VAPOUR-COMPRESSION REFRIGERATING MACHINES.*†

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Introduction.—The progress of refrigeration since the last Paper ‡ on the subject was read before this Institution has been enormous. The value of food-stuffs imported into this country, subjected more or less to refrigeration, is now about £130,000,000 per annum, while the application of refrigeration to home produce, to manufactures, to industries, and to the many miscellaneous

* In the Annual Report of the Institution for the year 1906 the Council referred to the question of additional research work, and invited suggestions thereon. In the succeeding Reports (*See Proceedings* 1908, page 86, and 1909, page 218) the Members were informed that inquiries were being made on three subjects with a view to the appointment of Research Committees. Mr. J. Wemyss Anderson was invited to write a preliminary Paper upon the features of Refrigerating Machinery in which further investigation is needed; and the present Paper has therefore been prepared for presentation to the Members, in order that views may be obtained which may be of assistance in the prosecution of the suggested Research.

† In view of some subjects in this Paper having been referred to in the Second French Congress of Refrigeration, September 1912, it is desirable to state that the manuscript of this Paper was received from the author on 2nd February 1912.

‡ "On Refrigerating and Ice-making Machinery and Appliances." By T. B. Lightfoot, *Proceedings, I.Mech.E.*, May 1886, page 201.

purposes, makes the subject one of great importance to scientists and engineers.

The "enormous progress" has, however, been more of a commercial nature than a scientific one; indeed, no branch of mechanical science has received less aid in this country from research or from published accounts of practical progress than that of mechanical refrigeration.

The field for research is very great and includes the machinery proper; the construction, equipment, and insulation of land cold storages; design of ships for carrying refrigerated cargoes; the construction of ammunition magazines (chiefly naval); transportation and storage of perishable produce; the right temperatures for storing various goods; the importance of steady temperatures in fruit carrying and similar cases; the methods of registering temperatures; the condition of the air in cold storages with regard to its temperature, purity (ventilation), humidity, and effect on the goods stored. This list might be extended *ad lib.* if references were made to industries, such as the drying of the air for blast-furnaces, chemical works, soap works, etc.

It is obvious that the actual refrigerating machine demands the first consideration, and of these the principal types in general use are:—

- (1) Vapour-compression machines.
- (2) Ammonia-absorption machines.
- (3) Cold-air machines.

The last named are not now relatively so important as when dealt with by Mr. Lightfoot in 1886.

The ammonia-absorption machine—an important section—can only hope to hold its own against compression machines where waste steam or waste heat in some other form is available.

The first named is, commercially, by far the most important, and on this ground it is proposed to confine this Paper entirely to vapour-compression machines.

Action of Refrigerating Machines.—Heat will naturally flow from a warm to a colder body, but if a quantity of matter is to be

made colder than its surroundings, heat must be removed from it by the expenditure of work. This latter action is the function of a refrigerating machine, which, in consequence, is often spoken of as a "heat-pump." A heat-pump is a reversed heat-engine. The latter receives heat at a high temperature, transforms a certain amount of this heat into mechanical work, and rejects the remainder at a lower temperature.

A heat-pump receives heat at a relatively low temperature, and by the expenditure of mechanical work raises it to a higher temperature, that is, sufficiently high to allow the heat to dissipate.* The total heat so dissipated is the heat removed from the cold body together with the heat equivalent of the work expended in removing or "pumping" it out.

If H_2 = heat extracted ;
 H_1 = heat dissipated ;
 $A W$ = heat equivalent of work expended ;
 then $H_2 + A W = H_1$
 and $A W = H_1 - H_2$ (1)
 so that $H_1 - H_2$ = heat equivalent of work expended.

The ratio of the work expended to the heat extracted is taken as a measure of the efficiency of the machine, but this ratio being generally greater than unity, the term "coefficient of performance" (indicated by η) is used to express the ratio.

Thus for any machine—

$$\text{Coefficient of performance} = \frac{\text{Heat extracted}}{\text{Heat equivalent of work expended}}.$$

Under ideal conditions, from Equation (1)—

$$\eta = \frac{H_2}{H_1 - H_2} \quad . \quad . \quad . \quad (2)$$

* In 1852 the late Lord Kelvin pointed out that the heat from a reversed heat-engine could be economically employed for warming purposes. In the United States refrigerating machines are now employed for cooling large private houses and other buildings in the summer, and warming them in the winter—efficient ventilation being part of the scheme.

Also under ideal conditions the heat H_2 would all be taken in at the temperature τ_2 , and H_1 all discharged at τ_1 ; hence

$$\frac{H_2}{H_1 - H_2} = \frac{\tau_2}{\tau_1 - \tau_2}$$

and for an ideal machine*—

$$\eta = \frac{\tau_2}{\tau_1 - \tau_2} \quad . \quad . \quad . \quad (3)$$

Cycle of Operations in Vapour-Compression Machines.—The heat-engine, which in its reversed action most closely resembles the vapour-compression machine, is the reciprocating condensing steam-engine, and it will be an advantage to consider this well-known cycle first.

* An asterisk indicates that the subject is further treated or demonstrated in an Appendix.

The boiler A, Fig. 1, is supplied with heat and steam generated at τ_1 and p_1 .

This steam does work in the cylinder B, part of its heat being converted into work, the steam afterwards being condensed (that is, a certain amount of heat is lost or dissipated) in the condenser C at τ_2 and p_2 . The condensed water is forced back into the boiler by the feed-pump D—this feed-pump calling for an expenditure of work. It is an advantage to keep this feed-water as high in temperature as possible.

In the vapour-compression machine a liquid (refrigerant) is vaporized in the evaporator C, at a pressure p_2 , by the supply of a quantity of heat from the body (or fluid) to be cooled at temperature τ_2 . The vapour so formed is drawn into the compressor B, at a pressure p_2 , and then, by virtue of work expended, forced into the condenser A, where it is liquefied (generally by the agency of water) at τ_1 and p_1 . By means of a regulating valve D the liquid at p_1 is allowed to pass in regulated quantities into the evaporator C at p_2 , for re-evaporation, Fig. 1 or Fig. 2 (page 957). This liquid should be kept as low in temperature as possible.

In an ideal vapour machine the regulating valve D would be replaced by an expansion* cylinder, in which the liquid on passing from the higher to the lower pressure would do work. This, it will be seen, would give a closer analogy to a reversed steam-engine, as it would as nearly as possible reverse the action of the feed-pump.

In this cycle of operations there are four thermodynamic losses, namely:—

(a) A loss due to the free expansion at the regulating valve; this operation, being non-reversible, renders the entire cycle non-reversible, with a corresponding thermodynamic loss.

(b) The refrigerant in the compressor becomes superheated, and this superheat has to be given to the condensing water by a non-reversible operation, the temperatures of the refrigerant and the water being different, resulting in a further thermodynamic loss.

(c) To facilitate the transfer of heat between the refrigerant

* See footnote on page 952.

and the condensing water, an appreciable difference of temperature is allowed between them, rendering the cycle further non-reversible.

(d) The same applies to the heat transfer between the refrigerant and the brine (or other body to be cooled), supplying the necessary heat for vaporizing the refrigerant in the evaporator.

With regard to the two latter, (c) and (d), it would be fruitless to discuss the losses further, as the differences of temperature are only of a very moderate nature in practice (10° to 20°) and no greater than actually required for practical purposes.

The loss (a) varies with the refrigerant used, and is dealt with in Appendix IV (page 1019), and must be again mentioned in the next section on refrigerants.

Loss (b) is the subject of much controversy, and the author has not been able to find conclusive tests on the relative values of "wet" and "dry" compression.* In this country, where so many ammonia and carbonic anhydride machines are used, a compromise may be said to be the general practice.

The vapour is brought back to the compressor in a somewhat wet state (that is, particles of liquid suspended in the vapour), and the amount of superheat is relatively small, making the use of a water-jacket unnecessary.* In the United States as much "cooling effect" is taken out of the vapour as possible, and it goes to the compressor in a dry or super-dry state, and generally becomes highly superheated on compression, the compressors being water-jacketed, although recent practice points to the adoption of the British method of working, which is known in the United States as the "flooded" system.

Refrigerants.—Thermodynamically, it does not matter what fluid is employed in a heat-engine or reversed heat-engine, yet certain characteristics, physical properties, and practical considerations limit the choice.

Water-steam has never been seriously challenged as the most suitable fluid for engines working on the Rankine cycle. In the

* See footnote on page 952.

vapour-compression refrigerating machines, however, no one fluid holds an unchallenged position—quite a number are employed, of which three are of outstanding importance, namely, anhydrous ammonia (NH_3), carbonic anhydride (CO_2) and sulphurous anhydride (SO_2).

These are able to stand the temperatures and pressures employed (water, for instance, is not available, as it would freeze at the low temperatures required in refrigeration); their capacity for heat is great and they possess low temperature vaporizing points.

Taking -4°F. and $+86^\circ \text{F.}$ as limiting temperatures for the purpose of comparison, the Tables given in Appendix II (pages 1007–9) show that the corresponding pressures for NH_3 are 27 lb. per square inch absolute and 170 lb.; for CO_2 , 288 lb. and 1,038 lb.; and for SO_2 , 9 lb. and 66 lb. per square inch respectively.

None of these pressures present serious difficulties, but in CO_2 machines special care has to be taken, particularly in the compressors, to prevent leakage past the piston and piston-rod and through the stuffing-box or boxes.

In SO_2 machines the working pressure in the evaporator, for ice-making and similar low temperature work, is below atmospheric pressure, and there is a danger of air leaking in, which in itself would eventually seriously affect efficient working. This has rather told against SO_2 for low temperatures, but for the temperatures required in dairy work the evaporator pressure is above that of the atmosphere; under these and in fact under all conditions, SO_2 forms a most efficient refrigerant.*

Ammonia-gas in large quantities is exceedingly noxious, and it is impossible for a man to live in the fumes; hence the Board of Trade will not allow a NH_3 machine in the main engine-room of a steamer.

CO_2 is not poisonous, but it will not support life. It is much heavier than air, and consequently in a closed space would displace the air necessary for life; hence the Board of Trade will allow only small machines in the main engine-room of steamers.

* See footnote on page 952.

Regarding the thermodynamic losses, that at the regulating valve* tells rather heavily against CO_2 — NH_3 and SO_2 being much the same. It is not suggested that an expansion cylinder should be used, but the relative losses must be taken into account when discussing the merits of the three refrigerants. The theoretical loss due to superheating again tells against CO_2 , while SO_2 is rather better than NH_3 . (*See Appendix IV, page 1019.*)

Taking both these losses into account, it would appear that SO_2 is very slightly more efficient than NH_3 , while CO_2 is a very bad third. Here again really reliable comparative tests are wanting, the author having found very conflicting evidence in the published results of actual trials. (*See Appendix VI, page 1028.*)

The practical preference for NH_3 over SO_2 appears to be in the fact that for all ordinary working temperatures the pressures for NH_3 are above the atmosphere, and with any leak the ammonia makes itself felt, while for SO_2 air may leak in without indication, as previously pointed out.

The distinct advantage of CO_2 is that it is practically inodorous. If reliable tests proved CO_2 to be as inefficient as theory points out, the machine would still justify its use on large passenger steamers, but it would be difficult to justify its use on, say, a man-of-war. Such a vessel may live its life without going into action, and supposing an ammonia machine was fitted, it would be quite easy to blow off the charge before coming in touch with the enemy.

Finally, in comparing the three refrigerants, it should be said for a given refrigerating effect* SO_2 is the largest machine and CO_2 the smallest, but the working pressures (and hence thickness of metals) are in the reverse order, so as regards weight and cost there is very little to choose between the three types of machines.

Practical Examples of Compression Machines.—Fig. 2 indicates the principal parts of all vapour-compression machines, viz. :—

(1) Regulating valve, D.	(3) Compressor, B.
(2) Evaporator, C.	(4) Condenser, A.

* See footnote on page 952.

It is proposed to describe these parts in detail. There are, however, other parts and fittings which are of greater or less importance according to the type of machine to which they are fitted. Fig. 2 indicates some of these. E is a *scale-trap* fitted on the suction-pipe between the evaporator and compressor, and so arranged that the vapour has to pass through a gauze-wire pocket

FIG. 2.—Principal Parts of Vapour-Compression Machines.

which effectually traps any scale coming from the long lengths of pipe in use on the machines, and thus prevents scoring of the compressor walls and irregularity in the working of the mushroom valves.

F and G combined form a *rectifier*, the object of which is to catch oil or other fluid passing over from the compressor. F is a cylindrical vessel into which the discharged vapour from the compressor passes by means of an internal pipe, the oil thereby being thrown to the

bottom, the vapour escaping to the condenser by a pipe fitted at the top of the vessel. After a certain period, the length of which would depend on the type and size of machine and on the dimensions of the vessel F, the oil is blown into a second vessel G, after which the connection between F and G is again closed. By a simple arrangement easily followed in the diagram, the vessel is brought under the lowest pressure in the system, and the oil can then be blown off into buckets for re-use.

