refrigerations.

With regard to corrosion and the dissolution of lubricating oils, the freon group is very similar to ammonia. Most refrigerants have a solvent influence on rubber, fibre, and various synthetic resins. It is important, therefore, to use materials for packing glands which do not contain those soluble materials.

The harmless characteristics of freon recommend it for use in domestic refrigerators, but its poor thermal properties do not make it suitable for adoption in larger units.

10. Methyl Chloride. With a critical pressure of 969 lb./in.^{2} and a freezing point of $-144°$ F., methyl chloride possesses suitable extremes of critical pressure and temperature. Medium-sized industrial refrigerators use methyl chloride, as it is about the best commercial refrigerant for low-pressure plants. In addition to such thermal and physical properties, methyl chloride produces no reaction with lubricants such as mineral oil or glycerine.

Methyl chloride tends to be more corrosive than other refrigerants. Zinc, aluminium, and magnesium alloys should never be used with this refrigerant. It is non-corrosive to common engineering metals in the absence of any moisture. As with freon, methyl chloride is a solvent for many materials, principally rubber and asbestos. Metallic gaskets and packing are best for these compressors.

High concentrations of methyl chloride have no toxicity influence, but the vapour is somewhat inflammable. Concentrations of between 8 per cent and 17 per cent vapour with air may be explosive. Gas leaks may be detected by the faintly sweet odour, although some additions of irritating gases are made to provide a more effective warning. Methyl chloride leaks have very little effect upon either foodstuffs or other goods.

The use of this vapour with medium-sized refrigerators is therefore justified by its several advantages, although the slight explosion hazard which it involves prohibits its use in domestic appliances.

11. Sulphur Dioxide. This refrigerant is suitable for domestic appliances and its thermal properties are superior to those of the freon class. Sulphur dioxide has a latent heat of evaporation of 169·0 B.Th.U./lb. at 5° F., while Freon 12 is 69·5 B.Th.U./lb. The critical pressure of this refrigerant is 1141 lb./in.^{2}, far in excess of any requirements, while its freezing point of $-99°$ F. is more than suitable for any commercial or domestic refrigerator.

Water-free sulphur dioxide will not corrode metals, but when water is present, corrosion is rapid. Special precautions must be taken to avoid the entrainment of moisture in these circuits.

In other respects, sulphur dioxide compares favourably with freon for domestic use. Owing to its irritant odour, sulphur dioxide is not so popular as freon. Vapour leaks are easily detected by their smell. This is a deterrent and commercially undesirable, leading to its replacement by the less suitable, but safer, freon vapours.

12. Carbon Dioxide. When used with quite warm condensing water, as in this country, carbon dioxide is a suitable refrigerant. It is not so thermally efficient as other refrigerants, but can be used with small compressors for heavy loads.

Carbon dioxide is practically non-corrosive, with or without moisture and imposes no limitations as to materials either for glands or gaskets.

This vapour is absolutely harmless, so it is suitable for use in enclosed spaces such as hospitals and ships. Furthermore, carbon dioxide has a "smothering effect," and in the event of fire it may be used as an extinguisher.

Carbon dioxide has a freezing point at $-70°$ F. and may be easily converted by pressure into a solid which sublimes to a vapour without forming a liquid. This is the only refrigerant whose properties are of interest in the solid phase. Solid carbon dioxide is called "cardice." This cardice is used extensively for certain operations and is competing with ice manufactured from water, as a cooling medium. Direct sublimation to a vapour is a great asset.

Calculations

1. Calculate the temperature reached by 4000 lb. brine, constant specific heat 0·66, after a heat leakage of 71,600 B.Th.U. Initial temperature of brine $-10°$ F. [17·1° F.]

2. Assuming a specific gravity of 1·10 and a specific heat of 0·856, calculate the brine velocity through a 3 in. diameter pipe if the brine receives 30,000 B.Th.U./hr. and its temperature rises 3·5° F. Density of water 62·5 lb./ft.^{3} [0·815 ft./sec.]

3. Calculate the pipe diameter, assuming viscous flow, in a brine circuit 700 ft. long, using brine of viscosity 1·68 centipoises at a velocity of 6 ft./sec. with a total pressure drop of 1·18 lb./in.^{2} [2·0 in.]

4. Ammonia vapour has a specific volume of 8·15 ft.^{3}/lb. at 5° F. and a latent heat of 565·0 B.Th.U./lb. Saturated sulphur dioxide at 5° F. has a specific volume of 6·42 ft.^{3}/lb.

and a latent heat of 169 B.Th.U./lb. Compare the volumetric flow of each refrigerant if the condenser extracts 40,000 B.Th.U./hr. [2·635 : 1.]

SOLUTIONS TO CALCULATIONS

1. The temperature rise must be calculated and so the final temperature deduced.

Let t = temperature rise, ° F.

$$\therefore \text{Heat gained by brine} = 4000 \times t \times 0{\cdot}66 = 71{,}600$$

$$\therefore t = 27{\cdot}1° \text{ F.}$$

$$\text{Hence, final brine temperature} = 27{\cdot}1 - 10$$

$$= \underline{\underline{17{\cdot}1° \text{ F.}}} \quad \textit{Answer.}$$

2. The brine flow must first be calculated by weight from its thermal relationship to the heat reception and then reduced to volumetric flow from the density. The pipe cross-sectional area then allows calculation of velocity.

Let W = brine flow by weight, lb./sec.

$$\text{Heat gained by brine/hr.} = (W \times 60 \times 60) \times 0{\cdot}856 \times 3{\cdot}5 \doteq 30{,}000$$

$$\therefore W = \frac{30{,}000}{60 \times 60 \times 0{\cdot}856 \times 3{\cdot}5}$$

$$= 2{\cdot}78 \text{ lb./sec.}$$

$$= \frac{2{\cdot}78}{68{\cdot}75}$$

$$= 0{\cdot}04 \text{ ft.}^3\text{/sec.}$$

$$\text{Density of brine} = 62{\cdot}5 \times 1{\cdot}10$$

$$= 68{\cdot}75 \text{ lb./ft.}^3$$

$$\text{Sectional area of pipe} = \frac{\pi}{4}(3)^2$$

$$= 7{\cdot}06 \text{ in.}^2$$

$$= \frac{7{\cdot}06}{144}$$

$$= 0{\cdot}0491 \text{ ft.}^2$$

Let V = brine velocity, ft./sec.

$$\therefore \text{Brine flow by volume} = V \times 0{\cdot}0491 = 0{\cdot}04$$

Hence, brine velocity $= \dfrac{0{\cdot}04}{0{\cdot}0491}$

$= \underline{\underline{0{\cdot}815}}$ ft./sec. *Answer.*

3. Using the formula for viscous flow,

Let d = pipe diameter, in.,

$$\text{Pressure drop, } p = 0{\cdot}000668 \frac{\mu \, . \, lV}{d^2}$$

$$= 0{\cdot}000668 \frac{1{\cdot}68 \times 700 \times 6}{d^2} = 1{\cdot}18$$

$$\therefore d^2 = \frac{0{\cdot}000668 \times 1{\cdot}68 \times 700 \times 6}{1{\cdot}18}$$

$$= 4{\cdot}00$$

$\therefore$ Pipe diameter, $d = \underline{\underline{2{\cdot}0}}$ in. *Answer.*

4. Let W_1 = mass flow of ammonia, lb/hr.

$\therefore$ Condensed heat/hr. $= W_1 \times 565 = 40{,}000$

$$W_1 = \frac{40{,}000}{565}$$

$= 70{\cdot}8$ lb./hr.

$\therefore$ Volumetric ammonia flow $= 70{\cdot}8 \times 8{\cdot}15$

$= 576$ ft.^{3}/hr.

Let W_2 = mass flow of sulphur dioxide, lb./hr.

$\therefore$ Condensed heat/hr. $= W_2 \times 169 = 40{,}000$

$$W_2 = \frac{40{,}000}{169}$$

$= 236{\cdot}5$ lb./hr.

$\therefore$ Volumetric sulphur dioxide flow $= 236{\cdot}5 \times 6{\cdot}42$

$= 1520$ ft.^{3}/hr.

Thus, for volumetric flow

Sulphur dioxide flow $= \dfrac{1520}{576}$

$= \underline{\underline{2{\cdot}635 \times \text{ammonia flow.}}}$ *Answer.*

CHAPTER III

COMPRESSORS

The vapour-compressor is required to compress the vapour to such a pressure that it can be easily liquefied by the condenser. Power is needed to drive the compressor. Much attention must therefore be given to the efficiency of this machine. In the whole refrigerating cycle, the compressor is the only part which requires the application of energy.

1. **Reciprocating Compressors.** The piston of a reciprocating compressor moves to and fro (it reciprocates). As shown in Fig. 9, the piston is driven from a crankshaft. Vapour is sucked into the cylinder through one valve, then compressed until it opens and passes through, another valve.

A diagram may be drawn showing variations of pressure with volume of vapour, as in Fig. 10, called an "indicator" diagram. This diagram shows the gas admitted along line *ab* and then compressed along *bc*. The discharge valve lifts when the spring force is balanced by that in the cylinder. At *c*, the vapour is discharged to the condenser at constant pressure along *cd*. This discharge pressure is controlled by the force, or tension, in the valve springs. Vapour temperature depends upon the compression pressure and influences the condenser performance.

2. **Suction Pressure.** The pressure of vapour in the cylinder during the suction stroke depends upon two factors—

1. Piston velocity.
2. Valve port area.

Each of these are contributory factors in the wire-drawing and throttling of the vapour. When the piston velocity is excessive, a large vacuum is formed, and vapour flows in so rapidly and turbulently that it does not fill the cylinder completely. Similarly, if the valve area is small, the vapour will not fill the cylinder before the suction valve closes.

Wiredrawing causes the volume of vapour actually drawn into the cylinder to be much less than the actual stroke capacity and it also reduces the suction pressure. To form some

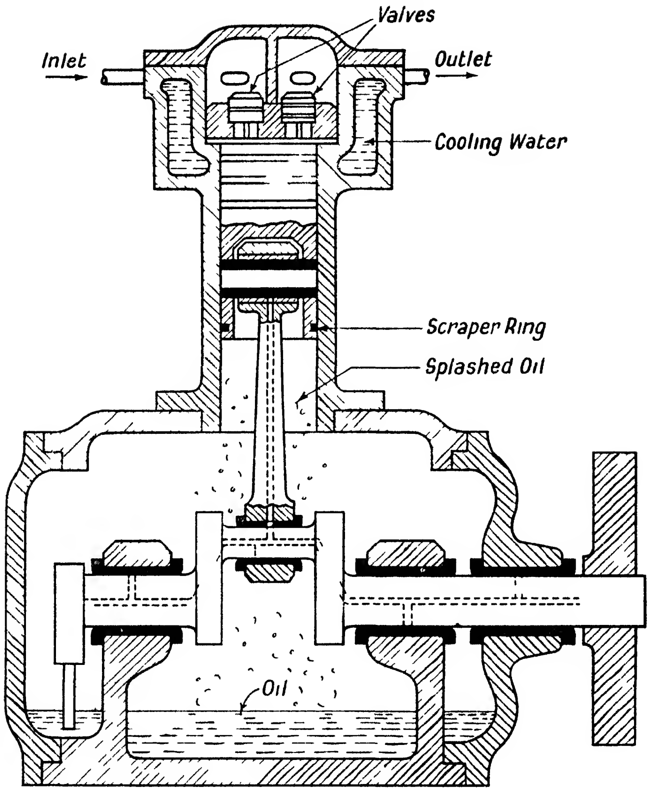

FIG. 9

estimate of the final suction pressure we use Boyle's Law, since the suction temperature is constant.

i.e. $$P_1V_1 = P_2V_2$$

EXAMPLE. A compressor of stroke volume 1·5 ft.^{3} admits 1·2 ft.^{3} of vapour at an initial pressure of 16 lb./in.^{2} Estimate the mean suction pressure inside the cylinder.

SOLUTION. The vapour which is admitted with an initial pressure of 16 lb./in.^{2} has an original volume of 1·2 ft.^{3} After the vapour fills the cylinder its pressure falls to P_2.

Using Boyle's Law $16 \times 1{\cdot}2 = P_2 \times 1{\cdot}5$

$$\therefore P_2 = \frac{16 \times 1{\cdot}2}{1{\cdot}5} = \underline{12{\cdot}8 \text{ lb./in.}^2} \quad \textit{Answer.}$$

Thus, although the pressure before the suction valve was 16 lb./in.^{2} the pressure inside the cylinder was 12·8 lb./in.^{2}.

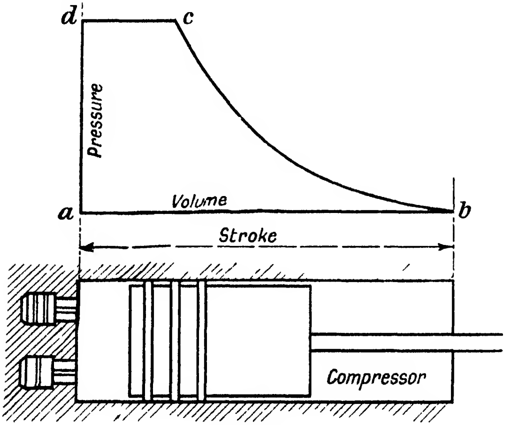

FIG. 10

The compressor therefore has to compress from 3·2 lb./in.^{2} lower than in the "low side," and therefore does more work.

In addition to the extra work done on the low-suction pressure the resultant compression temperature is higher. Suction pressure therefore influences the condenser performance.

3. **Compression Index.** The area of an indicator diagram represents the work done by the compressor per stroke. If the compressor is single acting it makes one working stroke per revolution. If it is double-acting it makes two working strokes per revolution.

When the vapour is compressed, its pressure and volume vary according to Boyle's Law modified, however, by the following conditions—

1. Internal condition of the vapour and its thermal properties.

2. Heat transference through the cylinder walls.
3. Mean piston speed.

These are expressed by the compression index N.

Thus PV^N = constant

where P = vapour pressure

V = vapour volume

N = compression index

The initial conditions of the vapour are determined by the suction pressure. Compression commences at a point such as

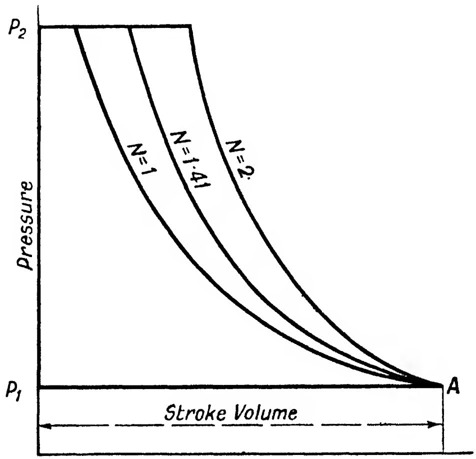

FIG. 11

A, Fig. 11. According to the compression index, so the area of the indicator diagram varies, thus the area increases with the index. The power supplied to the compressor therefore depends upon the compression index. It is greater for large indices than for small.

Theoretically, the work done in compressing a vapour from a pressure P_1 to pressure P_2, according to the law PV^N = constant, is given by—

$$WD = \frac{N}{N - 1} (P_1V_1 - P_2V_2)$$

where V_1 and V_2 are the initial and final volumes respectively.

For isothermal compression, when $N = 1$, the work done is given by

$$WD = P_1V_1 \log_e \frac{P_2}{P_1}$$

Note that logarithms are taken to the natural or Naperian Case e.

EXAMPLE. Calculate the horsepower of a single-acting compressor running at 350 r.p.m. with a stroke volume of 0·8 ft.^{3}, suction pressure 12 lb./in.^{2}, discharge pressure 120 lb./in.^{2} (i) if $N = 1$ and (ii) if $N = 2$.

SOLUTION. (i) $N = 1$. The formula $WD = P_1V_1 \log_e \frac{P_2}{P_1}$ must be used for the index $N = 1$. Pressure in lb./in.^{2} must be brought to lb./ft.^{2} by multiplying by 144.

$$\begin{aligned} \therefore \text{ Work done/revolution} &= (144 \times 12) \times 0{\cdot}8 \log_e \frac{120}{12} \\ &= 144 \times 12 \times 0{\cdot}8 \times 2{\cdot}303 \\ &= 2185 \text{ ft.-lb.} \end{aligned}$$

The compressor is single-acting and so makes only one working stroke per revolution. It makes 350 revolutions or working strokes per minute.

$$\begin{aligned} \therefore \text{ Work done/minute} &= 350 \times 2185 \\ &= 765{,}000 \text{ ft.-lb.} \\ \text{i.e. h.p.} &= \frac{765{,}000}{33{,}000} \\ &= \underline{23{\cdot}2}. \textit{ Answer.} \end{aligned}$$

(ii) $N = 2$. The formula $WD = \frac{N}{N-1}(P_1V_1 - P_2V_2)$ must be used for this index. We do not know the final volume V_2 but must calculate it.

$$\begin{aligned} \text{Since} \quad PV^N &= \text{constant} \\ 12(0{\cdot}8)^2 &= 120(V_2)^2 \\ V_2 &= 0{\cdot}253 \text{ ft.}^3 \end{aligned}$$

Substituting for $N = 2$

$$WD = \frac{2}{2 - 1}(144 \cdot 120 \cdot \cdot 253) - (144 \cdot 12 \cdot 0 \cdot 8)$$

$$= 5980 \text{ ft.-lb.}$$

At 350 r.p.m.

$$WD/\text{min.} = 5980 \times 350$$

$$= 2{,}090{,}000$$

$$\text{h.p.} = \frac{2{,}090{,}000}{33{,}000}$$

$$= \underline{\underline{63 \cdot 3}}. \quad \textit{Answer.}$$

This example shows that an increase from $N = 1$ to $N = 2$ very nearly trebles the compressor power.

The vapour may be either wet or dry when admitted to the cylinder. If it is wet, then it becomes dry with compression, and if it is already dry, the compression superheats it. In changing its condition from wet to dry or from dry to superheated, the thermal properties of the vapour control the heat used. This heat is partly taken from the work done in compressing the vapour, and therefore has some bearing on the compression index.

Heat is continually leaking into, and out of, the compressor according to the vapour temperature. This heat transference through the cylinder walls affects the total heat added to the gas. The compression index is thus again variable according to the amount of this heat leakage.

The extent of this heat leakage depends upon the rapidity with which the vapour is compressed. If the vapour is compressed slowly, the heat is able to leak away more easily than with rapid compression. This influence of piston speed on heat leakage is reflected in the compression index.

4. Discharge Pressure. Pressure at discharge depends upon the force exerted on the valves through the valve springs. This is illustrated in the following example.

EXAMPLE. Calculate the force on the valve springs of a 3 in. bore compressor so that it will discharge at 160 lb./in.^{2}. Valve diameter 1·2 in.

SOLUTION. The pressure of 160 lb./in.^{2} acts upon the valve area and is resisted by the force of the spring.

$$\text{Valve area} = \frac{\pi}{4}(1\cdot2)^2$$
$$= 1\cdot13 \text{ in.}^2$$
$$\therefore \text{Total force on valve} = 160 \times 1\cdot13$$
$$= \underline{181 \text{ lb.}} \quad \textit{Answer.}$$

This is the force which must be exerted by the valve springs. If the spring, when fitted, is compressed $1\frac{1}{4}$ in. to give a force of 181 lb., its original strength will be $\frac{181}{1\frac{1}{4}} = 145$ lb./in. compression.