H indicates a *charging cock*, through which new charges of refrigerant are introduced into the machine. The exact position of this fitting varies with different makers.

Further, a *liquid receiver* is often fitted between the condenser and regulating valve, while in the United States a "drier," consisting of a cylindrical vessel in which quick-lime is placed, is so arranged as to be capable of being brought into the vapour circuit of ammonia machines, the object being to extract any aqueous vapour that might have worked its way into the circuit and which seriously affects the coefficient of performance. The author considers the question of always fitting *driers* to machines of 10 tons ice-making per day and upwards should receive careful consideration.

Regulating Valves.—The amount of liquid refrigerant passing from the condenser to each separate section of the evaporator or evaporators has to be most carefully regulated according to the actual amount of cooling to be performed and the temperature to be carried. The design of the valves necessary for this purpose varies in practice, but in the main they are simple, ingenious and effective.

Fig. 3 shows an ammonia regulating valve which consists essentially of a spindle actuated in the usual way by means of a hand-wheel and plus thread. A part of this spindle is so arranged as to form an ordinary valve, by means of which the liquid can be completely cut off. For regulating purposes the spindle is extended and works as a plug in the body of the valve casting. This plug has a Λ -shaped notch, which not only widens but also deepens from the top downwards. This notch (and the

plus thread) permits of the most minute regulation of the amount of liquid passing. The wheel is often fitted with a finger moving on a dial to indicate the amount the valve is actually open.

FIG. 3.—*Ammonia Expansion Valve. Half size. (Sterne.)*

Evaporators.—Although evaporators differ very much in general form, the principle underlying their action and dominating their construction is the same in all machines, namely, the refrigerant is allowed to vaporize in small (1 inch to $2\frac{1}{2}$ inches diameter) tubes or

pipes, the pressure of vaporization being regulated to correspond to the desired temperature. The exceptions to this general rule as to the size of the pipes or vessels used for evaporators are so few as to hardly merit any further mention. Lap-welded wrought-iron pipes electrically welded or jointed together are generally used, but in CO_2 machines heavy copper pipes are also employed, both in the evaporator and condenser. It should be noted that copper pipes cannot be used with NH_3 .

The heat necessary for the vaporization of the refrigerant is supplied by the medium surrounding the pipes—air or brine in the majority of cases, and water, milk, wort, beer, etc., in special machines. This actual cooling constitutes the refrigerating effect of the machine, and in the case of water or milk cooling (say) concludes the work of the machine. Air and brine, on the other hand, are generally but agents, whereby cooling or freezing can be effected at points more or less remote from the machine.

Fig. 4 shows a CO_2 vertical evaporator, circular in form, with concentric coils made up of pipes $\frac{1}{8}$ -inch bore. This type is common to all vapour-compression machines. As a rule, each coil is separately connected to the inlet header, or manifold at the bottom, and to a corresponding outlet at the top, each coil *either* having the same pitch and different lengths of pipe, *or* the same length with varying pitches. In the present example there are six coils of equal pitch with only two CO_2 inlets and two outlets, these four openings being side by side at the bottom. The first inlet passes direct to the innermost or first coil, and on reaching the top is taken to form the second coil, which on reaching the bottom forms the third coil, and this at the top is continued as the fourth, the bottom of which forms the first outlet or suction to the compressor. The second inlet connects direct to the fifth coil, which passes over at the top to the sixth or outer coil, which at the bottom forms the second outlet. The mild-steel shell is made in halves, which are bolted together as indicated. This construction facilitates erection, inspection, and cleaning.

The brine inlet is at the top and the outlet diametrically opposite at the bottom, thus ensuring efficient circulation—a most important matter, considering the difference of temperatures

FIG. 4.
Carbonic Anhydride
Evaporator.
(Haslam.)

between the refrigerant and the brine is not likely to exceed 20° F. Evaporators are often fitted with agitating paddles (*see* Fig. 25, page 989) to keep up an effective "scrubbing" action between the brine and the walls of the pipes. A cover fitted with a relief-valve is shown, as the evaporator, Fig. 4, is intended for marine use, and therefore a closed brine circuit is desirable. The coils are carefully stayed in position by plates and clips. This type of evaporator is also extensively used for land purposes, the cooled brine being afterwards circulated either in pipes placed in the chambers of the cold stores or through ice-tanks for ice-making.

Evaporators take varying forms to suit particular requirements, and some of these forms will be described later in the Paper under "General Arrangements" (page 981).

Condensers.—Condensers for vapour-compression machines may be divided into three main classes:—

- (1) Submerged;
- (2) Atmospheric;
- (3) Double-pipe.

In all three classes the refrigerant is kept inside small tubes or pipes (seldom exceeding $2\frac{1}{2}$ inches diameter), the condensing medium being on the outside. In this respect the condenser resembles the evaporator—indeed, for marine purposes, connections are often arranged so that the two may be interchangeable.

The submerged condenser is either rectangular or circular in plan. In the former case the pipes (up to $1\frac{1}{2}$ inch diameter), in continuous lengths, are bent backwards and forwards within the given length to form grids or elongated coils. This type is commonly used in marine work, the grids being fitted into box-shaped bed-plates, which carry the engine and compressor. Submerged condensers may also be made of 2-inch or $2\frac{1}{2}$ -inch pipes, screwed and then soldered into return-bends of $3\frac{1}{2}$ -inch or 4-inch centres, the construction of the grids being similar to those in Fig. 5.

The circular form of condenser is similar to the evaporator, Fig. 4, just described. The hot gas enters a manifold at the top and is distributed through concentric coils, the liquid collecting

through a manifold at the bottom. The water enters at the bottom and flows out at the top—the liquid refrigerant being thus cooled to the lowest possible temperature. Theory and practice indicate that the cooling of the liquid refrigerant, after condensation, to the coldest temperature reached in the plant is a distinct advantage—in but few machines, however, are even the most elementary steps taken to secure this advantage. Submerged condensers are only to be recommended when water is plentiful and cheap—as on shipboard.

FIG. 5.—*Contraflow NH₃ Condenser.* (Haslam.)

When water is a consideration, considerable economy can be effected by using an atmospheric condenser, that is, condensers where either the air or water and air combined perform the necessary cooling. They consist of grids of pipes (1-inch or 1½-inch pipes in continuous lengths or 2-inch to 2½-inch pipes built up by return bends) stacked together in vertical sheets. If air is to be the cooling agent, the sheets must be boxed in and air circulated by a fan or fans—a method adopted in the United States. If water is used—the ordinary British method—it is allowed to trickle down the grids into a pan or tank underneath, the coils being protected from the direct rays of the sun (if necessary) by louver boards. The water may run direct to waste or be used over and over again.

At Messrs. Ruddin's Central Cold Stores erected in the heart of Liverpool, the author has found that an exceptionally large tank (22 feet $\times$ 11 feet $\times$ 2.5 feet) for the size of the plant has resulted in a most marked economy—the amount of water used in a week not exceeding that generally required per hour—the reasons being the utilization of rain-water and the natural cooling of the water in the tank during the night.

Fig. 5 shows an atmospheric contraflow ammonia condenser. The vapour from the compressor is carried to a tee or manifold at the bottom and there distributed to vertical grids. Water falls on the outside and consequently the warmer water meets the warmer vapour. Liquid will, of course, be formed at varying heights, and small pipes are fitted at different points in the grid to carry off the liquid as soon as formed to the liquid tee or manifold.

Fig. 6 illustrates a special type of condenser that is rapidly finding favour where conditions are not suitable for either atmospheric or submerged condensers. It is known as the "double-pipe" condenser, and in this example $1\frac{1}{4}$ -inch water-pipes run through 2-inch ammonia-pipes—the details of construction can be readily followed from the drawing. The hot vapour is brought in at the top and the cold water at the bottom, the whole forming an effective and efficient condenser.

It must be here mentioned that exactly similar constructions are being utilized as evaporators or "brine coolers" and even triple pipes are employed—the refrigerant passing through the annular space so formed. Particular forms of condensers will be mentioned later when describing "General Arrangements."

Compressors.—Compressors may be divided into two main classes, namely :—

- (1) Double-acting.
- (2) Single-acting.

These two classes may be again sub-divided into two sections, namely :—

- (i) Horizontal.
- (ii) Vertical.

FIG. 6.—*Double-Pipe NH₃ Condenser.* (Haslam.)

Broadly speaking, horizontal compressors are double-acting and the vertical, single-acting.

A fairly uniform piston-speed is maintained in all cases, and consequent on the fact that horizontal machines have a longer

stroke (compared to diameter) than the vertical ones, it follows that the latter type run at a greater number of revolutions, and in certain makes, quite a high speed (revolutions) is maintained.

Apart from purely mechanical considerations, one point in the

design of a compressor stands out as all important, that is, the clearance between the piston and covers at each end of the stroke must be a minimum—the reason being that the compressor should, if possible, draw on the suction-valve immediately the return stroke begins. The vapour that remains trapped in the clearance space is at the superior pressure of the system, and the piston must travel a very definite distance before this vapour can expand below the suction or inferior pressure, and so allow the fresh charge to be admitted.

Apart from making the clearance as small as possible and keeping the bearings and all other parts mechanically sound, nothing further is done as a general rule. An outstanding exception

FIG. 8.—*Section of NH₃ Compressor. (Haslam.)*

is the "De la Vergne" system, where vertical compressors are used, and the top of the piston carries a layer of oil sufficiently deep to ensure that the whole of the vapour (ammonia) is forced out at the end of the stroke.

Fig. 7 is a 10-inch diameter by 18-inch stroke horizontal double-acting belt-driven ammonia compressor. The main casting is a simple one, being a plain cylinder provided with feet or lugs for bolting to the bed-plate, no jacket or liner being fitted. The covers carry the suction and discharge valves and the cover at the front also carries the stuffing-box. The valves are arranged so as to be easily accessible, and on the removal of a cover a sleeve can be withdrawn containing the whole of the valve and

FIG. 9.—Arrangement of Compressor Cylinders, Piston and Valves for 12" × 21" Triplex NH_3 Compressor.
(The L. R. Co.)

its parts. The machine is designed to run at 80 revolutions per minute, and the scale will allow the leading sizes and the floor space occupied to be determined. Photos of similar compressors are shown on Plate 42.

Fig. 8 shows the type adopted by Messrs. Haslam (Derby) for all their ammonia compressors. In general outline it is very similar to Fig. 7, and, taken with Fig. 9, it illustrates the degree of standardization which has been reached in horizontal ammonia compressors. The valves, however, differ in important detail, and, together with the stuffing-box, will amply repay careful examination.

Fig. 9 is a 12-inch diameter by 21-inch stroke horizontal double-acting ammonia compressor. The main difference between this Figure and Figs. 7 and 8 lies in the liner, which is fitted and kept in position by the covers. This arrangement has all the advantages generally claimed for liners, and the claim that so simple a casting can be made with a specially close-grained cast-iron is one of great importance, inasmuch as ammonia readily finds out any slightly extra porosity. The need of reducing the clearance space to a minimum has already been emphasized, and, as the suction valves open inwards, particular care is taken that these valves are well guided so as to prevent any possibility of "sticking." It should be further noted that the weakest part of the suction-valve spindle is at the end in the thread under the nut controlling the spring. If the spindle should break at this point, a nut lower down is so arranged to prevent the valve dropping into the compressor.

The lift of the discharge-valve is regulated by a stop kept in position by a strong spring. This serves as a buffer and reduces the shock or jar, and further, on occasion, will allow the valve an increased lift should an undue pressure (say from liquid ammonia) be set up at the end of the stroke. The stuffing-box is arranged for metallic packing, and in the main box will be noticed a sleeve designed to leave an annular space between the rod and the box. This space is connected (*see* Fig. 11, page 971) with the suction side of the compressor and fulfils two functions—first, it takes back any high-pressure ammonia that might have so far leaked past, and, secondly,

it provides that the remainder of the box shall only be under suction pressure. This ingenious and simple device is adopted by most makers.

Fig. 10 shows an elevation of Fig. 9—three such compressors being driven direct from the tail-rods of the cylinders of a triple-expansion engine—the whole forming a powerful and compact installation for shipboard. A photo of the machine is given on Plate 43.

Extra precautions have to be taken in refrigeration to ensure (particularly in marine work), as far as possible, immunity from complete breakdown, and in Fig. 10 the possible combinations in case of necessity are too numerous to mention, but it must be stated

FIG. 10.—12" $\times$ 21" *Triplex NH₃ Compressor*. (See also Plate 43.)
(The L. R. Co.)

that any two cylinders of the engine can be worked together compound, and also any single cylinder by itself; further, as long as the crankshaft and rods hold out, any one cylinder will drive any one compressor. Small faces are provided for gauge and similar connections indicated in Fig. 11.

Fig. 11 gives the details of connections similar to those fitted to the compressors given in Figs. 9 and 10, although the actual dimensions are for a longer stroke, namely, 12 inches by 24 inches. The arrangements are so clear in the drawing as to call for little or no comment, but the stuffing-box relief connections should be noted and also the lubricator to the rod, which is so arranged as to keep a space in an outer gland (see Fig. 9) flooded with a special oil, a further small gland

FIG. 11.—Arrangement of Compressor Cylinder. 12" × 24" NH₃. (The L. R. Co.)

FIG. 12.

9'' x 15'' NH_3 Duplex Compressor.
(The L. R. Co.)

FIG. 13.
8" x 12" NH_3 Compressor.
(The L. R. Co.)

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keeping this oil in place. It is sometimes an advantage to so arrange the suction connections, that each end of the compressor (*see also* Fig. 25, page 989) can draw from separate evaporators—it being remembered that temperatures are regulated by suction pressures—and different suction pressures could be carried in the present case by having a separate connection with valve for each suction end. A cross connection with valve would also enable the full compressor to draw from any one evaporator or from both under the same suction pressure.

Fig. 12 shows an arrangement of two 9-inch diameter by 15-inch stroke ammonia compressors by the same firm as the three previous figures. The main difference (apart from the number of compressors) between this arrangement and that shown in Fig. 10, and the reason for its introduction, is that two separate and distinct ammonia condensers—one for each compressor—are placed in the box bed-plate, this being in accordance with ordinary marine practice, a cross connection allowing any one compressor to work with any one condenser. In ordinary working, the two condensers are often worked as one whole. The ammonia and water connections can be readily followed and call for no description. The liquid connection to the evaporator is shown at the bottom left hand of the elevation. A photo of a similar arrangement is given on Plate 43.

Fig. 13 is an 8-inch by 12-inch vertical ammonia compressor. On shipboard, while admitting space is always restricted, yet the greatest restriction lies in the height between decks, and hence, as seen in Figs. 10 and 11, the more powerful marine plants are fitted with horizontal compressors. In certain business premises on land, more particularly in the heart of a city, it is the floor space that is the most valuable, the height being in no case so restricted as on shipboard. In such locations, particularly with a belt drive, a vertical compressor is the better type to employ. In main design the Figure is very similar to the horizontal compressors previously described.

The whole of the working parts are enclosed, and a portion of the crank-chamber to just above the shaft is filled with special (low-point freezing) oil, the remainder of the space being connected to the suction circuit, the usual pressure of which may be taken as

15 lb. per square inch above the atmosphere. The shaft is provided with a stuffing-box, the function of which is to prevent the oil from leaking out, the oil effectually sealing this possible source of leakage from the ammonia gas. The whole arrangement as shown is equivalent to one 8-inch by 12-inch double-acting compressor, but, whereas the double-acting compressor would not likely run above 80 revolutions per minute, this compressor is designed to run

FIG. 14.

4" $\times$ 4" NH_3 Enclosed Compressor.
(Sterne.)

at 120 revolutions per minute. A photo of the machine is shown on Plate 42.

Fig. 14 is a vertical ammonia compressor. The design, although distinctive, follows much the same lines as those previously described, and oil is used as in Fig. 13. The main point of note is the short stroke, which is the same as the diameter, namely, 4 inches. It is frequently found that high-speed shafting or an electric motor happens to be the most convenient source of power, and in such cases a quick-revolution compressor is a distinct

advantage. In the present case the compressor is designed to run at 200 revolutions per minute as a maximum.

Fig. 15 is an SO_2 horizontal double-acting belt-driven compressor, 8 inches diameter by 9 inches stroke. The same main features, as

FIG. 15.

8" $\times$ 9" Sulphurous Anhydride Compressor.
(Douglas.)

previously outlined in ammonia compressors, are to be observed. The construction is simple, the compressor-wall being a liner with a water-circulating space between it and the main body casting. The water is first circulated round the liner and then by means of a

FIG. 16.
Section of $3\frac{1}{4}'' \times 10''$ CO_2 Compressor.
(The L. R. Co.)

small pipe, clearly indicated, is taken to the front cover and circulated round the compressor end of the stuffing-box. This box is of the long single type, and is all that is required for the pressures to be met. The whole design is simple and makes a very effective machine.

Fig. 16 is a $3\frac{1}{4}$ -inch by 10-inch CO_2 compressor. It consists of special cast-iron liner fitting for practically its whole length into a cast-steel body; the end-covers which carry the valves and keep the liner in position are of mild steel, the stuffing-box being bolted on as a separate piece on the front end. The very high condensing pressure of CO_2 (average about 900 to 1,000 lb. per square inch) calls for special care in the stuffing-box, and in the present case it is arranged to supply the centre of the box with oil at a pressure slightly higher than the condensing pressure. Cup leathers (woodite rings) prevent this oil from passing outwards, and it tends to pass into the compressor, the whole design effectively preventing the escape of CO_2 . A special oil-trap (*see* Fig. 18, page 980) is arranged to catch this oil on the discharge pipe.

The great difference between the evaporator (suction) and condenser (discharge or delivery) pressures (say 600 to 700 lb. square inch) means that the greatest care must be exercised in the finish of the bore of the compressor and also in the design and manufacture of the piston. In the present example the piston and piston-rod are turned from a single bar of special steel, the six packing-rings being threaded on in five rings, kept in position by a ring at the end made in halves, the two parts simply slipping into position. The clearance spaces, on account of the position of the valves, cannot relatively be so finely adjusted as in ammonia compressors, but the greater ratio that always exists in CO_2 machines between the diameter and stroke compensates for the clearance loss.

Fig. 17 is a CO_2 compressor, $2\frac{3}{8}$ inches by $7\frac{1}{2}$ inches. In the main this compressor is similar to Fig. 16, and again serves to show how the result of practical experience tends to bring about a standardization of design in machines performing exactly the same class of work. The details of both the stuffing-box and valves differ slightly from Fig. 16, but they are so clearly shown that the construction can be readily followed.

Fig. 18 (page 980) is a general arrangement of what is known as a $4\frac{1}{2}$ -inch by 15-inch CO_2 "duplex machine" for marine purposes, that is, the double-acting compressors, each of the sizes given, are arranged to work as one machine with a common condenser in the box bed-plate, but each compressor can, if desired, work alone with its own separate condenser, while in the case of a breakdown in one or more sections of the machine there are several possible cross combinations.

Although of larger size the views given of the compressors may be taken to supplement those given in Fig. 16, and, together with

FIG. 17.—Section of $2\frac{3}{8}'' \times 7\frac{1}{2}''$ CO_2 Compressor. (Haslam.)

the connections, are so clear that detailed description is unnecessary. The pressure lubricator, however, deserves special mention. It consists essentially of a little plunger-pump. The vapour at the superior pressure from the discharge-pipe is allowed to act on the side of the plunger remote from the rod. On the other side, in the annular space between the rod and the barrel, there is a special oil, which, on account of the difference in area, is forced along a small copper pipe to the stuffing-box, at a pressure slightly higher than the discharge pressure of the machine, the result being, as previously explained, the CO_2 is prevented from leaking outwards, the oil rather leaking in. The condensers, as mentioned, are formed by the box bed-plate and solid-drawn copper pipes made up

FIG. 18.

 $4\frac{1}{2}'' \times 15''$ CO_2 Duplex Compressor. (The L. R. Co.)

into oval coils placed in sets, one inside the other, to economize space, six separate circuits being arranged in each condenser all connecting to headers or manifolds at the top and bottom, the whole making a very compact machine, a photo of which is given on Plate 43.

Vertical CO_2 compressors are also extensively used, but they do not present any distinctive features from those already described—excepting, perhaps, that they are frequently cast in bronze. Photos of vertical CO_2 compressors are given on Plate 44. Some makers

FIG. 19.—Sectional Diagram of a SO_2 Refrigerating Machine for Milk Cooling by means of a Compound Cooler. (Douglas-Conroy.)

cut both horizontal and vertical CO_2 compressors from a solid block of mild steel; an excellent example is shown in the bottom photo, Plate 44.

General Arrangements.—Fig. 2 (page 957), as already described, indicates diagrammatically the cycle of operations in a compression machine; it is now proposed to examine, as typical examples only, a few actual general arrangements. It should be noted that although methods, uses, and details vary greatly, there is no departure whatever from the cycle of the refrigerant.

Fig. 19 and Plate 45 show a milk-cooler. A single-acting, water-jacketed, vertical SO_2 compressor is mounted on the condenser which encloses a single coil of pipe. The vapour from the compressor enters the top of this coil, the resulting liquid passing away from the bottom by an internal vertical pipe to the regulating valve. The condensing water enters at the bottom and leaves at the top. After passing the regulating valve the SO_2 is taken direct to the bottom of an internal coil in the evaporator; the top of this coil is connected with the bottom of a larger coil, and after passing through this to the top the expanded vapour is led by a suction pipe back to the compressor. Brine is circulated round the coils and supplies the heat necessary for the complete evaporation of the SO_2 , the brine being thereby cooled.

Through the top portion of the actual milk-cooler water is circulated, and through the lower portion the cold brine from the evaporator is passed—the water doing the preliminary cooling and the brine the final cooling of the milk. The milk is thus the *heat inlet* to the machine, the complete heat circuit being:—milk to brine in the cooler, brine to SO_2 in the evaporator, SO_2 to the condensing water. The condensing water is therefore the *heat outlet*, and the total heat carried away would be that taken from the milk plus the heat equivalent of work expended in the compressor plus the heat leakage into the machine and its connections. (For the approximate amounts of these three quantities of heat in various sized machines, see Appendix V, page 1027.)

The chief mechanical details can be easily followed from the Figure, but it should be noted that, by an arrangement of two valves and a by-pass overflow pipe, the amount of cold brine flowing through the cooler can be so regulated that the milk runs off the cooler at the desired temperature.

The insulation around the evaporator is to prevent an undue influx of external heat to the brine. A photo. of a complete SO_2 machine for land use is shown on Plate 45.

Fig. 20 is a water-cooling plant for a brewery. Here the ammonia compressor is duplicated as being the part requiring most attention—repairs and re-adjustments—and to avoid, as far

FIG. 20.—Water-Cooling Plant in Brewery. Arrangement of Piping in Refrigerating Engine Room. (Haslam.)

as possible, complete stoppage by breakdown. The evaporator consists of a series of horizontal pipes formed into vertical grids, down which, on the outside, the water to be cooled trickles. The ammonia vaporizes inside these pipes by virtue of the heat supplied by the water, the resultant vapour in the pipes being drawn off by the compressor and discharged into the condenser, which is of the atmospheric type; the resultant liquid

FIG. 21.—General Arrangement of Cold Stores. Cooled by Direct Ammonia Expansion Pipes. (Sterne.)

gravitates back to a liquid receiver. The drawing is very complete, all the connections being shown, and particular attention is again drawn to the separator placed between the compressor and condenser for catching oil passed over with the ammonia vapour. The capacity of the machine (that is, with one compressor working) is 4,400 gallons of water per hour cooled from 60° to 50° F. or half that quantity cooled from 70° to 50° F.

Fig. 21 shows a general arrangement of a cold storage cooled by what is known as the "direct expansion" method. The evaporator

in this case consists of grids of pipes placed directly in the chambers to be cooled, the air supplying the heat (taken from the goods stored) necessary for vaporization. The compressor (shown in detail in Fig. 7, page 966) draws the ammonia vapour back from these grids and discharges it into an atmospheric condenser placed on the roof. The liquid ammonia gravitates into tanks placed in the engine-room, and by means of suitable tee-pieces and a number of regulating valves the liquid is distributed to the direct-expansion pipes in the separate rooms for re-evaporation. The whole forms a most efficient and economical system.

Figs. 22 and 23 (pages 986-7) show a general arrangement of the machinery for an ammonia plant on shipboard. The compressors are directly driven from a steam-engine, the whole standing on a box bed-plate inside of which are placed the grids of pipes forming the condenser (*see* photo, Plate 44). Two distinct types of evaporator are provided in this installation: (1) Circular coils placed in vertical shells for cooling the brine which is circulated through grids of pipes placed in those holds where *chilled* meat or produce is carried; these evaporators are termed "brine coolers" in the drawing. (2) Longitudinal grids of ammonia pipes in which the liquid vaporizes by the heat brought to the grids by the air which is circulated around them by a fan—the air being drawn up from those holds when *frozen* meat or produce is carried and after being cooled is sent down once more, suitable air-suction and delivery ducts being fitted. These evaporators are termed "air-cooling batteries" in the drawing. The evaporators are placed in insulated spaces and even the cold ends of the brine pumps are also encased.

It should be noted that while the air-blast system has much to recommend it, it is impossible by its means to regulate the temperatures carried by this system to the very fine point (one or two degrees variation only) of regulation required for *chilled* produce. Hence the two systems shown in the Figure.