The dimensions of a valve spring to give a required strength can be calculated from the formula for deflection

$$\delta = \frac{64WR^3n}{Cd^4}.$$

where δ = deflection, i.e. extension or compression of the spring

W = load on spring, lb.

n = number of coils

R = radius of coil, in.

d = diameter of wire, in.

C = modulus of rigidity = 12,000,000 lb./in.^{2}

EXAMPLE. Calculate the diameter of wire for a spring with 6 coils of $\frac{3}{4}$ in. radius, using wire for which C = 16,000,000 lb./in.^{2} to have a "strength" of 81 lb./in. compression.

SOLUTION. Transposing the formula

$$\delta = \frac{64WR^3n}{Cd^4}$$
$$d = \sqrt[4]{\frac{64WR^3n}{C\delta}}$$
$$= \sqrt[4]{0\cdot000820}$$
$$= \underline{0\cdot162 \text{ in.}} \quad \textit{Answer.}$$

The discharge pressure depends entirely upon the strength of the valve spring. When the force of the vapour is sufficient to lift the valve, this vapour discharges at constant pressure.

5. **Clearance.** To discharge the cylinder completely of vapour is a practical impossibility. At the end of the stroke, no matter how well the compressor has been built, a small volume of high-pressure vapour remains. This is known as the "clearance volume."

When the piston moves back on the suction stroke this high-pressure vapour expands down to the suction pressure before the suction-valve can open. The volume occupied by the expanded vapour reduces the volume entering the cylinder so as to reduce the "pumping capacity" of the compressor.

EXAMPLE. A single-acting compressor of stroke volume 1·45 ft.^{3} has a clearance volume of 0·03 ft.^{3}, suction pressure 15 lb./in.^{2}, discharge pressure 240 lb./in.^{2}. Calculate the actual inlet volume per stroke, assuming the re-expansion of vapour follows the law $PV^3 =$ constant.

SOLUTION. The volume of vapour at pressure 240 lb./in.^{2} is 0·03 cu. ft.^{3}.

Let $V_2 =$ volume of vapour after expanding down to 15 lb./in.^{2}.

Since
$$PV^3 = \text{constant}$$
$$240(0{\cdot}03)^3 = 15V_2^3$$
$$\text{Hence } V_2 = 0{\cdot}075 \text{ ft.}^3$$
$$\therefore \text{ Actual inlet volume} = 1{\cdot}45 - 0{\cdot}075$$
$$= \underline{\underline{1{\cdot}375\,ft.^3}} \quad \textit{Answer.}$$

Thus, although the clearance volume is only 2·07 per cent of the stroke volume, the inlet volume is decreased by 5·18 per cent. This may not appear to be very great, but with a large clearance volume the effect is considerable and seriously interferes with the compressor system.

If, therefore, the compressor is working at 50 h.p. then it is exerting this power usefully on only 1·375 ft.^{3}, whereas it could have been exerted on 1·45 ft.^{3}/stroke. This represents a loss of power and as such it must be reduced to a minimum for the efficient operation of the compressor.

Vapour in the clearance volume does do some work in re-expanding, but it is negligible compared with that which must be exerted in its re-compression.

6. Suction and Discharge Valves. Both suction and discharge valves may be of either the poppet or plate-valve type.

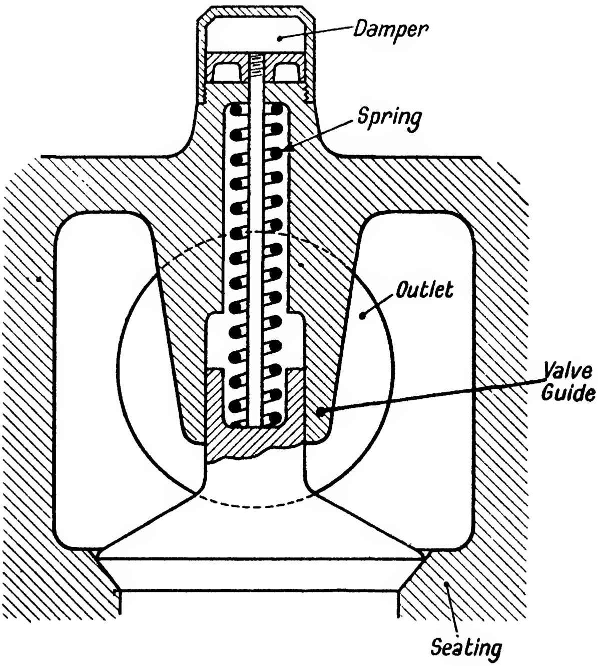

FIG. 12

The poppet valve is designed for a line-contact. It is provided with a damper in the form of an air dash-pot in order to close the valve without shock. Such a valve is shown in Fig. 12.

The plate or rubber valve (Fig. 13) is now used almost universally in place of the poppet valve. Although the strong line contact of the poppet valve is replaced by an overlapping flap, no serious disadvantages have been experienced. This is not so heavy as the poppet valve, and is therefore more sensitive.

Plate valves are particularly suitable for high-speed compressors.

7. Compound Cylinders. When very high pressures are required, several difficulties are encountered with single cylinder compressors. In the first place, the whole construction must be massive to withstand the terrific forces and, secondly, an excessive amount of work is done in compressing the gas. Compound cylinders assist in overcoming these difficulties.

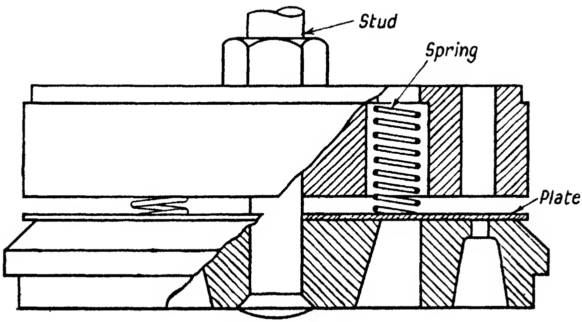

Fig. 13

If the high-pressure expansion is carried out in a small, separate cylinder, this may be suitably reinforced and a low-pressure cylinder designed. By this means, wastage of material in the single cylinder is overcome.

By arranging intercoolers between the various stages of a compound cylinder, deviations from the isothermal curve may be partly controlled. Thus, as shown in Fig. 14, wide variations in pressure occur at high pressures. Vapour is sucked into the first stage and compressed along AB, and it follows the desired isothermal curve very closely. At some intermediate pressure it is discharged into an intercooler BC. The second stage is then entered at a lower temperature and a smaller volume represented by CD. Compression occurs along DE until the final pressure is reached and the vapour at E discharges along EF. This final volume at E reached by the multi-stage compressor is nearer to the isothermal than for the direct compression. Removal of heat by the intercooler reduces the indicator area so that less work need be done in compression. The saving in work is shown by the area *BDEG*.

Intermediate pressure for a two-stage compressor is given by—

$$P_M = \sqrt{P_1 P_2}$$

EXAMPLE. A two-stage compressor, stroke volume 1·9 ft.^{3}, compresses from 16 lb./in.^{2} to 361 lb./in.^{2}, according to the law

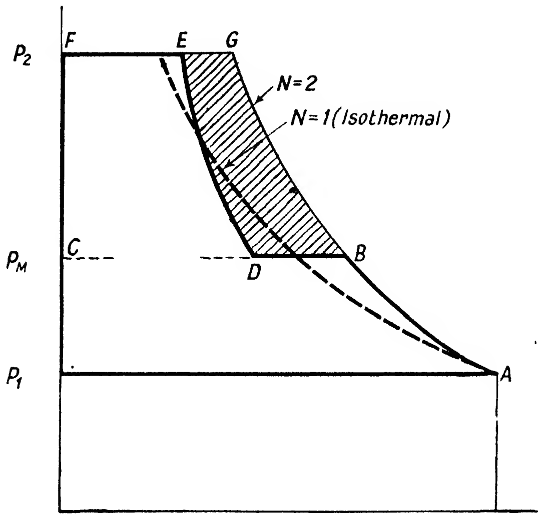

FIG. 14

PV^2 = constant. Suction volume of the second stage is 0·6 ft.^{3}. Calculate the work saved by the compressor.

SOLUTION. From the foregoing formula, the intermediate pressure is

$$P_M = \sqrt{16 \times 361}$$
$$= 76 \text{ lb./in.}^2$$

First stage.

Volume of gas after first stage expansion is given by

$$16 \times (1{\cdot}9)^2 = 76 \times V^2$$
$$V = 0{\cdot}87 \text{ ft.}^3$$

Hence, since the work done per stroke is given by

$$WD = \frac{N}{N-1}(P_1V_1 - P_2V_2)$$
$$= \frac{2}{2-1} \cdot 144(76 \times 0{\cdot}87 - 16 \times 1{\cdot}9)$$
$$= 10{,}290 \text{ ft.-lb.}$$

Second stage.

Volume of gas after second stage expansion is given by

$$76 \times (0{\cdot}6)^2 = 361 \times V^2$$
$$V = 0{\cdot}275 \text{ ft.}^3$$
$$\therefore WD/\text{stroke} = \frac{2}{2-1} \cdot 144(361 \times 0{\cdot}275 - 76 \times 0{\cdot}6)$$
$$= 15{,}400 \text{ ft.-lb.}$$
$$\therefore \text{Total } WD/\text{stroke} = 10{,}290 + 15{,}400$$
$$= 25{,}690 \text{ ft.-lb.}$$

Single stage.

Volume after complete compression is given by

$$16 \times (1{\cdot}9)^2 = 361 \times (V)^2$$
$$V = 0{\cdot}4 \text{ ft.}^3$$
$$\therefore WD/\text{stroke} = \frac{2}{2-1} \cdot 144(361 \times 0{\cdot}4 - 16 \times 1{\cdot}9)$$
$$= 32{,}900 \text{ ft.-lb.}$$

Thus, work saved by compressor

$$= 32{,}900 - 25{,}690$$
$$= \underline{\underline{7210}} \text{ ft.-lb./stroke. } \textit{Answer.}$$

This represents a gain of 21·9 per cent on the single-stage compressor. Intercooling of the intermediate vapour is responsible for this saving in power.

Water is used for the intercooling of large compressors, although some smaller units use air-fin coolers, while with ammonia vapour a bleeding circuit may be used to cool the ammonia, from the plant itself. All ordinary intercoolers are exactly similar to small condensers.

8. Lubrication. Vapour compression involves slightly different lubrication problems from those encountered in air-compressors. Vapour circuits are closed. After operating for some time, oil from the compressor leaks into the circuit. Oil may collect in the condenser and form a "muddy paste"

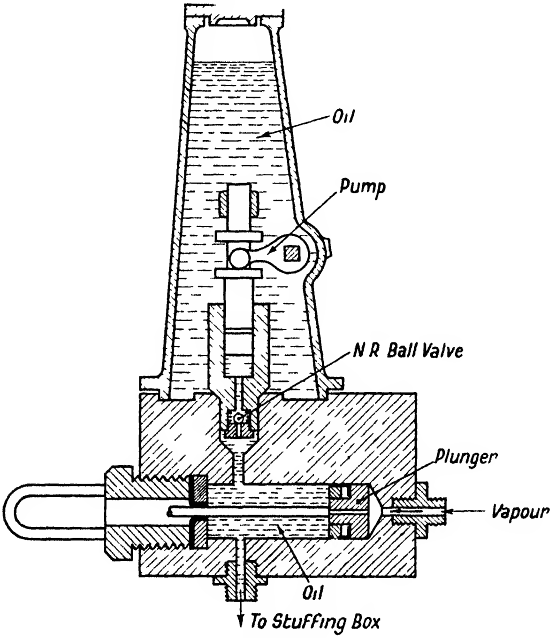

FIG. 15

on the sides of the piping. This paste prevents the condenser from working properly. While it is necessary to use lubricants, only the very minimum is desirable.

Cylinder walls and piston rings are usually lubricated by either a splash lubricator from the crank case as shown in Fig. 9 or by forced lubrication. In the forced lubrication system, oil is pumped at pressure through hollow shafts to the rubbing surfaces, such as the gudgeon pins and bearing brasses. Glands are lubricated by some form of force-pump, such as that shown in Fig. 15. This pressure lubricator is frequently used on semi-automatic machinery. Oil is forced behind the plunger by the force-pump. This oil pressure causes the piston to move against the vapour pressure, and so admit oil through the non-return ball valve. When the plunger barrel is full, the indicator moves right home, as shown. The vapour exerts a pressure on the plunger, which slowly forces oil into the gland. When the barrel is empty, both plunger and indicator are full out. The barrel may be refilled by pumping

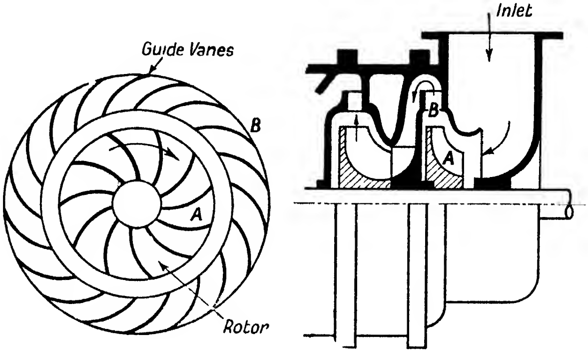

FIG. 16

so that the lubrication is maintained in this semi-automatic way.

If oil does leak into the system it may eventually reach the expansion valve. At this point, the low temperature would cause it to freeze and impede the correct operation of the refrigerator. Consequently, in addition to placing filters before the condenser, it is of the utmost importance to ensure that the system is free from oil at the expansion valve. Large oil separators or filters are therefore frequently incorporated in the system.

9. **Rotary Compressors.** For many purposes, where pressures less than 120 lb./in.^{2} are required, it is possible to use rotary or centrifugal compressors. A section through a large compressor of this type is shown in Fig. 16.

Vapour enters the rotor *A* through an annular passage around the shaft. Rapid rotation of the shaft impresses a centrifugal force on the vapour, which then enters the guide vanes *B*. These guide vanes are designed so as to convert the centrifugal force into pressure and to guide the vapour into the second stage. The vapour receives more centrifugal energy in this stage, which again increases its pressure. The pressure so given to vapour in a rotary compressor is obtained in a number of stages.

Large compressors are designed so that each stage gives a pressure rise of 6 lb./in.^{2}. Thus sixteen stages are necessary to give an overall pressure rise of about 96 lb./in.^{2}, the minimum required for refrigeration. This type of compressor gives the vapour a high velocity, since rotor speeds vary between 600 and 1000 ft./sec. Under such circumstances pressure rises rapidly and the high air velocities produce large frictional losses with the formation of eddy currents and generation of heat.

Heat generated has the effect of lowering the efficiency of this compressor and increasing the compression index. The value of N for such compressor is generally about 1·57 to 1·69. It is almost the same as for adiabatic compression. The ordinary equation for the work done in compressing a vapour according to the law PN^N = constant applies to rotary compressors.

Calculations

1. Calculate the mean piston speed for a compressor with a 3 in. crank turning at 300 r.p.m. [5 ft./sec.]

2. Assuming PV = constant, calculate the suction pressure in a compressor of stroke volume 1·8 ft.^{3}, suction volume 1·6 ft.^{3}. Vapour pressure before suction 14 lb./in.^{2}. [12·45 lb./in.^{2}.]

3. The index of compression for a reciprocating compressor of stroke volume 2·0 ft.^{3} is 1 at 20 r.p.m. and 2 at 500 r.p.m. Calculate the discharge volume at each speed, assuming suction at 12 lb./in.^{2} discharge 150 lb./in. No clearance. [0·16 cu. ft., 0·565 ft.^{3}.]

4. Calculate the compression index for a compressor, suction 12·5 lb./in.^{2}, stroke volume 1·6 ft.^{3}, discharge 80 lb./in.^{2}, discharge volume 0·25 ft.^{3}. [N = 1]

5. Estimate the h.p. of a compressor running at 350 r.p.m., efficiency 80 per cent, compressing from 25 lb./in.^{2}, volume 1·6 ft.^{3} to pressure 200 lb./in.^{2}, volume 0·36 ft.^{3}. N = 1·5. [207.]

CHAPTER IV

CONDENSERS

THE function of a condenser is to remove sufficient heat from the compressed refrigerant to liquefy it Since the refrigerant

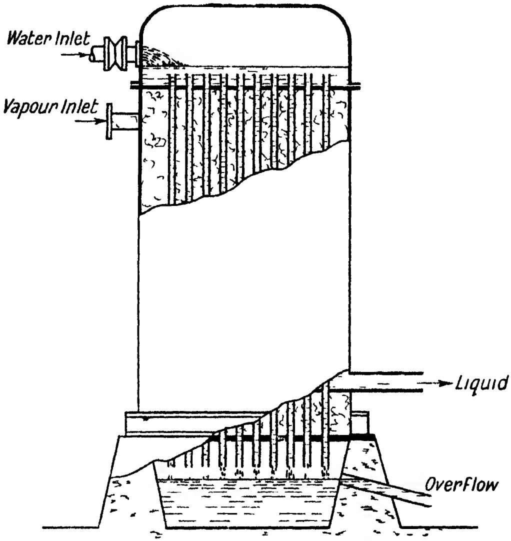

FIG 17

must remain pure, surface condensers alone may be used. These may be classified as follows—

1 Shell and tube condenser
2 Evaporative condenser
3 Double pipe condenser
4 Finned condenser

The type of condenser used depends upon the size and position of the plant

1. Shell and Tube Condenser. This is very similar to the type of condenser in Fig 17 Water passes down through vertical tubes from a header at the top. Vapour enters at a port near the top and liquid refrigerant is tapped from the bottom. After passing through the tube, the cooling water enters a trough, from which the overflow pipes lead off surplus water.

Such condensers as this are easily built in heights ranging from 10 ft. to 16 ft. Their simple construction allows a low

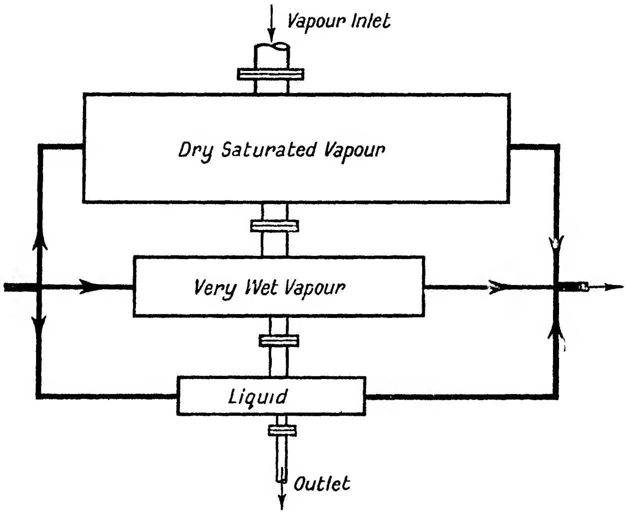

FIG. 18

cost of installation and their design is such that a large condenser capacity may be installed in a small floor space. Among other advantages, this type of condenser is easily cleaned by removing the head.