Fig. 24 (page 988) is a general arrangement of a CO₂ plant. The compressors are horizontal and directly driven (as in the previous example) by steam-engines. The great value of this

Figure lies in the fact that it shows all the arrangements that have to be made to allow the withdrawal, inspection, and cleaning of all the coils in the machine. These are all so clearly shown as to require no further explanation.

Fig. 25 (page 989) shows the arrangement of an engine-room in Cold Stores at Wigan. An ammonia compressor 11 inches

FIG. 22.—*Arrangement of Refrigerating Machinery on Shipboard.*
(Haslam.)

diameter by 20 inches stroke is arranged to do all the work required at 60 revolutions per minute, namely, make 12 tons of clear hard ice per day and keep the Cold Stores at the required temperature. A smaller compressor is fitted as a stand-by, the connections (*see* Fig. 25) being so arranged that in case of a breakdown the ice-making is stopped and the Stores kept in condition. Further, the small compressor may be used for winter work on the Stores supposing the ice stock is sufficient to meet the

small demand. Both compressors are separately driven by specially designed electric motors taking current from the Corporation mains; the larger motor is rated for 90 amperes at 460 volts and 300 revolutions per minute, the normal load being 70 to 75 amperes.

FIG. 23.—*Arrangement of Refrigerating Machinery on Shipboard.*
(Plan of Fig. 22.) (Haslam.)

A by-pass connection on each of the compressors enables an easy start to be made by relieving the compression at the end of each stroke for the first minute or so, thus allowing the whole machine to get "under weigh" before the suction valves are opened, thus dispensing with the use of loose pulleys.

This method of starting was first adopted, as far as the author is aware, at Messrs. Ruddin's Central Cold Stores, Liverpool, in 1897, where gas-engines are used, and both at Wigan and Liverpool the result has been most satisfactory and is now commonly adopted.

Fig. 25 also shows an evaporator of the circular type with

FIG. 24.—*Marine Arrangement of CO₂ Refrigerating Machinery.* (Haslam.)

concentric coils. A central agitator, formed by two radial vanes or paddles secured to a vertical shaft and extending practically the whole depth of the evaporator and driven from the counter-shaft by a belt and bevel gearing, is fitted, to keep up a good "scrubbing" action between the brine and the walls of the ammonia coils; the importance of such action, where the difference of temperature between the refrigerant and the brine is small (seldom exceeding

FIG. 25.—Arrangement of NH_3 Refrigerating Machinery to make 10 tons of Ice per Day at Wigan.

20° F.), has already been pointed out. This evaporator cools the brine used for making ice on the cell system. The ammonia suction-pipe from this evaporator leads back to *one end* of the main compressor.

The Stores are cooled by blowing air over a stack of brine-cooled pipes which are placed directly over a rectangular shaped tank containing brine, in which the ammonia coils forming the evaporator are placed. The air is circulated by means of a fan electrically driven; specially constructed ducts deliver, distribute, and return the air to the cooler. The ammonia suction from the Cold Stores evaporator (which is placed on the first floor) is taken back to the *other end* of the main compressor, a connection also being made to both ends of the stand-by compressor. The whole arrangement of ammonia connections should be carefully noted, as it affords a typical example of good practice.

The condenser is of the atmospheric type where the ammonia enters the top of the vertical grid-like coils, a very large tank and centrifugal pump (*see* Fig. 25) ensuring a good and constant circulation of water over the pipes. The tank is placed over out-houses in a position exposed to the sun; the coils are in consequence shielded from the direct rays by louver boards. The connections from the compressors to the condenser and the liquid return to the two regulating valves—one for each of the evaporators—are clearly shown in the Figure. The contractors for the refrigerating plant were The Liverpool Refrigeration Co., and for the electrical power and light installation Messrs. Drake and Gorman, Manchester and London.

Fig. 26 is a sectional profile of one of Messrs. H. and W. Nelson's latest meat-carrying steamers (S.S. "Highland Laddie," and similar boats). The main machinery and boilers are amidships, dividing the insulated (insulation indicated by thick lines) and refrigerated cargo space of about 350,000 cubic feet into two sections—one forward and the other aft. The refrigerating engine-room is aft, and opens out of the main engine-room. The holds are cooled by means of 2-inch diameter galvanized-iron brine-pipes fitted in each hold and between decks—the quantity

FIG. 26.—Diagram showing Meat-carrying Steamer.

FIG. 27.—Diagram Arrangement of Attemperated Brine System on Steamer.

and temperature of the brine for each circuit being regulated from a distributing house. In this installation there are two such houses—one forward and the other aft. These houses are supplied with two distinct feeds of brine—one being at about 5° F., technically known as zero or freezing brine, the other at about

FIG. 28.—Arrangement of Refrigerating Engine Rooms. (Nelson Liners.)

35° to 38° F., known as chilling or attemperated brine. There is still a third circuit known as "thawing-off" brine, which is used *after* all the refrigerated cargo has been taken out, the purposes being, first, to thaw off all the ice and snow collected on the brine-pipes [in the refrigerated spaces, and, secondly, to dry out thoroughly the holds ready for the outward cargo.

Fig. 28 is a plan of the arrangement of the refrigerating-machine room. Three horizontal double-acting ammonia compressors are fitted, a section of one of which is given in Fig. 9 (page 968), each compressor being driven direct off the tail-rods of a triple-expansion engine; the actual arrangement is shown and has been described under Fig. 10 (page 970). The compressors discharge into three condensers, all the connections of which are so arranged that each condenser can be worked, if necessary, quite independently of the other two and with any one compressor. Sea-water circulated from a pump in the main engine-room is used for condensing purposes. The resultant liquid ammonia is taken to a liquid receiver, placed in the insulated chamber (a practice theoretically and practically sound) containing the three evaporators. These are exact duplicates of the condensers, and consist of rectangular shaped mild-steel tanks, each containing six separate circuits of lap-welded wrought-iron coils, connected top and bottom by headers or tees.

Four brine-pumps (the cold parts of which are inside the insulated chamber) are used to circulate the brine, two in general being used for freezing and two for chilling or attemperating brine. A warm brine-pump (thawing-pump), a brine attemperator and a steam-heated warm brine-tank (brine-heater) are also shown in the Figure. The scheme of brine circulation is explained under Fig. 27 (page 991) and Figs. 29 to 32 (pages 994-5).

A small and compact machine is fitted for the provision-room serving the passengers and crew on the outward voyage. This machine consists of an ammonia compressor driven direct by a single steam-cylinder, both of which are mounted on a box bed, one half of which contains the ammonia condenser and the other half the evaporator or brine-cooler, the brine connections being such that the provision-room on the homeward voyage can be cooled by the larger or cargo machine.

Fig. 29 shows diagrammatically the general principle of the brine distribution. It must be again pointed out that chilled meat or produce must be carried at a very steady temperature; and to attain this object the makers in the present case (1) circulate a large quantity of brine just below the desired temperature of the

FIG. 29.—Diagram Arrangement of Attenuated Brine System.
(The L. R. Co.)

FIG. 30.—Main Brine Attenuator.

FIG. 31.—Diagram Arrangement of Attemperated Brine System. (The L. R. Co.)

FIG. 32.
Attemperator Cock.

air in the chamber, rather than circulate a small quantity of brine at a much lower temperature, (2) to enable chilled and frozen produce to be carried economically by one machine without change of system or carrying evaporators working with different temperatures, and therefore with varying suction pressures.

In Fig. 29 the zero or freezing brine-pump draws directly from the evaporator and discharges brine at about 5° F. to a main which is connected with a series of "distributing feed-headers." Each of these headers supplies a number of "brine leads," or separate circuits leading to one hold. A second feed of attemperated brine, at about 35° to 38° F., is also taken to the headers, and a cock (auxiliary attemperator cock) is cross connected between the two feeds, and is so designed as to make it possible to circulate (1) all freezing brine through the leads; (2) all attemperated brine; (3) brine at any temperature between 5° and 38° F. (say). A corresponding cock on the return directs the brine into the main return most suited to its temperature.

The greater portion of the chilling brine flows back to the attemperator, where it is mixed with a little of the freezing brine, and then once more sent back to the holds as "chilling brine." The attemperator (Webb's or other designs) is a simple but effective device for mixing the highest returning brine (say 39° or 40° F.) with the lowest temperature brine (say 5° F.) in order to get a resultant brine of the right temperature (say 35° F.). A simple slide, Fig. 30, operates over two openings, one being the zero brine supply and the other the return attemperated brine, the combined areas of the openings always being a constant; if one is completely shut off the other must be full open. A hand-wheel and thermometer allow a perfect adjustment to be made.

The design of the auxiliary attemperator cock can be followed from the enlarged section in Fig. 32; the thermometer fitted allows each header to be supplied with brine at a pre-arranged temperature. An ordinary cock or valve is fitted to each lead or circuit, so that (1) any circuit can be cut out completely, if necessary; (2) a thermometer being fitted on each return circuit, the separate return cock enables the brine to be so regulated that the quantity best

suiting for the work to be done by that circuit can be accurately adjusted.

The thawing brine arrangements are simple and are easily followed in the diagram. It can readily be understood, however, that instead of having a separate lead, as shown, one of the other mains can be used, inasmuch as no other temperature brine will be in circuit when the thawing-off brine is being used. A further possible arrangement of the auxiliary attemperator cocks is shown in the right-hand portion of Fig. 31, where the cock is placed between the two headers, and the brine lead or circuit is taken directly from the bottom.

The diagrammatical arrangements shown in Figs. 29 and 31 will enable the double set (forward and aft) of duplicated feeds followed on the Nelson boats to be easily understood. The two brine-distributing houses shown on Fig. 26 (page 991) are each supplied with freezing and chilling brine-feeds, as shown on Fig. 27, the main attemperator being fixed in the machine space. Auxiliary attemperator cocks are placed in the distributing houses and cross-connected to the two feeds in exactly the same way as shown in Fig. 29 (page 994). A header is attached to the bottom or outlet from which the separate circuits are fed into the holds. Corresponding return connections are made in the same distributing house, the main returns taking back the brine to the machinery space. The arrangement, which has given great satisfaction, was fitted by Messrs. The Liverpool Refrigeration Co. to the specifications of Mr. A. R. T. Woods, *Member*, the Superintendent Engineer for Messrs. Nelson.

The Rating of Refrigerating Machines.—The generally accepted units for the commercial rating of refrigerating machines are—

- (1) The ice-melting capacity ;
- (2) The ice-making capacity,

both expressed in tons of 2,240 lb. per day of twenty-four hours.

Ice-melting Capacity.—A machine rated as “1 ton ice-melting capacity” would mean that under *assumed* or *named conditions as to range of temperature* the machine would remove the number of

thermal heat-units equivalent to that required to *melt* 1 ton of ice at 32° F. into water at 32° F.

The exact value of this unit has not been agreed upon by refrigerating engineers, the latent heat of water (fusion of ice) being variously taken as 142, 143, 143·7 and 144 B.Th.U. per lb., the possible values for the unit varying between 318,080 and 322,260 B.Th.U., according to the value selected for the latent heat of water. It is obvious, however, that in selecting any figure within the range given no actual unit for the 1-ton ice-melting capacity, or *ton of refrigeration* as it is more usually called, can be accepted which does not fix the temperatures between which the machine is supposed to work. Thus, a machine may be rated as x tons of refrigeration when cooling water or milk and of y tons when making ice, x being greater than y , because the range of temperature in the machine will be less in the former case, and consequently the coefficient of performance, that is, $\frac{\tau_2}{\tau_1 - \tau_2}$, will be greater.

In the United States, 288,000 B.Th.U. per day per ton of refrigeration is the recognized figure, being made up of 144 B.Th.U. per lb. of ice melted and a ton of 2,000 lb. This gives 12,000 B.Th.U. per ton per hour and 200 B.Th.U. per ton per minute, the temperatures selected through which the machine is supposed to work being 90° F. in the condenser and 0° F. in the evaporator, and these would seem suitable temperatures for the States. No definite value has yet been accepted as a British standard, and it is suggested that 322,000 B.Th.U. per ton might be accepted, being the nearest round figure obtained by the product of 143·7 and 2,240, while a reasonable range of temperature for this country would be from 70° F. in the condenser to 0° F. in the evaporator.

Ice-making Capacity.—The ice-making capacity is a measure of the actual weight of ice made by a machine (designed for ice-making) in tons per twenty-four hours. It generally assumes the normal conditions of a British summer, the assumed ranges of temperature, as may be reasonably expected, varying with different makers.

Manufactured ice may be broadly divided into two kinds, *clear* and *opaque*, and these require varying refrigerating effects, but the difference is small compared with the large allowances necessary for the influx of external heat at the ice-tanks, evaporator, and through the pipe connections and the losses due to "thawing off," being as much as 70 per cent. for small machines (say up to 5 tons ice-making per day), and 25 per cent. for machines making 100 tons per day. Taking a typical example: supposing 1 ton of water at 63° F. is to be made into ice at 22° F., then approximately—

2,240 lb. water at 63° F. to water at 32° F.	=	69,440	B.Th.U.
2,240 lb. water at 32° F. to ice at 32° F.	=	321,888	„
2,240 lb. ice at 32° F. to ice at 22° F.	=	11,200	„
Total		<u>402,528</u>	„

This figure, together with the allowances mentioned, would become 700,000 B.Th.U. per ton of ice made for a machine of 5 tons ice-making capacity per day to 510,000 B.Th.U. per ton for a machine making 100 tons per day.

For a machine not designed for ice making, the unit of ice-making capacity would appear of little value, and the ice-melting unit should be used. Approximately the ice-making capacity is half the ice-melting capacity. As a fair practical rule, the author has found that

$$y = 0.6x - 2,$$

where x = ice-melting capacity in tons per day;
 y = ice-making capacity in tons per day.

Conclusions.—British refrigerating machinery by the leading makers is quite in keeping with the best traditions of the British engineer, but it would be idle to contend that the machines and their method of working are not capable of improvement. Vapour-compression machines have the merit of a wonderful simplicity and practically a standardization of design.

The two outstanding wants in refrigeration are:—

(1) A standard unit of refrigeration—say a "ton of refrigeration" of fixed value in B.Th.U., and taken between standard limits of temperature.

(2) A standard refrigerating machine of comparison.

Both these wants could be satisfied by a committee of this Institution in conference with the leading makers.

Careful research is also required to determine:—

(a) The relative efficiencies of the principal refrigerants at varying temperatures.

(b) The advantages and disadvantages of both wet and dry compression.

(c) The value of compound compression.

(d) The effect of varying the amount of refrigerant used in a given machine (undercharged and overcharged machines).

(e) The value of cooling the liquid refrigerant in (or by) the evaporator before expansion.

(f) Deterioration and fatigue of the refrigerant due to constant re-use. The presence of foreign vapours (such as water or oil) in the closed circuit. Value of by-pass driers or scrubbers.

The author's thanks are due to Messrs. L. Sterne and Co., Crown Iron Works, Glasgow; The Liverpool Refrigeration Co., Water Street, Liverpool; The Haslam Foundry and Engineering Co., Derby; and Messrs. Wm. Douglas and Sons, Putney, London, S.W., for supplying those drawings which, in the opinion of the author, best suited the scheme of the Paper, each drawing sent being by the firm at the author's special request. The above firms have also supplied photos which have greatly added to the value of the illustrations.

The Paper is illustrated by Plates 42 to 45 and 32 Figs. in the letterpress, and is accompanied by 6 Appendixes illustrated by 5 Figs.

[The Discussion on this Paper was combined with that on the Paper by Dr. J. H. Grindley, and commences on page 1054.]

APPENDIX I.

NOTATION.

British Standard Units.

- H = quantity of heat, B.Th.U. per lb.
 W = work, foot-pounds.
 J = Joule's equivalent.
 A = reciprocal of J .
 τ = absolute temperature °F.
 t = temperature °F.
 S = sensible heat, B.Th.U. per lb.
 L = latent heat (L_i internal, L_e external), B.Th.U. per lb.
 p = pressure in pounds (absolute).
 v = specific volume of vapour, cubic feet.
 s = specific volume of liquid, cubic feet.
 c = specific heat of liquid.
 K_p = specific heat, constant pressure.
 K_v = specific heat, constant volume.
 E = intrinsic energy.
 x = dryness fraction.
 i = enthalpy.
 ϕ = entropy.
 η = coefficient of performance.

APPENDIX II.

PROPERTIES OF VAPOURS AND REFRIGERANTS.

For a dry saturated vapour the heat, H , necessary to change its state from a liquid at 32° F. into dry saturated vapour at the temperature corresponding to its pressure is

$$H = S + L;$$

also
$$H = A \{E + p(v - s)\}, \quad . \quad . \quad (a)$$

where $p(v - s)$ is the external work, all the units being measured at or from 32° F. ;

further
$$L = L_i + L_e,$$

and
$$L_e = A p(v - s)$$

from (a)
$$E = J H - p(v - s), \quad . \quad . \quad (b)$$

also
$$\begin{aligned} E &= J(S + L_i) \\ &= J(H - L_e). \end{aligned}$$

For wet vapours or mixtures of liquid and vapours, if x is the dryness fraction per unit of total weight,

$$H_1 = S_1 + x L_1,$$

$$H_1 = A \{E_1 + x p(v - s)\}, \quad . \quad . \quad (c)$$

H_1 being the heat required to form wet vapour from liquid at 32° F. , S_1 and L_1 being measured at the saturation temperature.

For superheated vapours the additional heat necessary to raise the temperature from τ_1 (saturation temperature) to τ_3 is

$$K_p(\tau_3 - \tau_1).$$

Hence the total heat H_3 for a superheated vapour is given by

$$H_3 = S_1 + L_1 + K_p(\tau_3 - \tau_1),$$

where
$$S_1 = \int_{491.6}^{\tau_1} c d\tau,$$

c being the specific heat of the liquid.

If K_p is not constant, but has a value that can be given in terms of τ_1 , let $K_p = f(\tau)$, then

$$H_3 = \int_{491.6}^{\tau_1} c d\tau + L_1 + \int_{\tau_1}^{\tau_3} f(\tau) d\tau. \quad . \quad (d)$$

Enthalpy.—If we consider a unit weight of a substance in any condition and write

$$i = A(E + pv), \quad . \quad . \quad . \quad (e)$$

the function i is called the enthalpy of the substance, and is a function of particular value in refrigeration.

Equations (a) and (e) differ only by the small quantity Apv , which can be neglected in practical refrigeration; and in this case the enthalpy can be taken as a measure of the total heat required to form saturated or superheated vapour at constant pressure.

Hence we may write

$$H = A(E + pv),$$

$$dH = A(dE + p dv)$$

$$\text{or} \quad J dH = dE + p dv;$$

$$\text{but} \quad d(pv) = p dv + v dp;$$

$$\begin{aligned} \text{hence} \quad J dH &= dE + d(pv) - v dp \\ &= J di - v dp. \quad . \quad . \quad . \quad (f) \end{aligned}$$

If the pressure is constant during any operation,

$$dp = 0$$

$$\text{and} \quad dH = di;$$

$$\text{hence} \quad i_2 - i_1 = H_2 - H_1, \quad . \quad . \quad . \quad (g)$$

a result which renders a $\tau - i$ or $\phi - i$ diagram particularly useful.

If adiabatic compression is assumed in the compressor, the change of i during the adiabatic compression alone is equal to the work done during the complete cycle of operations in the compressor.

$$\text{For} \quad W = \int_{p_2}^{p_1} v dp,$$

and from equation (f) $J dH = J di - v dp$, since for adiabatic operations $dH = 0$,

$$J di = v dp,$$

$$\text{or} \quad J(i_1 - i_2) = \int_{p_2}^{p_1} v dp;$$

$$\text{hence} \quad W = J(i_1 - i_2), \quad . \quad . \quad . \quad (h)$$

that is, the change of i measured in work units will give the work done in the compressor per unit weight of fluid per cycle, on the

assumption that the compression is adiabatic—an assumption that does not lead to serious error in practical work.

Enthalpy and the Free Expansion Operation.—Consider a unit weight of refrigerant on the condenser side of the regulating valve, under condition p_1, v_1, τ_1, E_1 , no heat is received from or given to outside sources and no external work is done.

In passing the valve, the work done on the liquid is

$$v_1(p_1 - p_2);$$

assuming v_1 constant for liquid,

$$\text{work done by vapour} = p_2(v_2 - v_1);$$

therefore, total work done

$$\begin{aligned} &= p_2(v_2 - v_1) - v_1(p_1 - p_2) \\ &= p_2 v_2 - p_1 v_1. \end{aligned}$$

This work is done at the expense of the internal energy of the fluid.

$$\text{Therefore} \quad E_2 = E_1 - (p_2 v_2 - p_1 v_1),$$

$$E_1 + p_1 v_1 = E_2 + p_2 v_2;$$

$$\text{that is,} \quad i_1 = i_2.$$

The enthalpy of the fluid is, therefore, unchanged by the free expansion operation.

Dry Saturated Vapours.—Experimental results relating to dry saturated vapours lead to the following conclusions:—

(1) The temperature τ at which a liquid vaporizes depends solely on the pressure p to which the liquid is subjected.

(2) The specific volume v of the dry vapour depends on the temperature τ of vaporization.

(3) The sensible heat s and latent heat L each depend on the temperature τ at which the liquid vaporizes.

In general, for any dry saturated vapour, if values for τ, s , and either v or L are determined experimentally, the remaining quantities H, L or v, ϕ and i can be calculated.

Refrigerants.—For the fluids in general use in refrigeration, namely, anhydrous ammonia (NH_3), carbonic anhydride (CO_2), sulphurous anhydride or sulphur di-oxide (SO_2), it cannot be said that the figures available are beyond dispute, while complete $\tau - \phi$, $\tau - i$ and $\phi - i$ diagrams or charts would be of the utmost value.

Anhydrous Ammonia (NH_3).—For the dry vapour state the connection between the pressure and temperature has been determined by Regnault; recent researches, while differing somewhat, tend on the whole to confirm his figures. For temperatures below 32°F ., the values of L and v have been computed only. For temperatures higher than 32°F . the experiments of Dieterici have furnished values of the specific volume v and s of the dry vapour and liquid respectively, enabling L to be calculated and giving for the specific heat of the liquid

$$c = 1.118 + 0.001156 (t - 32),$$

and s can be obtained from

$$s = \int_0^t c \, dt.$$

From c and L the values of ϕ of the vapour at different pressures can be determined.

$$\begin{aligned} \phi &= \int_{\tau_2}^{\tau_1} c \frac{d\tau}{\tau} \\ &= \int_{\tau_2}^{\tau_1} \int (\tau) \frac{d\tau}{\tau}. \quad \dots \quad (h) \end{aligned}$$

If τ_1 is the temperature at which the liquid vaporizes, then the entropy of the liquid at τ_1 is given by the equation (h), writing $\tau = \tau_1$, the resulting entropy being denoted by ϕ_w , being the total entropy in the liquid state.

To change the liquid at constant temperature τ from liquid into dry saturated vapour, the change of ϕ is given by

$$\begin{aligned} \phi &= \int \frac{dH}{\tau} \\ &= \frac{1}{\tau_1} \int dH, \end{aligned}$$

since $\tau = \tau_1$; and $\int dH$ is equal to the whole heat taken in during evaporation, that is, latent heat L , so that

$$\phi = \frac{L_1}{\tau_1},$$

this ϕ being denoted in the Tables by ϕ_s . Hence for a dry saturated vapour its entropy is given by the sum of ϕ_w and ϕ_s .

Properties of NH_3 .

Critical temperature	. . .	266.0° F.
Critical pressure	. . .	1624.0 lb. per sq. inch.
Specific volume of liquid	. . .	0.0256 cubic foot (mean).
Specific heat of liquid	. . .	1.02.
K_p	. . .	0.508.
K_v	. . .	0.393.
γ	. . .	1.29.

Carbonic Anhydride (CO_2).—The properties of this vapour have been difficult to determine for the complete range of temperatures found in refrigeration, for, at the higher temperatures of this range the vapour is near its critical temperature, and throughout the entire range the pressures are relatively very great.

For temperatures above 32° F., Amagat (following others) experimentally determined the density of the dry saturated vapour and of the liquid, his results leading to the values of v and s given in the Table. Amagat also found experimentally the connection between the pressure p and the temperature t in the saturated vapour, and with the previous values of v and s the values of L and ϕ_s have been calculated, leading to values of p , t , L and ϕ_s given in the Table.

The values of the specific heat c of the liquid have not been experimentally determined, but Mollier has computed what may be regarded as reliable values for c , s and ϕ_w .

Properties of CO_2 .

Critical temperature	. . .	88.43° F.
Critical pressure	. . .	1,071 lb. per sq. inch.
Specific heat of liquid	. . .	0.98 (mean).
K_p	. . .	0.217.
K_v	. . .	0.171.
γ	. . .	1.26.

TABLE 1.*—Saturated Vapour of Anhydrous Ammonia (NH₃).
(Dieterici and Wobsa.)