The horizontal shell and tube condenser consists usually of a bank of single units placed one above the other. Each single unit consists of a shell through which several tubes pass. Condensing water passes through each unit in parallel, so that each receives cold water. Vapour is liquefied by passing through each unit in series so that heat is removed by each pipe bank. The combined series-parallel connections are shown diagrammatically in Fig. 18. It will be seen that the size of the condenser units decrease as the vapour is reduced owing to its decrease in specific volume as latent heat is removed.

EXAMPLE. A shell and tube type condenser is to remove 24,000 B.Th.U./hr. from ammonia at 90° F. with cooling water at 66° F.

Calculate the length of $\frac{1}{2}$ in. diameter tubes necessary, assuming a heat transfer coefficient of 160 B.Th.U./°F./hr./ft.²

Let A = surface area in ft.²

Then, with a temperature difference of 90° F − 66° F. = 24° F.,

$$\text{Heat transferred/hr.} = A\ 24 \cdot 160 = 24{,}000$$

$$A = \underline{\underline{6\cdot 25 \text{ sq. ft.}}}$$

This will be the circumferential area of the pipe.

$$\text{Circumference of pipes} = \pi \cdot (\tfrac{1}{2}) \text{ in.}$$

$$\therefore \qquad = 1\cdot 571 \text{ in.}$$

$$= 0\cdot 131 \text{ ft.}$$

Let L = length of piping

$$\text{Circumferential area} = 6\cdot 25 \text{ ft.}^2$$

$$L = \frac{6\cdot 25}{0\cdot 131} \text{ ft.}$$

$$= 47\cdot 7 \text{ ft.}$$

Thus, assuming the condenser tubes were made 7 ft. long, seven tubes would use a length of 49 ft., which would be suitable for design purposes and allow a suitable tube arrangement.

Evaporative Type of Condenser. Atmospheric cooling is utilized in this type of condenser. The evaporative condenser as shown in Fig. 19 combines the features of a condenser and cooling tower. This type of condenser is used extensively both in refrigeration and air conditioning. A fan draws air over the condenser coils while water is sprayed on to them. This has a dual effect of cooling by the air movement and by evaporating the water spray. An eliminator prevents spray from entering the fan, while excess water falls into the water tank.

Since an evaporative process is used, each pound of water extracts about 1000 B.Th.U. In a water-cooled evaporator, only about 15 to 30 B.Th.U. are extracted per pound of cooling water. The water consumption is therefore much less than for surface condensers.

Atmospheric conditions have a predominating influence upon the performance of such condensers. The temperature and

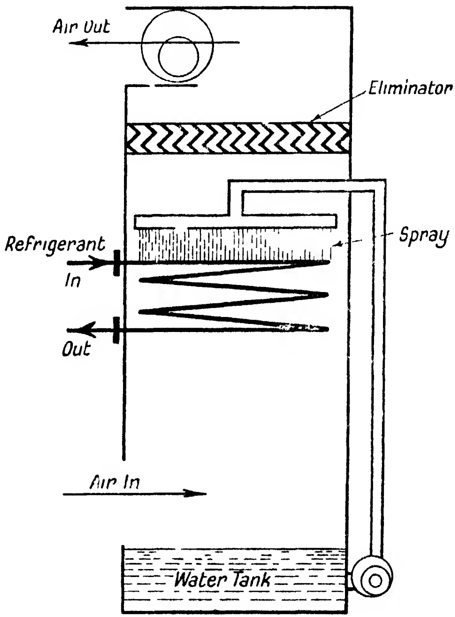

FIG. 19

humidity of the air affect the latent heat of the cooling water and consequently its extraction ratio.

EXAMPLE. Assuming water has a latent heat of 1130 B.Th.U./lb., calculate the flow to condense 3·2 lb. ammonia with a latent heat of 490 B.Th.U./lb. per min. Assuming a condensing surface area of 80 ft.^{2}, calculate the rate of evaporation of water.

Let W = mass of water/min.

$\therefore$ Heat extracted/min. = W. 1130 B.Th.U./min.. . (1)

Since 3·2 lb. of ammonia are liquefied

$$\text{Heat extracted/min.} = 3{\cdot}2 \times 490$$
$$= \underline{\underline{1570}} \quad . \quad . \quad . \quad . \quad . \quad (2)$$
$$\therefore W1130 = 1570$$
$$W = 1{\cdot}39 \text{ lb./min.}$$

The evaporation rate for any area is the number of lb. of liquid evaporated per ft.^{2}/hr.

$$\text{Liquid evaporation from 80 ft.}^2 \text{ in 1 hr.} = 60 \times 1{\cdot}39$$
$$= 83{\cdot}4 \text{ lb.}$$
$$\text{,, \quad ,, \quad ,, \quad 1 ft.}^2 \text{ in 1 hr.} = \frac{83{\cdot}4}{80}$$
$$= 1{\cdot}041 \text{ lb.}$$
$$\text{i.e. Evaporation rate} = \underline{\underline{1{\cdot}041}} \text{ lb./ft.}^2\text{/hr.}$$

This evaporation rate is an important factor in the design of this condenser, and depends not only upon the water temperature, but also the air condition and heat transference rate through the condensing coils.

Double-pipe Condenser. This is a surface type condenser suitable for most medium-sized installations. The ammonia pipe is covered by a concentric water pipe.

Hence transference between the refrigerant and water depends upon the directional flow. With parallel flow, the condensing water becomes heated during its passage along the tubes, and consequently the temperature difference falls, but transference decreases. Counter flow has the opposite effect, and the temperature difference increases throughout the condenser and the heat transference therefore increases.

Finned Condenser. Known more correctly as an extended surface air-cooled condenser; this is used principally with small domestic refrigerators. The fins, which are fitted perpendicularly to the condenser tube, provide a large heat-absorbing area for a minimum cost and space.

The tube material depends upon the refrigerants and upon the possibility of corrosion. Refrigeration equipment generally uses copper owing to its corrosion resistance and the ease of attaching fins. Aluminium is used for ammonia, sulphur dioxide, and freon 12, if there is no danger of exterior surface corrosion.

Fins may also be made from copper or aluminium. Although copper is more expensive and heavier than aluminium, it is easier to bond with the pipe. A satisfactory bond between fins and tube is very essential for the correct transference of heat and liquid features of the refrigerator. Fins may be either soldered to the tube or brazed. Cheaper methods include press fitting of fins to the tubes. Aluminium is rather difficult to either solder or braze, but lends itself to the less reliable method of press fitting.

Hence, transference, in the case of fin condensers, takes place in two stages, air to fin and fin to refrigerator. The air is passed through the condenser at a high velocity by a fan. The fan is generally driven off the compressor shaft and designed as a part of the flywheel.

The transference of heat from the fins to the refrigerant is influenced by the physical properties and velocity of the refrigerant, the surface condition, and the temperature difference. Such conditions are difficult to analyse theoretically.

Experimental results, which are used in the design of finned condensers are similar to those shown in Fig. 20. This figure shows that the tube distribution has an important influence on the transference of heat. Fin spacing also influences this heat transference. Selection of the most suitable tube arrangement and fin spacing is dependent upon characteristics of the design, which is controlled by individual circumstances.

Heat Transference. Only a slight introduction to the complex theory of heat transference and the influence of actual working conditions may be attempted in these paragraphs.

Heat is transferred to a condenser either by radiation, conduction, or convection and by two subsidiary methods of condensation or evaporation.

Radiation is the process by which heat is transferred from a hot radiating body to another absorbent body by a wave motion without heating the intervening space. This occurs in such cases as fin condensers exposed to the sun. Radiation of heat by the sun to the condenser causes a heating of the liquid and prevents efficient condensation. It should therefore be reduced as far as possible. To reduce radiation, the absorptive power of the body, finned condenser in this case, must be decreased. Black bodies absorb more heat than polished, shiny surfaces which reflect radiation.

Finned condensers should therefore be kept in a clean, well-polished condition and preferably plated with a reflective

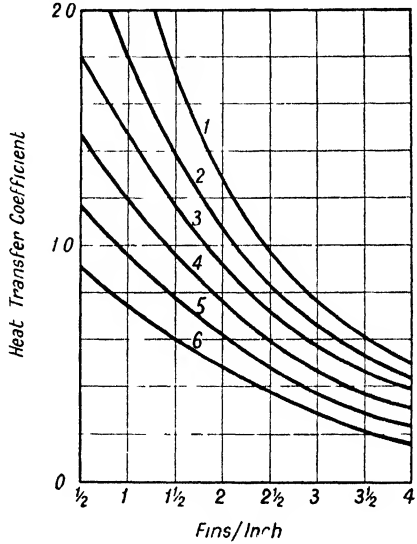

FIG 20

plating such as chromium or nickel. The expense of the plating prohibits its use in most domestic installations and tin plating serves almost as well.

Conduction is the process by which heat travels along a body by heating the intervening particles. This is particularly the case when a poker is kept in the fire; the handle becomes hot by the transference of conducted heat. Conduction has little influence on condensers in cold stores, which are well insulated from the floor by heat-insulating tiles, in order that heat does not leak into the system and reduce its refrigerating capacity.

Convection is the heat transference process in gases and liquids. When a liquid or gas is heated it becomes less dense and rises. This sets up warm currents, which assist in the transference of heat. Similarly, if a liquid is being cooled, cold liquid falls and sets up cold currents which assist condensation of the vapour by bringing the liquid continuously in contact with the cooling surface.