Temperature <i>t</i> Fahr.	Pressure lb. per sq. in. <i>p</i> .	Specific Volume of Vapour <i>v</i> .	Sensible heat of liquid S.	Latent Heat.			Entropy of the Liquid ϕ_w .	$\frac{L}{T}$ ϕ_s .	Temperature <i>t</i> Fahr.
				Total L.	External L _e .	Internal L _i .			
- 22	16.93	15.81	- 58.88	590.3	49.5	540.8	- 0.1265	1.350	- 22
- 13	21.47	12.66	- 49.34	584.0	50.2	533.8	- 0.1048	1.309	- 13
- 4	26.98	10.21	- 39.70	577.3	50.9	526.4	- 0.0835	1.269	- 4
+ 5	33.61	8.30	- 29.88	570.4	51.5	518.9	- 0.0622	1.230	+ 5
14	41.51	6.81	- 20.06	563.5	52.1	511.4	- 0.0414	1.190	14
23	50.82	5.63	- 10.02	555.9	52.5	503.4	- 0.0206	1.152	23
32	61.73	4.66	0	547.9	53.1	494.8	0	1.116	32
41	74.44	3.91	+ 10.15	539.7	53.5	486.2	0.0205	1.079	41
50	89.05	3.29	20.45	531.4	53.8	477.6	0.0405	1.042	50
59	105.80	2.79	30.81	522.2	54.0	468.2	0.0607	1.006	59
68	124.85	2.37	41.34	512.6	54.2	458.4	0.0805	0.971	68
77	146.26	2.04	51.89	502.7	54.3	448.4	0.1003	0.936	77
86	170.53	1.75	62.55	492.4	54.2	438.2	0.1207	0.901	86
95	197.47	1.51	73.42	481.5	54.1	427.4	0.1392	0.869	95
104	227.36	1.30	83.19	471.2	53.7	417.5	0.1583	0.834	104

* Tables 1, 2, and 3 have been compiled by Emile Gouault, Sub-Lieutenant in the French Navy (Ingénieur frigoriste), for the International Association of Refrigeration. (Slightly corrected.)

TABLE 2.

Saturated Vapour of Carbonic Anhydride (CO₂).
(Amagat and Mollier.)

Temperature <i>t</i> Fahr.	Pressure lb. per sq. in. <i>p</i> .	Specific Volume.		Sensible heat of liquid S.	Latent heat.			Entropy of Liquid ϕ_w .	$\frac{L}{T}$ ϕ_s .	Temperature <i>t</i> Fahr.
		Liquid <i>s</i> .	Vapour <i>v</i> .		Total L.	External L _e .	Internal L _i .			
- 22	213·0	0·0155	0·4323	- 25·72	126·13	16·40	109·73	- 0·0533	0·2863	- 22
- 13	248·5	0·0157	0·3674	- 21·87	122·67	16·18	106·49	- 0·0448	0·2748	- 13
- 4	288·3	0·0160	0·3132	- 17·87	118·86	15·88	102·98	- 0·0363	0·2611	- 4
+ 5	333·7	0·0164	0·2674	- 13·73	114·71	15·53	99·18	- 0·0276	0·2470	+ 5
14	384·8	0·0167	0·2286	- 9·38	110·12	15·08	95·04	- 0·0186	0·2326	14
23	440·2	0·0172	0·1952	- 4·82	105·04	14·55	90·49	- 0·0097	0·2170	23
32	502·7	0·0176	0·1669	0·00	99·34	13·89	85·45	0·0000	0·2021	32
41	573·8	0·0182	0·1422	+ 5·17	92·91	13·15	79·76	+ 0·0099	0·1857	41
50	648·9	0·0188	0·1205	10·76	85·54	12·23	73·31	0·0205	0·1679	50
59	732·7	0·0197	0·1010	17·01	76·84	11·09	65·75	0·0321	0·1482	59
68	825·0	0·0210	0·0840	24·21	66·15	9·61	56·54	0·0452	0·1255	68
77	928·7	0·0228	0·0672	33·19	51·51	7·23	44·28	0·0613	0·0968	77
86	1038·0	0·0268	0·0474	47·50	26·88	3·98	22·90	0·0868	0·0493	86
87·8	1060·7	0·0298	0·0412	53·77	15·11	2·22	12·89	0·0981	0·0275	87·8
88·4	1069·8	0·0346	0·0346	61·43	0·00	0·00	0·00	0·1120	0·0000	88·4

TABLE 3.
Saturated Vapour of Sulphurous Anhydride (SO₂).
(Cailletet and Mathias.)

Temperature <i>t</i> Fahr.	Pressure lb. per sq. in. <i>p</i> .	Specific Volume of Vapour <i>v</i> .	Sensible heat S.	Latent heat.			Entropy of Liquid ϕ_w .	$\frac{L}{T}$ ϕ_s .	Temperature <i>t</i> Fahr.
				Total L.	External L _e .	Internal L _i .			
- 22	5.54	13.177	- 16.29	175.99	13.50	162.49	- 0.0351	0.4023	- 22
- 13	7.24	10.307	- 13.72	174.42	13.81	160.61	- 0.0293	0.3907	- 13
- 4	9.23	8.223	- 11.07	172.66	14.04	158.62	- 0.0234	0.3791	- 4
+ 5	11.79	6.668	- 8.39	170.68	14.26	156.42	- 0.0176	0.3675	+ 5
14	14.77	5.290	- 5.65	168.48	14.45	154.03	- 0.0117	0.3559	14
23	18.32	4.328	- 2.84	166.08	14.62	151.46	- 0.0059	0.3443	23
32	22.44	3.574	0.00	163.48	14.76	148.72	0.0000	0.3327	32
41	27.40	2.950	+ 2.90	160.65	14.87	145.78	+ 0.0059	0.3210	41
50	33.23	2.437	5.85	157.61	14.90	142.71	0.0117	0.3094	50
59	39.90	2.036	8.86	155.36	14.94	140.42	0.0176	0.2978	59
68	47.57	1.715	11.92	150.93	14.94	135.99	0.0234	0.2862	68
77	56.23	1.443	15.03	147.28	14.90	132.38	0.0293	0.2746	77
86	66.31	1.218	18.20	143.40	14.81	128.59	0.0351	0.2629	86
95	77.53	1.042	21.42	139.32	14.69	124.63	0.0410	0.2513	95
104	90.17	0.882	24.68	135.05	14.55	120.50	0.0468	0.2397	104

Sulphurous Anhydride (SO_2).—Regnault established the relationship between the temperature and pressure, and furnished the data for p and t given in the Table. The values of v and s have been given by the experiments of Cailletet and Mathias, and those of c by Mathias. The value of c enables s to be calculated, while the values of v and s enable L to be determined.

Knowing S and L , H , ϕ_w and ϕ_s can be found, and in this way the figures given in the following Table have been obtained.

Properties of SO_2 .

Critical temperature	. . .	312.8° F.
Critical pressure	. . .	1,159.6 lb. per sq. inch.
Specific volume of liquid	. . .	0.0112 cubic foot (mean).
Specific heat of liquid	. . .	0.40 (mean).
K_p		0.154.
K_v		0.123.
γ		1.25.

APPENDIX III.

GENERAL THEORY OF COMPRESSION MACHINES.

The function of a refrigerating machine being to pump out or remove heat, the merit or “coefficient of performance” of such a machine is given by the ratio of the heat equivalent of work expended to the heat removed, or

$$\eta = \frac{H_2}{A W} \dots \dots \dots (k)$$

The ideal refrigerating machine or heat pump is a reversed heat engine working in the Carnot cycle. The “coefficient of performance” therefore being

$$\eta = \frac{\tau_2}{\tau_1 - \tau_2}, \dots \dots \dots (l)$$

on a $\tau - \phi$ diagram (Fig. 33)

$$\begin{aligned} \eta &= \frac{\text{Area DCMN}}{\text{Area ABCD}} \\ &= \frac{ND}{DA} \\ &= \frac{\tau_2}{\tau_1 - \tau_2}. \end{aligned}$$

its state at constant pressure from liquid at τ_2 to dry saturated vapour at τ_1 .

Work is now done in a cylinder, expansion being assumed adiabatic down to the temperature τ_2 .

By condensation the vapour can now be brought to a liquid at τ_2, p_2 to its original state at A. In Fig. 34 the area under AB represents the heat absorbed during the operation AB, namely, the increase in the sensible heat of the liquid, so that the area $ABQR = S_1 - S_2$. Also the area under BC gives the latent heat L, taken in during the change of state from B to C, while the area under DA gives the heat rejected to the condenser. The whole heat transferred to work is given by the area ABCD.

Assuming no superheating occurs in the compressor of a refrigerating machine, and following the operations in the reverse order, namely, evaporation AD, adiabatic compression DC, condensation CB, and further cooling to bring the refrigerant to its original state A, the area ABCD now gives the work done by the compressor on the fluid.

If it were a practical proposition to fit an expansion cylinder in place of the free expansion through the regulating valve, the work represented in heat units by the triangular area ABF would be restored by the refrigerant, and the net work done would be given by the area FB CD, which represents a Carnot cycle.

Refrigerating Effect.—In an ideal vapour-compression machine cycle, the refrigerating effect is given by the area under FD, which is equal to

$$\tau_2 \cdot FD = \tau_2 \cdot BC,$$

that is,

$$\tau_2 \frac{L_1}{\tau_1} \quad . \quad . \quad . \quad . \quad (m)$$

With, however, the free expansion, as adopted in practice, the effect is reduced to MDPQ, Fig. 35, where the area FMQR is equal to ABF in the ideal machine.

In practice these areas are only approximately equal, as will be indicated.

The enthalpy is unchanged by the free expansion operation, that is, $i_1 = i_2$, (Appendix II) and further i is *practically* the same as the heat supply at constant pressure, that is,

$$i_1 = S_1 + A p_1 v,$$

for S_1 is the total heat given to the liquid at pressure p_1 , to bring it into the condition shown at B, Fig. 35, at temperature τ_1 from liquid at 32°F .

The cooling of the fluid from B to A is due to a portion of the

liquid vaporizing on passing the regulating valve, the heat required to do this being absorbed from the wet liquid at B, which is thereby cooled to A. As this cooling is done by internal means, it follows that, not only is the *expansion work lost* but its equivalent in cooling has to be supplied from the fluid itself, and the *refrigerating effect is thereby further reduced*. This matter is a very important one in refrigeration and must be further considered, but it can be noted at once that the steeper the line AB the less the loss will be, while the longer the line AD the greater the refrigerating effect. The slope of the line AB depends on the specific heat of the liquid—a

low specific heat meaning a greater slope, while the length of the line A D depends on the latent heat—a high latent heat meaning a long line.

In a good refrigerant, therefore, *the ratio of the latent heat of the liquid to the specific heat of the liquid should be large.*

Further, let x_1 be the dryness fraction of the fluid after passing the regulating valve, then

$$i_2 = (S_2 + x_1 L_2) + A p_2 s,$$

for $S_2 + x_1 L_2$ measures the heat given to the fluid to form vapour of dryness fraction x_1 , under constant pressure p_2 . Hence, equating i_1 and i_2 , we obtain

$$S_1 - S_2 = x_1 L_2 - A (p_1 - p_2) s, \quad \dots \quad (n)$$

or if we neglect the relatively small quantity $A (p_1 - p_2) s$,

$$\text{we get} \quad x_1 = \frac{S_1 - S_2}{L_2}, \quad \dots \quad (p)$$

which gives in Fig. 35 the value of the ratio $\frac{AM}{AE}$, and so fixes the point M.

The approximate equality of the areas A B F and F M Q R can be shown, since the area under A B or A B R N is equal to $S_1 - S_2$, the heat taken in from A to B. The area under A M

$$\begin{aligned} A N \cdot A M &= \tau_2 \frac{x_1 L_2}{\tau_2} \\ &= x_1 L_2, \end{aligned}$$

so that by equation (p) the area A B R N is equal to the area A M Q N. Taking away the common area A F R N, we get

$$A B F = F M Q R.$$

The equality of these areas can only be accepted in approximate calculations, as the area A B F represents the loss of refrigerating effect only on the assumption that

$$A (p_1 - p_2) s$$

is small compared with $S_1 - S_2$.

This assumption is justified when using NH_3 or SO_2 , but the percentage error when dealing with CO_2 cannot be overlooked.

For example, with CO_2 , taking $t_1 = 86^\circ \text{ F.}$ and $t_2 = 14^\circ \text{ F.}$, from the Tables (Table 2, page 1008)

$$S_1 - S_2 = 56.88$$

and

$$A(p_1 - p_2)s = 2.15;$$

an error of 4 per cent. would thus be introduced.

The consequences of the free expansion, to summarize, are twofold:—

(1) The balance of the work done per unit weight of the refrigerant per cycle has been increased.

(2) The refrigerating effect has been diminished by an approximately equal amount.

The consequent loss is not a constant for all refrigerants, and must be taken into account when comparisons are made (Appendix IV).

Coefficient of Performance with Wet Compression.—The coefficient of performance in any assumed case can be obtained as a ratio of two areas on the $\tau - \phi$ diagram, if such be available, or it may be calculated as follows:—

On Fig. 35 the coefficient of performance η is given by the ratio of the area M D P Q to the area A B C D.