The condensation and evaporation processes occur during

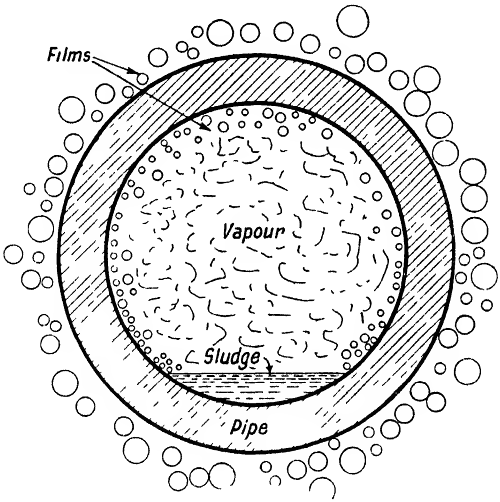

FIG 21

the snowing-up of condensers by the condensation of moisture in the atmosphere of finned condensers, or formations of ice films round the tubes of water condensers. There is removal by latent heat in both processes. The influence of this heat transference is still under investigation, and further results should give great assistance in the design of evaporation condensers.

The transference of heat in condensers depends upon the manner in which refrigerant and cooling mediums flow. Two flows are distinguishable, stream-lined and turbulent. With stream-lined flow the liquid flows with stream line or laminar lines, straight lines without ripples or whirls. Turbulent flow is mixed, with whirls and ripples in confusion. Each flow depends upon the velocity of flow, diameter of pipe, viscosity and density of the liquid, which all affect the manner in which the heat is removed.

Film coefficients are of great importance in the performance of condensers, as shown in Fig. 21.

Gas films form on both the interior and exterior of the pipe and impede the transference of heat by conduction from the

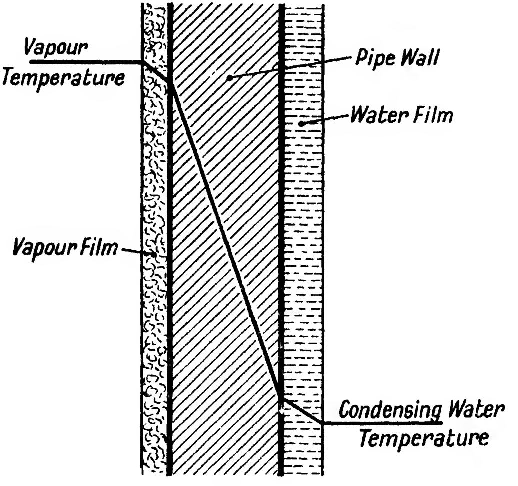

FIG. 22

vapour. Sludge within the pipe has an exceedingly adverse effect in that it reduces the pipe section area, corrodes the pipes, and increases the film thickness. Very little can be done to remove ordinary physical films, but sludge must always be prevented by using suitable filters. The temperature drop between the vapour and the condensing units is shown in Fig. 22, from which it will be seen that gas films create large temperature falls of great importance to heat transference.

Filters. Fine copper gauze is the material most used for the filtration of refrigerants. Vegetable filters such as pads impose too great a resistance to vapour flow for use in refrigerators and are more liable to decompose under the action of many refrigerants.

To facilitate cleaning and repairs, two filters should always be arranged in parallel with appropriate stop cocks. This allows one filter to be used while the other is dismantled. In the design of such a filter, steps must be taken to ensure that it has a sufficiently large capacity to take the complete vapour flow.

Filters should be fitted immediately before the condenser in order to remove sludge which might get into the condenser directly in the liquid state. Oil vapours from the compressor mixed with the refrigerant will not filter, but are combined with the refrigerant. This same sludge must necessarily enter the condenser and will pass to the expansion valve. Instead of vaporizing, sludges freeze on the expansion valve and restrict its operation. A suitable filter must therefore always be placed before the expansion valve.

Where duplication of the filter is undesirable, a blow-off cock is fitted. Since this is on the high-pressure side of the circuit, the vapour will blow out accumulated sludge and clean the filter. This method is frequently used on large industrial installations for which the installation of duplicate filters would be too costly.

Piping. The selection of piping material for condensers depends upon economic factors as well as strength and corrosion considerations.

Steel, either solid drawn or welded, is used most for ordinary condensers. It can be easily heated and bent in a machine without undue difficulty. In addition to this, steel tubes maintain a consistent cheapness and do not corrode over-easily.

Compared with copper, although lacking in certain physical properties, steel condensers are cheaper for large industrial refrigerators. Small domestic units sometimes use copper condensers.

Calculations

1. Calculate the heat extracted by a 6 ft. shell and tube condenser containing 9 tubes, $\frac{5}{8}$ in. diameter, with a heat transfer coefficient of 120 B.Th.U./°F./hr./ft.^{2}, assuming a temperature difference of 17° F. between compressor and condensing unit. [18,000 B.Th.U./hr.]

2. An evaporation condenser uses $1\frac{1}{2}$ lb. water/min. Assuming latent heat of water to be 1070 B.Th.U./lb., calculate the

ammonia flow for the condenser, assuming latent heat of ammonia 475 B.Th.U./lb. [3·38 lb./min.]

3. Compare the heat transference of two fin condensers; one containing 15 ft. of 1 bank tubes and the other with 5 banks of tubing 3 ft. long. Assume $1\frac{1}{2}$ fins/in. with heat transference coefficients of 1·63 and 0·81 B.Th.U./°F./fr./hr. respectively and the same temperature drop across each. [2·01 : 1.]

CHAPTER V

EXPANSION VALVES

THE general principles embodied in the expansion valve have been discussed in previous chapters. Some idea is given here of the features contained in actual expansion valves such as are used in modern refrigerating plants.

In general, units which are assembled always require some kind of adjustment owing to variations which arise during installation and during operation.

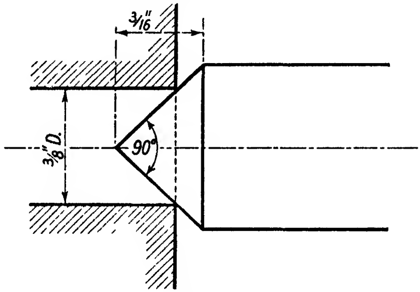

FIG. 23

Hand-adjusted Valves. Hand adjustment is the simplest method of adjusting a valve. It consists of a needle which has a screwed stem and fits into a circular seat. The adjustment depends principally upon the pitch of the thread on the valve. If the load is reasonably constant, this control may be used quite conveniently. Thus this valve may be most suitably employed with large industrial units under the care of skilled operators. It is of little use with small domestic apparatus with rapid changes of load and under the care of unskilled operators.

EXAMPLE. An expansion valve port is $\frac{3}{8}$ in. in diameter and the stem is threaded 32 T.P.I. Calculate the number of turns to be made in opening the valve, assuming it is 90° conical.

For a 90° triangle the length of the two perpendicular sides are equal, hence, in Fig. 23 the apex of the conical valve is

$\frac{3}{16}$ in inside the port. Before the valve is fully open, the apex must be withdrawn this $\frac{3}{16}$ in.

For 1 turn of the valve the apex is withdrawn $\frac{1}{32}$ in.

,, N ,, ,, ,, ,, ,, $\frac{3}{16}$ in.

$$\therefore N = \frac{3}{16} \times \frac{32}{1} = 6$$

This allows quite a good adjustment on this valve and

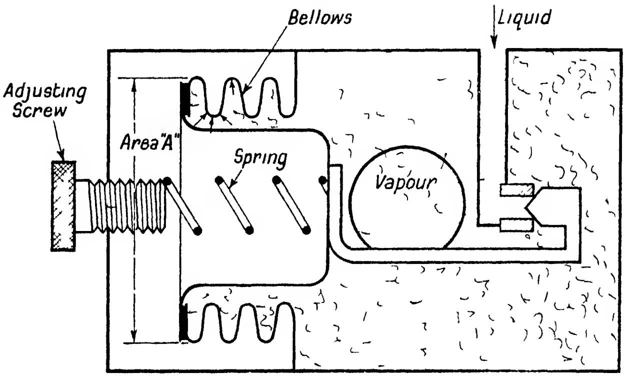

FIG 24

provides for a much larger variation of conditions than is probable in practice.

AUTOMATIC EXPANSION VALVES

For rapid variations of load or for use with unskilled labour, an automatic valve is suitable. There are two principal types of this valve—

1. Constant pressure expansion valve.
2. Thermostatic expansion valve.

1. Constant Pressure Expansion Valve. This maintains a constant pressure on the low side. A pressure-balancing arrangement incorporating either bellows or diaphragm is usual with this valve, bellows being most common.

A bellows valve is shown in Fig. 24. The bellows are operated by the vapour pressure. Any fluctuation produces corresponding variations in the extension, and this causes variations in the valve area, since the valve port is controlled by the needle valve connected to the bellows. Fine adjustment of the needle valve may be made through the adjusting screw.

Decrease in low side pressure causes the bellows to collapse inwards, which allows the valve to uncover more port. The influx of liquid soon increases the low side pressure, and the resultant expansion of the bellows causes the valve to return to equilibrium. Similarly, an increase in pressure causes the valve to close until the pressure decreases when more refrigerant is admitted.

A bellows provides a flexible, pressure sensitive gland element and is designed so that the force produced by the low pressure acting on the large area is just balanced by the high pressure acting on the small valve area.

EXAMPLE. A valve operates under a low side pressure of 20 lb./in.^{2}, and a high side pressure of 160 lb./in.^{2}, through a $\frac{1}{4}$ in. diameter port. Calculate the outside diameter of the low-pressure bellows.

The force produced by 20 lb. acting on the large bellows area shown by A in Fig. 22, is balanced by the force of 160 lb./in.^{2}, acting on a $\frac{1}{4}$ in. circle.

Let D = diameter of bellows.

Force on low-pressure side $= 20(\pi D^2/4)$

Force on high-pressure side $= 160(\pi d^2/4)$

Equation— $$20(\pi D^2/4) = 160(\pi d^2/4)$$

$$D^2 = \tfrac{1}{2}$$

$$\therefore \quad D = \underline{\underline{0{\cdot}707 \text{ in.}}} \quad \textit{Answer.}$$

By calculations of this nature, the dimensions of this valve may be estimated, but applied only to the pressure stated. Variations in condition are adjusted by the spring, which provides an additional force on the low-pressure side.

EXAMPLE. Calculate the spring tension if a valve operates with a low side pressure of 16 lb./in.^{2}. High side 180 lb./in.^{2} through a $\frac{3}{8}$ in. diameter port with bellows 0·75 in. diameter.

Force due to pressure on low side $= 16(\pi D^2/4)$

$= 7$ lb. approx.

Force due to pressure on high side $= 180(\pi d^2/4)$

$= 19{\cdot}9$ lb.

$\therefore$ Spring tension required for equilibrium $= 19{\cdot}9 - 7$ lb.
$= 12{\cdot}9$ lb.

This automatic valve is relatively simple, inexpensive, and reliable. Under correct conditions this valve will operate with entire satisfaction. The application of this valve is restricted,

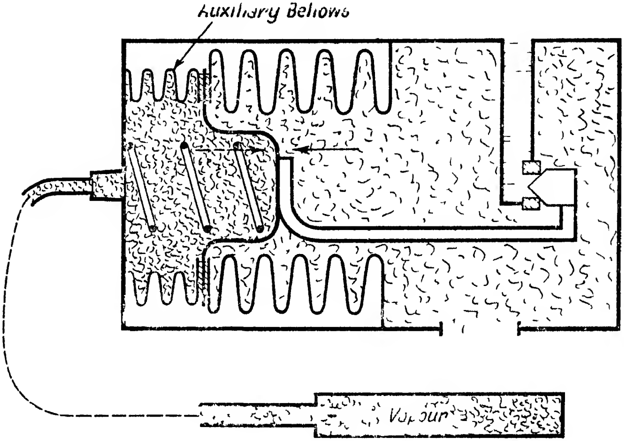

FIG 25

however, by the load condition, for although it will operate satisfactorily under moderate pressure variations, it is less satisfactory over wide ranges. If the load increases abnormally refrigerant evaporates quicker than the compressor can deal with it. This builds a large back-pressure, and consequently the valve closes. At such a time, more refrigerant is required. Restriction of flow by closing the valve impedes the performance of the refrigeration. Similarly, opening of the valve with exceptionally low pressures admits large quantities of liquid refrigerant, which freezes the compressor suction line.

Such disadvantages in the use of this valve have led to its replacement by the thermostat valve.

2. Thermostatic Expansion Valve. As shown in Fig. 25, this is an automatic valve provided with a thermostatic element in place of the spring adjustment. This thermostat allows of immediate control of the valve to suit the actual operating conditions.

Thus, the first bellows element is sensitive to the low side pressure and operates accordingly. The second bellows element is operated by the pressure of the vapour which depends upon the bulk temperature. As the low side pressure acts on the bellows and is constant, then volume increases occur according to the law—

$$\frac{V}{(t + 460)} = \frac{V_1}{(t_1 + 460)}.$$

EXAMPLE. Assuming with a temperature of 40 F. the vapour volume was 3·2 in.^{3}, calculate the volume at 30° F.

$$\frac{3\cdot 2}{(40 + 460)} = \frac{V_1}{(30 + 460)}$$

$$V_1 = 3\cdot 14 \text{ in.}^3$$

If the bellows have a mean constant diameter of 0·7 in., then the alteration in volume from 3·2 to 3·14 in.^{3}, i.e. a contraction of 0·06 in.^{3} is caused by the collapsing of the second bellows.

Let t = movement of bellows, mean diameter 0·7 in.

$$\text{Contraction of volume} = \left(\frac{\pi D^2}{4}\right) t = 0\cdot 06 \text{ in.}^3$$

$$t = \frac{0\cdot 06}{\pi D^2/4}$$

$$= 0\cdot 156 \text{ in.}$$

This calculation gives some approximation of the valve movement which may be expected, and consequently the amount of valve closing which will occur.

The thermostat bulb is clipped to the suction end of the evaporator so that the valve is controlled according to the suction temperature. Variation in this suction temperature causes the vapour to expand and contract, which causes the valve to open or close accordingly. Such control ensures that the evaporator does not freeze up, since cooling causes the valve to close and reduce the admission of liquid.

This valve suffers none of the disadvantages inherent in automatic valves. It is not very expensive to construct and may be made robust without interfering with its sensitivity. For these reasons thermostatic expansion valves are used extensively in many types of refrigerator.

Calculations

1. A hand-controlled expansion valve is $\frac{1}{4}$ in. diameter, with a 90° needle valve. Calculate the ratio of valve area half open and full open. [1 : $\frac{3}{4}$.]

2. A needle valve $\frac{5}{16}$ in. diameter has a cone-shaped head, the apex being $\frac{3}{8}$ in. under the valve. If the spindle is threaded 28 T.P.I., calculate the number of turns required to close the valve from full open. [$10\frac{1}{2}$.]

3. Calculate the equilibrium high side pressure for a constant pressure valve operating on a low side of 12 lb./in.^{2} bellows 1 in. diameter, valve 0·4 in. diameter. [74 lb./in.^{2}]

4. With a spring tension of 1·6 lb./in.^{2} bellows 0·7 in. diameter, high side 110 lb./in.^{2}, valve $\frac{1}{4}$ in. diameter, calculate the low side pressure. [10 lb./in.^{2}]

5. Assuming mean bellows diameter of 0·9 in. vapour volume 2·8 in. at 60° F., calculate the valve movement if temperature falls to 30° F. [0·155 in.]

CHAPTER VI

EVAPORATORS

An evaporator is the unit in which cold, wet vapour absorbs latent heat, transforming it to the saturated state at low temperature. Two main classes of evaporator are commonly used—

1. Brine coolers—indirect system.
2. Fin cooler—direct system.

The former is used principally in connection with large industrial units which can use the storage effect of brine to reduce the operation of the compressor. Fin coolers are used principally with small commercial and domestic refrigerators.

Brine Coolers. Heat exchange between the cold refrigerant and brine cools the brine to a low temperature. This brine is circulated around the stores and thereby cools them. By means of this system it is possible to run the compressors intermittently as described previously in Chapters I and II.

Various designs of brine coolers may be used, although the horizontal shell and tube type similar to Fig. 17 is most usual. The principal consideration in the design and operation of all units associated with interchange of heat is the coefficient of heat transference. Film coefficients are very important and depend upon the physical properties of brine and refrigerant. The refrigerant does not vary much, but brine velocity, density, and temperatures are of great importance. The variations of heat transfer with brine velocity are given in Fig. 26.

In each case the heat transfer coefficients increase with the brine velocity owing to the suppression of convection currents, and consequently the great absorption of heat. A further indication is that reduction in pipe diameter from 2 in. to $1\frac{1}{4}$ in. causes an appreciable increase in the heat coefficient. This is to be anticipated with the formation of fewer convection currents in the smaller tube.

Example. A small horizontal shell and tube evaporator uses 150 ft.^{3} of brine/min. It has 20 2 in. tubes 4 ft. long. Calculate the heat transference, assuming a temperature difference of 12° F. How many lb. of ammonia (latent heat of vaporization 570 B.Th.U./lb.) will this evaporate?

$$\text{Let } V = \text{velocity of brine ft./min.}$$

$$V = \frac{\text{vol. per min.}}{\text{orifice area}}$$

$$= \frac{150 \text{ ft.}^3}{20(\pi R^2) \text{ ft.}}$$

$$= 343 \text{ ft./min.}$$

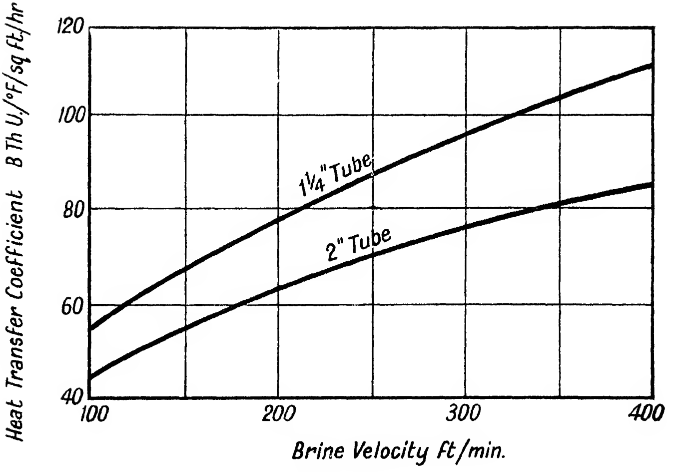

Fig. 26

From Fig. 26, heat transfer coefficient for a 2 in. tube with this flow is 82 B.Th.U./°F./ft.²/hr.

$$\text{Total surface area} = 20 \cdot \pi \left(\frac{2}{12}\right) \cdot 4$$

$$= 41{\cdot}9 \text{ ft.}^2$$

Since there is a temperature difference of 12° F,

$$\text{Heat transferred} = 41{\cdot}9 \times 82 \times 12$$

$$= \underline{\underline{41{,}200}} \text{ B.Th.U./hr.} \quad \textit{Answer.}$$

$$\therefore \text{ Rate of evaporation} = \frac{41{,}200}{570}$$

$$= \underline{\underline{72 \cdot 5}} \text{ lb./hr. } \textit{Answer.}$$

EXAMPLE. A brine cooler consists of 25 tubes $1\frac{1}{4}$ in. diameter and 6 ft. long. Calculate the temperature drop between the brine inlet and brine outlet if specific heat is 0·68, velocity 150 ft./min., density 69 lb./ft.^{3}, assuming a coefficient of heat transfer of 65 B.Th.U./°F./ft.^{2}/hr. Temperature difference refrigerant to brine is constant at 12° F.

$$\text{Surface } A = 25 \,.\, \pi \left(\frac{1\frac{1}{4}}{12}\right) . \, 6 \text{ ft.}^2$$

$$= 49 \text{ ft.}^2$$

$$\text{Heat transferred in 1 hr.} = 49 \times 65 \times 12 \text{ B.Th.U.}$$

$$= 38{,}300 \text{ B.Th.U.}$$

$$\text{Volume of brine flowing/hr.} = 150 \,.\left(\frac{\pi D^2}{144 \cdot 4}\right). \, 60 \text{ ft.}^3$$

$$= 76 \cdot 8 \text{ ft.}^3$$

$$\text{Weight ,, ,, ,, ,,} = 76 \cdot 8 \times 69 \text{ lb.}$$

$$= 5{,}300 \text{ lb.}$$

The transferred heat is lost by the brine. This causes the outlet brine to be lower in temperature than the inlet brine.