Let $x_2 = \frac{AD}{AE}$ be the dryness fraction at the beginning of the compression, then the entropy at D is

$$c \log_e \frac{\tau_2}{491.6} + \frac{x_2 L_2}{\tau_2},$$

and the entropy at C, assuming constant specific heat for the liquid,

$$c \log_e \frac{\tau_1}{491.6} + \frac{L_1}{\tau_1}.$$

But the entropy at C is equal to the entropy at D, therefore

$$c \log_e \frac{\tau_1}{491.6} + \frac{L_1}{\tau_1} = c \log_e \frac{\tau_2}{491.6} + \frac{x_2 L_2}{\tau_2},$$

which gives
$$x = \frac{\tau_2}{L_2} \left\{ c \log_e \frac{\tau_1}{\tau_2} + \frac{L_1}{\tau_1} \right\} \quad . \quad . \quad (q)$$

or, using Tables,
$$x = \frac{\phi_{s_1} + (\phi_{w_1} - \phi_{w_2})}{\phi_{s_2}}.$$

Thus x_2 can be calculated. The value of η can now be determined.

The area A B C D = area under A B C - area under A D

$$= (S_1 - S_2 + L_1) - x_2 L_2,$$

$$\begin{aligned}
 \text{and the area } MDPQ &= \text{area under } AD - \text{area under } AM, \\
 &= x_2 L_2 - \text{area under } AB \\
 &= x_2 L_2 - (S_1 - S_2);
 \end{aligned}$$

$$\text{hence} \quad \eta = \frac{x_2 L_2 - (S_1 - S_2)}{S_1 - S_2 + L_1 - x_2 L_2} \quad (r)$$

With CO_2 it is probably advisable to apply the small correction $-A(p_1 - p_2)s$ to the refrigerating effect.

Within the ordinary working temperatures of refrigerating machines no serious error is introduced by assuming the liquid line AB to be straight, and by the use of Tables the amount of calculation to determine η can be reduced.

Thus, in Fig. 36 (the essential part of Fig. 35),

FIG. 36.

$$\begin{aligned}
 \text{the refrigerating effect} &= \text{area under } MD \\
 &= \text{area under } FD - \text{area under } FM; \\
 \text{area under } FD &= \tau_2 \cdot FD = \tau_2 \cdot BC = \tau_2 \phi_{s_1}; \\
 \text{area under } FM &= \text{area } ABF \\
 &= \frac{1}{2} AF \cdot FB \\
 &= \frac{1}{2} (\phi_{w_1} - \phi_{w_2}) (\tau_1 - \tau_2).
 \end{aligned}$$

$$\begin{aligned}
 \text{Further, the area } ABCD &= \text{the work done in heat-units} \\
 &= ABF + BCFD;
 \end{aligned}$$

$$\text{area } BCFD = \phi_{s_1} (\tau_1 - \tau_2).$$

Thus all the elements necessary to find η are determined.

Example.—In an ammonia compression machine, assuming $t_1 = 68^\circ \text{ F.}$ and $t_2 = 14^\circ \text{ F.}$, to find η with wet compression.

From Table, Appendix I,

$$\begin{aligned}
 \phi_{s_1} &= 0.971, \quad \phi_{w_1} = 0.080, \quad \phi_{w_2} = -0.041, \\
 \tau_1 &= 68 + 460 (\text{approx.}) = 528^\circ, \quad \text{and} \quad \tau_2 = 474^\circ, \\
 \eta &= 8.2.
 \end{aligned}$$

Dry Compression.—If the process of evaporation had been complete before compression commenced, the adiabatic operation would be represented by EG , Fig. 37, and the temperature τ_3 at the end of compression would be found most easily from the diagram, by finding the intersection of the adiabatic EG with the constant pressure (p_1) line $ABCG$. The vapour is then superheated by an amount $\tau_3 - \tau_1$.

The direct effect is to increase the work done by an amount

FIG. 37.

represented by the area below CG as far as the horizontal at E , and to increase the refrigerating effect by the area thence to KP . From a theoretical standpoint, this results in a loss.

Coefficient of Performance using Dry Compression.—The coefficient of performance may be obtained by measurement of areas on the $\tau - \phi$ diagram, if available; otherwise, by calculation, since the area $CGKP$ is equal to $K_p(\tau_3 - \tau_1)$, we require to know τ_3 . This is given directly by the fact that ϕ at G is equal to the ϕ at E ; equating these values, we get

$$4 \text{ A } 2$$

$$\phi_{w_1} + \phi_{s_1} + K_p \log_e \frac{\tau_3}{\tau_1} = \phi_{w_2} + \phi_{s_2}, \quad . \quad (s)$$

which suffices to determine τ_3 if K_p is known and the Tables of ϕ are available.

The area MEKQ, which gives the maximum refrigerating effect possible with dry compression, may be given in the form

$$L_2 - (S_1 - S_2),$$

$$\text{and} \quad \eta = \frac{L_2 - (S_1 - S_2)}{S_1 - S_2 + L_1 - L_2 + K_p (\tau_3 - \tau_1)}, \quad . \quad . \quad (t)$$

an equation of doubtful value unless the quantities involved can be accepted without question.

A further method is to determine the work done AW directly from the $\tau - \phi$ diagram, or, by assuming the liquid line AB to be straight, proceeding as in the following example.

Assuming $t_1 = 68^\circ \text{ F.}$ and $t_2 = 14^\circ \text{ F.}$, find η in an ammonia machine using dry compression.

From Table 1 (page 1007),

$$\begin{array}{ll} \phi_{s_1} = 0.971 & \phi_{w_1} = 0.080 \\ \phi_{s_2} = 1.190 & \phi_{w_2} = -0.041 \end{array}$$

Taking $K_p = 0.508$, we obtain from equation (s) (or from $\tau - \phi$ diagram) that $\tau_3 = 682^\circ \text{ F.}$ (approx.).

Determining the equivalent areas to ABF, BXEF and CGX, in Fig. 37, on a $\tau - \phi$ diagram, we obtain

$$\eta = \frac{518.9}{72.92} = 7.12.$$

APPENDIX IV.

THEORETICAL COMPARISON OF NH_3 , CO_2 , AND SO_2 ,
AS REFRIGERANTS.

Thermodynamic Losses peculiar to the Refrigerants.—In an ideal machine, working in a reversed Carnot cycle, it does not matter, thermodynamically, what substance is used as a refrigerant. It has been shown, in Appendix III, that in a practical machine, even with an expansion cylinder, there is an unavoidable loss represented by the area under A F, Fig. 37 (page 1017). With a regulating valve, such as obtains in practice, the loss is represented by the area under A M, Fig. 37.

This loss varies with different refrigerants, but must be held as telling against them as refrigerants.

The following Table shows the percentage loss of refrigerating effect (that is, $\frac{AM}{AE} \times 100$, in Fig. 37), when the upper and lower temperatures are 68°F . and 14°F . respectively.

Refrigerant.	Latent heat L_2 .	Liquid heat $S_1 - S_2$.	Refrigerating effect.	Percentage loss $\frac{(S_1 - S_2) 100}{L_2}$.
CO_2	110.65	32.08	78.57	29.0
NH_3	577.4	58.50	518.9	10.1
SO_2	168.18	17.27	150.9	10.28

This loss tells very heavily against CO_2 . If we had taken the upper temperature limit at 86°F . instead of 68°F . the comparison would have been still more unfavourable to CO_2 .

Regulating Valve.—The use of the regulating valve in place of an expansion cylinder must now be considered.

Referring to Fig. 37 (page 1017), the area under F E represents

the refrigerating effect with the expansion cylinder, and the area under M E the refrigerating effect when a regulating valve is used.

Area under F E = area under A E - area under A F ;

$$= L_2 - \tau_2 (\phi_{w_1} - \phi_{w_2}) ;$$

area under M E = area under A E - area under A M ;

$$= L_2 - (S_1 - S_2).$$

Using Tables and taking the same temperatures as before, viz. 68° F. and 14° F., the following Table is obtained :—

Refrigerant.	$L_2 - \tau_2$ $(\phi_{w_1} - \phi_{w_2})$.	$L - (S_1 - S_2)$.	Difference.	Percentage difference.
CO ₂	80·44	78·57	1·87	2·33
NH ₃	522·19	518·90	3·29	0·63
SO ₂	151·85	150·91	0·94	0·62

Admitting that the use of the regulating valve is justified for practical reasons, the figures just given would seem to indicate that its use is more than justified, but it must be remembered that the effect of the substitution of the regulating valve for the impossible expansion cylinder on the coefficient of performance is more serious, for the heat representing the loss of refrigerating effect in the last Table is equivalent to the increase in the expenditure of work, since the area F M Q R is equal to the area A B F, Fig. 37 (page 1017).

With $t_1 = 68^\circ \text{ F.}$ and $t_2 = 14^\circ \text{ F.}$, as before :—

Refrigerant.	Percentage loss by free expansion.	Calculated η .	Carnot cycle η .	Percentage loss from Carnot's cycle.
CO ₂	17·4	6·70	8·78	24·7
NH ₃	5·1	7·12	8·78	18·8
SO ₂	5·3	7·47	8·78	14·9

APPENDIX V.

OUTLINE DESIGN OF AN AMMONIA COMPRESSION MACHINE.

Assuming average temperatures of 70° F. and 0° F. for the ammonia in the condenser and evaporator, and allowing for losses at the regulating valve, influx of heat into the pipe connections and cold parts, and by superheating in the compressor, 450 B.Th.U. per pound of ammonia circulated may be taken as available refrigerating effect.

Taking one ton of refrigeration as 322,000 B.Th.U. per day of twenty-four hours, then

$$\frac{322,000}{450} = \text{lb. of ammonia to be circulated per day per ton,}$$

or
$$\frac{322,000}{450 \times 24 \times 60} \text{ lb. per minute.}$$

Taking the volume of 1 lb. of ammonia vapour at 0° F. to be 9.1 cubic feet, then

$$\begin{aligned} \frac{322,000 \times 9.1}{450 \times 24 \times 60} &= \text{cubic feet of ammonia vapour to be} \\ &\quad \text{circulated per minute.} \\ &= 4.53 \text{ cubic feet of compressor} \\ &\quad \text{displacement per minute per ton} \\ &\quad \text{of refrigeration per day.} \end{aligned}$$

This displacement would be required for machines of 2 tons of ice-making capacity per day and under, but a little less than this figure may be allowed for machines making 5 tons of ice per day and over. The following Table (page 1022) may be taken as a typical example of the variations of allowance according to size.

This Table is intended to illustrate commercial conditions, and only 300,000 B.Th.U. is allowed for *effective* ice-melting capacity per ton, the remaining 22,000 B.Th.U. representing a reserve of about 7 per cent.

It should be noted that for a machine making 5 tons of ice per day 700,000 B.Th.U. per ton is allowed, while for 100 tons the

amount is only 510,000 B.Th.U. per ton, which leads to a somewhat corresponding decline in the compressor displacement per minute.

No. of Machine.*	Per day of 24 hours.			Compressor displacement.	
	Ice making.	Ice melting.	Effective B.Th.U. removed.	Rate. Cubic feet per min. per ton refrigeration.	Cubic feet per minute.
	Tons.	Tons.			
1	5	11·7	3,500,000	4·4	51·5
2	10	20·0	6,000,000	4·35	87
3	25	45·0	13,500,000	4·3	193·5
4	50	87·0	26,000,000	4·2	365
5	100	170·0	51,000,000	4·1	697

* For reference in this Appendix only.

Piston Speed.—Owing to the fact that most compressors depend on self-acting mushroom valves for both the suction and delivery, the piston speed is much below ordinary steam-engine practice.

The maximum speed is about 350 feet per minute—this speed being reached both in the case of the quick-revolution, single-acting, vertical compressors, as well as in the slow-running, horizontal, double-acting machines.

The revolutions per minute vary from 45 in the latter case to 200 in the former type.

Relation of Diameter to Stroke.—It is most important that the whole of the contents of the compressor should be swept out at each stroke. Unless oil (De la Vergne and Sterne systems) is used, this is practically impossible, and so, however small the clearance may be, the piston must be well on its return stroke before the entrapped gas can expand to the inferior pressure.

Thus (when oil is not used) the longer the stroke in relation to the diameter for a given capacity, the greater the efficiency.

A long stroke, however, means a long connecting-rod and long bed-plate, with a correspondingly increased cost all round, and so the following is found to be a reasonable ratio :—

$$\frac{l}{d_1} = 2 \text{ to } 2.4,$$

where l = length of stroke in inches,
 d_1 = diameter of compressor in inches.

Thickness of Metal in the Compressor.—The thickness of metal for the compressor walls is determined by the fact that cast-iron (the metal invariably employed) is porous, and 1 inch to $1\frac{1}{8}$ inch is required to ensure good working with ammonia at the pressures generally employed.

With very close-grained iron and plain castings, such as would be used for vertical machines, the metal may be reduced to $\frac{7}{8}$ inch, but it is not advisable to go below this, however small the compressor may be, nor should any cast-iron used in the circuit be thinner than this.

For compressors above 8 inches to 9 inches diameter a liner may be fitted with advantage, and this, if the rule given for the ratio of stroke to diameter be followed, should not be less than $1\frac{1}{8}$ inch, and may, with a compressor 21 inches diameter and over, be made $1\frac{1}{2}$ inch thick, it being remembered that the liner, as a rule, bears only at each end, and consequently must be made sufficiently stiff (see Fig. 9, page 968).