Let t = temperature drop outlet to inlet ° F.

$$\therefore \text{ Heat transferred in 1 hr.} = 5300 \,.\, 0 \cdot 68 \,.\, t$$

$$= 3600t$$

$$t = \frac{38{,}300}{3{,}600}$$

$$= 10 \cdot 6° \text{ F.}$$

Note here that a temperature drop exists—

1. Between refrigerant and brine, which is nominally constant at 12° F.

The factors affecting the temperature drop are a feature of the heat transference theory discussed in Chapter IV and shown in Fig. 22.

2. Between brine inlet and brine outlet which is calculated as 10·6° F.

Although the suppression of convection currents assists greatly in transference of heat, whirl strips are very satisfactory in that the spiral motion which they produce brings more liquid molecules in contact with the pipe heat.

Receiving systems of brine coolers include the common shell-and-coil type of cooler. With these, refrigerator coils are submerged in the brine tank while brine flows around them either agitated or in the course of its normal motion. This has the advantage of bulk storage effect, which eliminates the danger of local freezing and blocking up of tubes. It is also convenient to use float-type liquid control with such a system as this, so that the brine level in the tank is kept constant.

The heat transfer coefficient in the receiving system is more uniform than in the horizontal shell and tube type cooler. This is because the brine temperature is nearly constant over the whole coil surface. Variations in diameter have little influence on the heat transfer which is only effected by the brine velocity.

EXAMPLE. Calculate the length of 2 in. diameter tubing required by a shell-and-coil evaporator to evaporate 92·0 lb. ammonia in 1 hr., latent heat 580 B.Th.U./lb. from 0·15 dry to saturated, assuming a coefficient of 120 B.Th.U./°F./hr. and its temperature difference is 9·7° F.

First the surface area of the cooler must be calculated.

For 1 lb. of ammonia—

$$\text{Final latent heat} = 580 \text{ B.Th.U.}$$

$$\text{Initial latent heat} = 0{\cdot}15 \times 580$$
$$= 87 \text{ B.Th.U.}$$

$$\therefore \text{ Heat required to evaporate 1 lb.} = 580 - 87 \text{ B.Th.U.}$$
$$= 493 \text{ B.Th.U.}$$

$$\therefore \text{ Heat required to evaporate 92 lb. ammonia} = 493 \times 92 \text{ B.Th.U.}$$
$$= 45{,}300 \text{ B.Th.U.}$$

Let A = surface area of coils—

Heat transferred through coils/hr. $= 120 \times A \times 9{\cdot}7$

$= 1163\,A$ B.Th.U.

$$\therefore A = \frac{45{,}300}{1163}$$

$$= 39 \text{ ft.}^2$$

Circumference of a 2 in. diameter pipe $= \dfrac{\pi \times 2}{12} = 0{\cdot}524$ ft.

$$\therefore \text{Length of pipe required} = \frac{39{\cdot}0}{0{\cdot}524} = 74{\cdot}5 \text{ ft.}$$

If it is assumed that coils 3 ft. diameter are wound, then the number of coils $= \dfrac{74{\cdot}5}{\pi \cdot 3} = 7{\cdot}9$. Eight coils of a slightly smaller diameter would probably be used.

Fin Evaporators. These extended surface units are used chiefly with small domestic refrigerators. As with the fin cooler, this evaporator consists of plain piping, to which fins are secured, the additional surface provided by the fins assisting in transferring heat.

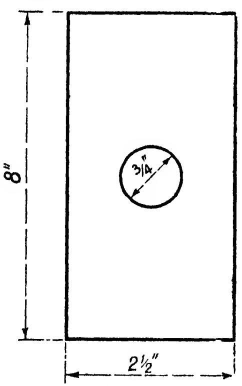

Fig. 27

This unit is of the direct evaporation method, air circulating in the cold store being blown over these fins. Heat is extracted from this air by the evaporator. Thus the store is cooled by the returning air.

Fin evaporators are built in much the same manner as condensers except that they must provide for the transference of more heat at the lower temperature.

Example. Calculate the surface area of 1 ft. finned tubing $\frac{3}{4}$ in. diameter finned 2 fins/in., each fin being 8 in. $\times$ $2\frac{1}{2}$ in.

Every foot of tube will contain $12 \times 2 = 24$ fins. Each fin will present two sides, one at the front, the other at the back, such as shown in Fig. 27. Surface area due to fin thickness may be neglected.

$$\therefore \text{ Area of one side} = (8 \times 2\tfrac{1}{2}) - (\pi \,.\, (\tfrac{3}{4})^2/4)$$
$$= 19{\cdot}56 \text{ in.}^2$$
$$\therefore \text{ Total surface area} = 2 \times 19{\cdot}56 \text{ in.}^2$$
$$= \underline{\underline{39{\cdot}12 \text{ in.}^2}}$$

Thus, in 1 ft. length pipe

$$\text{Fin surface area} = 24 \times 39{\cdot}12$$
$$= \underline{\underline{938{\cdot}88 \text{ in.}^2}}$$

The surface area contributed by the pipe itself will be

$$= 12 \times \pi \,.\, \tfrac{3}{4}$$
$$= \underline{\underline{28{\cdot}25 \text{ in.}^2}}$$
$$\therefore \text{ Total surface area} = 938{\cdot}88 + 28{\cdot}25 \text{ in.}^2$$
$$= \underline{\underline{967{\cdot}13 \text{ in.}^2}}$$

This result indicates that the fin supplies 97·0 per cent and the pipe 3·0 per cent of the total surface area.

Fins increase the bulkiness of the unit, yet provide a greater surface area.

EXAMPLE. Calculate the length of finned tubing required by an evaporator using 1·8 lb. liquid ammonia/per min., latent heat 570 B.Th.U./lb. with a temperature coefficient of 19·6° F. Assuming finned tubing has a surface area of 262·97 in.^{2} and a coefficient of heat transfer of 1·05 B.Th.U./°F./ft.^{2}/hr.

$$\text{Weight of ammonia/hr.} = 1{\cdot}8 \times 60$$
$$= 108 \text{ lb.}$$
$$\therefore \text{ Heat required to evaporate in 1 hr.} = 108 \times 570$$
$$= \underline{\underline{61{,}500 \text{ B.Th.U.}}}$$

Under the existing evaporation condition

$$\text{Heat supplied to evaporate by 1 ft. tubing in 1 hr.} = 262{\cdot}97 \times 1{\cdot}05 \times 19{\cdot}6$$
$$= 5410 \text{ B.Th.U.}$$
$$\therefore \text{ Length of tubing required} = \frac{61{,}500}{5{,}410}$$
$$= \underline{\underline{11{\cdot}35 \text{ ft.}}} \quad \textit{Answer.}$$

At higher pressures, such as exist in the condenser, the latent heat of the ammonia falls and is 455 B.Th.U./lb. at 286 lb./in.^{2} pressure. Thus, with the system outlined in the foregoing example and assuming the same temperature difference, the length of condenser tubing will be 9·06 ft.

If the temperature difference were 49·6° F., as is quite possible in practice owing to the high temperature of compression, the length of cooling tubing would be 3·58 ft.

This shows that the operating condition of the compressor permits the use of smaller units of finned tubing than is permissible with the evaporator.

Calculations

1. A horizontal shell and tube evaporator of surface area 35·0 ft.^{2} temperature difference vapour to brine 13·5° F., heat transfer coefficient 92·7 B.Th.U./°F./ft.^{2}/hr. evaporates 1·75 lb. ammonia per minute. Calculate heat added to each lb. of ammonia. [418 B.Th.U.]

2. Inlet brine 47·1° F., outlet brine 32·7° F., tube surface area 53·4 ft.^{2}, specific heat of brine 0·61, flow 4960 lb./hr. Calculate coefficient of heat transference for the cooler if temperature drop from vapour to brine is constant at 14·8° F. [55·1 B.Th.U./°F./ft./hr.]

3. Calculate the initial dryness of ammonia at 25° F., latent heat 548·9 B.Th.U./lb., if 125·3 lb. are just evaporated per hour in a coil 68·4 ft. long, $1\frac{3}{4}$ in. diameter, with a heat transfer coefficient of 129·6 B.Th.U./°F./ft.^{2}/hr. Assume temperature 38·6° F. [0·199.]

4. Estimate the width of fins spaced 3 per in. on a $\frac{1}{2}$ in. diameter pipe, each of which is $5\frac{1}{2}$ in. long, to give a surface area of 800 in.^{2}/ft. of pipe. [2·265 in.]

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