Compressor and other castings may, with advantage, be subjected to a solution of sal-ammoniac (ammonium chloride) under a pressure of 200 lb. per square inch. This has the effect of closing the pores by rusting them up; if the casting has to be machined, the treatment is more effective if done after the first cut has been taken.

Valves.—The diameter of the suction-valve may be made a little *above*, and the delivery valve a little *below*, one-third the diameter of the compressor,

Suction- and Discharge-Pipes.—Little or no difference is made between the diameter of the suction- and discharge-pipes, and they may both follow the rule

$$d_2 = 0.25 d_1,$$

where d_1 is the diameter of the compressor and d_2 the internal diameter of the connecting pipes.

Compressor-Rod.—The rod should be made of piston-rod steel, that is, of tensile strength of 35 to 40 tons per square inch with at least 21 per cent. elongation over 6 inches.

Taking f_t , the allowable stress in the rod, as 2,500 lb. per square inch (to allow for bending, etc.), and an effective pressure of 156 lb. per square inch on the piston, we have

$$\begin{aligned} \frac{\pi}{4} d_2^2 f_t &= \frac{\pi}{4} d_1^2 p \\ d_2 &= d_1 \sqrt{\frac{p}{f}} \\ &= 0.25 d_1, \end{aligned}$$

where

d_1 = diameter of compressor,

d_2 = diameter of rod.

Horse-power required to drive Compressor.—The mean effective pressure for an ammonia compressor working under conditions suited for ice-making may be taken as 60 lb. per sq. inch; from this the probable "indicated horse-power" of the compressor may be computed; by adding 25 per cent., the required i.h.p. of the engine may be found.

The following rule may be followed:—

$$b = 2a + c,$$

where b = i.h.p. of engine;

a = ice-making capacity of machine in tons per 24 hours;

c = constant = 7 for machines of 10 tons and below, and 10 for machines above 10 tons per day.

Condensers.—The square feet of surface required does not seem to depend on the ordinary laws for the transmission of heat through metal walls, but 300 to 550 B.Th.U. may be transmitted per square foot of mean surface per hour.

Assuming condensing water inlet not to exceed 65° F. :

Then for each ton of ice-melting capacity (ton of refrigeration),

For submerged.

Ample : 60 to 70 running feet of $1\frac{1}{4}$ -inch internal diameter pipe.

Small : 50 running feet of $1\frac{1}{4}$ -inch internal diameter pipe.

Atmospheric.

Ample : 90 running feet of $1\frac{1}{4}$ -inch pipe.

Small : 60 running feet of $1\frac{1}{4}$ -inch pipe.

Pipes $\frac{5}{32}$ inch to $\frac{3}{16}$ inch thick.

It should be mentioned that the reason why the difference between the ample and small allowances are closer in the case of the submerged than in the atmospheric, is, that the conditions, under which the submerged work, are much more stable than those of the atmospheric condenser. Again, one of the reasons why such variations in practice exist is due to the fact that, in many installations, allowances are made for sections to be cut out of circuit for inspection, renewals or repairs, without seriously affecting the working of the machine.

The grids of coils for atmospheric condensers may be spaced 12 inches to 20 inches apart, 12 feet to 20 feet long and 8 feet to 12 feet high.

NOTE.—Taking the mean diameter of the pipes one running foot of 2-inch pipe = 1.33 foot of $1\frac{1}{2}$ -inch pipe = 1.6 foot of $1\frac{1}{4}$ -inch pipe = 2 feet of 1-inch pipe.

Evaporators.—The many and very varied forms and uses of evaporators, the variations in practice with regard to the difference in temperature between the ammonia and the surrounding medium, make it quite impossible in the scope of this Appendix to give even the barest outline for the design of an evaporator. Suffice it to say

TABLE 4.—*Compressors.*

Number of Machine.	Per day of 24 hours.			I.H.P. of Engine.	Compressor (Dimensions in Inches).					Rod.
	Ice Making.	Ice Melting.	Effective B.Th.U. removed.		R.P.M.	Diameter and Stroke.	Suction Valves. Diameter.	Delivery Valves. Diameter.	Diameter of Pipes.	
	Tons.	Tons.								
1	5	11·7	$3·5 \times 10^6$	17	70 to 80	$7\frac{1}{4} \times 15$	$2\frac{1}{2}$	$2\frac{1}{4}$	2	$1\frac{7}{8}$
2	10	20·0	6×10^6	27	65 to 75	9×18	$3\frac{3}{8}$	$2\frac{7}{8}$	$2\frac{1}{2}$	$2\frac{1}{4}$
3	25	45·0	$13·5 \times 10^6$	60	60 to 70	12×24	$4\frac{1}{4}$	$3\frac{1}{8}$	3	3
4	50	87·0	26×10^6	110	55 to 65	$15·5 \times 30$	$5\frac{1}{2}$	5	4	$3\frac{7}{8}$
5	100	170·0	51×10^6	210	50 to 60	21×36	$7\frac{1}{2}$	7	5 or 6	$5\frac{1}{4}$

NOTE.—(a) Number of machine is for reference in this Appendix only.

(b) The higher figures in revolutions to meet overloads.

TABLE 5.

Heat Units and Condensing Water. Condensing Water on at 55° F. and off 80° F.

Number of Machine.	B.Th.U. per 24 hours.				Condensing Water. Gallons per hour. Submerged.	
	Removed from cold body or effective.	Allowance for leakage into Machine and Connections, etc.	Heat Equivalent of Work Expended.	Total Removed.	Total.	Per ton Ice Making.
1	3,500,000	265,000	1,041,000	4,806,000	800	160
2	6,000,000	440,000	1,660,000	8,100,000	1,350	135
3	13,500,000	990,000	3,700,000	18,190,000	3,030	121
4	26,000,000	2,000,000	6,800,000	34,800,000	5,800	116
5	51,000,000	3,740,000	12,890,000	67,630,000	11,800	113

that if brine is being cooled under conditions similar to those which exist when ice is being made by the brine, then :—

For each ton of ice-melting capacity allow 125 to 150 running feet of $1\frac{1}{4}$ -inch internal diameter pipe, it being understood that the greatest possible care is taken to ensure an efficient circulation of the brine.

Tables 4 and 5 summarize the preceding rules.

APPENDIX VI.

TESTING VAPOUR-COMPRESSION REFRIGERATING MACHINES.

The bewildering conflict of results of apparently reliable tests of vapour-compression machines, carried out, in many cases, by eminent authorities, more particularly in comparative and competitive tests between the three principal refrigerants in general use— NH_3 , CO_2 , and SO_2 —shows conclusively that much study and careful research is required to put the knowledge of this class of machine on a satisfactory basis.

It must, however, be pointed out that published results of comparative tests nearly always lack essential figures. It is quite easy to make any one of the three types mentioned to come out best in a series of trials, and yet the records—published exactly as taken—to indicate that the machines were (apparently) working under exactly similar conditions.

Theoretically, it is best to keep the refrigerant saturated during its cycle (wet compression—*see* Appendix III, page 1010). The clearance in the compressor (that is, a practical consideration) is apt, however, to tell heavily against this system as compared with dry compression. The clearances in the compressor or compressors should always be included in the records.

Further, practice seems to indicate that the evaporator is more efficient under wet and the condenser under dry compression.

It follows, therefore, that possibly good results would be obtained by working the evaporator under wet conditions, and then thoroughly drying the vapour by some mechanical means (say, in a spiral chamber) in its passage from the evaporator to the compressor, thus superheating the vapour in the compressor to a limited extent, this in turn allowing the condenser to do its best duty. Records should therefore be given of the temperatures of the refrigerant at all important points in the circuit.

Both theoretically and practically the coefficient of performance of a machine is improved by cooling the liquid refrigerant, after it leaves the condenser and before it passes the regulating valve, to the lowest available temperature in the machine itself. If this is done directly or indirectly the records should show it.

In the following suggested methods for recording tests, it is assumed that the actual amount of energy supplied for driving the machine can be determined. Electric driving by a motor of known efficiency has much to recommend it.

Brine circulation in the evaporator is also assumed and for laboratory tests the brine can be heated electrically or by steam, a check being thus afforded on the heat eliminated by the machine. Endless possibilities (for laboratory work) are offered in the way of heat exchangers for utilizing the heat wasted in the condensing water, etc.

DATA REQUIRED.

Compressor.

Type	{ Double or single acting, horizontal or vertical. If fitted with water-jacket, extra records to be inserted accordingly. }	
Diameter		
Stroke		
Clearance.—Back	Volume	
Front	Volume	
Diameter of compressor-rod		
Volume swept through by compressor-piston per revolution		

Condenser.

Type.....

Diameter of pipe { Internal.....
External.....

No. of sections.....

Length of pipe in each section ft.

Total length of pipe ft.

Estimated heating surface { Internalsq. ft.
Externalsq. ft.

Material of pipe.....

Remarks *re* circulation of water.....

Evaporator.

Same as for condenser, and in addition—

Method (if any) for agitating the brine.....

Insulated or in insulated space.....

Brine.

Salt employed.....

Tables of specific heats and specific gravities of the brine for ranges of temperature used in the test.....

Note.—It is better to run the machine for some hours before observations are taken, in order to avoid allowances, which otherwise must be made due to varying temperatures and consequently varying specific heats of the brine.

Methods of measuring and checking the quantities of brine circulated

Refrigerant.

Outline description of method employed for measuring the quantity (weight) of refrigerant circulated.

Water.

Methods of measuring and checking the quantities of water circulated.

Remarks.—Quality of water (town supply, well, canal, sea-water, etc. Hard or soft). Note if the water-supply be heated to a stated temperature before use, etc.

Regulating Valve.

Outline description. Record of movement (if any) during the test.

Temperatures.

Detailed account of method or methods adopted for reading temperatures.

General Remarks.

Outline description of any particular fitting likely to affect the test, such as a drier between the evaporator and compressor.

Amount of refrigerant in the machine. Interval of time between charging and testing the machine. Precautions taken to eliminate air or other foreign gases from both the refrigerant and brine circuits.

OBSERVATIONS REQUIRED.

Date Atmospheric conditions
 Number of trial
 Duration

Compressor.

- (1) Indicated horse-power
- (2) Heat equivalent of (1) B.Th.U.
- (3) Vapour entering compressor. Temp. Pressure.
- (4) „ leaving „ Temp. Pressure

Condenser.

- (5) Temperature of water, inlet
- (6) „ „ outlet
- (7) „ „ difference (5) and (6)
- (8) Quantity of water circulated
- (9) Heat rejected by condenser B.Th.U.
- (10) Vapour entering condenser. Temp. Pressure
- (11) Liquid leaving „ Temp. Pressure

Evaporator.

- (12) Temperature of brine, inlet
- (13) „ „ outlet
- (14) „ „ difference (12) and (13)
- (15) Quantity of brine circulated
- (16) Allowance $\pm$ B.Th.U. for variations in (12) and (13) during trial
- (17) Net refrigerating effect B.Th.U.
- (18) Refrigerant entering evaporator. Temp. Pressure
- (19) „ leaving „ Temp. Pressure

Heat Balance.

- (20) Net refrigerating effect (17)B.Th.U.
 (21) Heat equivalent of work expended in }B.Th.U.
 compressor (2) }
 (22) Total heat imparted to refrigerant }B.Th.U.
 (20) and (21) }
 (23) Total heat rejected at condenser (9)B.Th.U.
 *(24) Difference between (22) and (23)B.Th.U.

Efficiency.

- (25) Heat equivalent of energy supplied to machineB.Th.U.
 (26) Coefficient of performance (a). Ratio of (2) to (17)
 (27) Coefficient of performance (b). Ratio of (25) to (17)
 (28) Efficiency of driving. Ratio of (25) to (2).....
 (29) Capacity of machine. Ice melting per day of 24 hrs.....

General.

- (30) Weight of refrigerant circulatedlb.
 (31) Estimated refrigerating effect from (30).....
 (32) Difference between (17) and (31)B.Th.U.
 (33) Temperature of liquid refrigerant before passing the regulating
 valve.....

* This difference is generally fairly large. In commercial machines due allowances are made for "heat leakage" into the pipes and connections. See Table 5 (page 1027).

VAPOUR-COMPRESSION REFRIGERATING MACHINES. Pl. 42.

*Horizontal and Vertical Ammonia Compressors, Belt or Steam Driven.
(The L. R. Co., Sterne and Haslam.)*

Mechanical Engineers 1912.

VAPOUR-COMPRESSION REFRIGERATING MACHINES. Pl. 44.
Carbonic Anhydride Marine Compressors. (Haslam.)

Sulphur or Anhydride Compressors. (Douglas.)

Dairy, showing Milk cleaner, Pasteurizer, Separator, etc., and Compound Milk-Cooler (See Fig. 19.) working in conjunction with above plant.

Complete SO₂ Plant for general land work.