Elements of Refrigeration
FOR
Small Commercial Plants

By
WALLACE H. MARTIN

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March 1935

Engineering Experiment Station
Oregon State Agricultural College
CORVALLIS
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Elements of Refrigeration for Small Commercial Plants

By

WALLACE H. MARTIN
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I. INTRODUCTION

1. Purpose and scope. There are many small commercial plants in Oregon installed in connection with creameries, ice-cream plants, and markets that are too small to warrant the employment of a specialist on refrigeration. Many of these plants are being operated uneconomically because of lack of a clear understanding, on the part of their operators and owners, of the fundamentals of mechanical refrigeration. It was a realization of this need expressed by some of these men that led to the publication of this circular.

II. THE REFRIGERATION CYCLE

2. Refrigeration process. The mechanical system as most commonly used depends for its operation on the fact that when a liquid boils or evaporates, it absorbs heat from its surroundings. When water boils at atmospheric pressure, it absorbs about 970 B.t.u. per pound. Ammonia boiling at atmospheric pressure absorbs about 590 B.t.u. per pound. Other refrigerants such as sulfur dioxide, methyl chloride, and F-12 or freon, absorb smaller amounts of heat per pound when they evaporate.

A refrigerator could consist of only a supply of refrigerant, a regulating valve and a coil of pipe in which the refrigerant would evaporate. The evaporation of the refrigerant in the coil would absorb heat from the substance in which the coil is immersed and refrigeration would be produced. Such a simple system would, however, allow of using the refrigerant only once. The cost of most refrigerants is too great to permit of such extravagance, and the alternative is to recover the vapor and use it over repeatedly.

For this purpose some form of compressor is usually employed. The vapor coming from the refrigerator is taken into the compressor and its pressure increased. It is then discharged into a piece of equipment commonly called a condenser. Cooling water or air is caused to flow over the surface of the condenser. This water or air absorbs heat from the gas contained in it and causes the gas to condense to a liquid. This liquid is then drained by gravity into a container called a receiver, from which it flows to the regulating or expansion valve, and the cycle is repeated.

Figure 1 indicates diagrammatically the arrangement of the essential parts of a refrigerating system. In the evaporator the refrigerant is changed from liquid to vapor and during this change heat is absorbed. The vapor is compressed in the machine to a higher pressure and leaves the machine as a warm gas. It then passes to the condenser where it is first cooled and then condensed to a liquid. From the bottom of the condenser
it drains by gravity to the receiver. The receiver is merely a storage tank and serves as a receptacle for the excess refrigerant. From the receiver the refrigerant passes to the regulating valve, commonly called the expansion valve, and then to the evaporator, thus completing the cycle. From the expansion valve through the evaporator to the compressor the pressure to which the refrigerant is subjected is a comparatively low pressure and is called the evaporating pressure, back pressure, or suction pressure. From the compressor through the condenser and the receiver to the expansion valve the pressure is comparatively high and is called the discharge pressure, condensing pressure, or head pressure.

![Diagram of Refrigeration System](image)

**Figure 1. Diagram of Refrigeration System.**

This illustrates the process in use in the ordinary mechanically operated unit. The compressor is usually driven by an electric motor, but may of course be driven by any power unit available.

3. The evaporator. The evaporator is the most essential part of the whole system. The amount of heat removed per hour is dependent on three factors. These are: (a) the temperature difference, (b) the square feet of evaporator surface, and (c) a factor called the coefficient of heat transfer which we shall discuss more at length under the general topic of heat transfer. Written as a formula:

$$H = KS(t_b - t_r)$$

Where $H =$ heat transmitted per hour B.t.u.
$K =$ coefficient of heat transfer
$S =$ evaporator surface in square feet
$t_b =$ temperature of brine or other substance being cooled
$t_r =$ temperature of evaporating refrigerant

By changing any one of these factors the total heat transfer can be changed or the total heat transfer can be kept constant by increasing one
and decreasing the other in the same proportion. If the surface is doubled the temperature difference can be halved.

Ample surface in the evaporator permits a smaller difference between evaporating temperature and temperature of substance to be cooled with a consequent saving in compressor displacement.

Heat, like water, flows freely only in a down-hill direction. If a room or brine tank is to be held at a fixed temperature the substance which removes the heat must be at a lower temperature in order that heat may flow to it from the air in the room or from the brine. Ordinary ice will not cool a room below 32° F. In fact, temperatures much below 40° F. cannot be obtained with ice unless salt is used with it. If coils are immersed in brine and refrigerant is evaporated in the coils the evaporating temperature must be several degrees below the temperature at which the brine is to be held. For instance, suppose brine is to be cooled to -10° F. The refrigerant should probably be evaporated at approximately -20° F.

4. Suction pressure. The temperature at which the refrigerant is evaporated determines its pressure. By referring to any table of refrigerants, it will be noted that for every temperature there is a corresponding boiling pressure. At -20° F. the evaporating pressure of ammonia is 18.3 pounds per square inch absolute or 3.6 pounds per square inch gage. At normal atmospheric pressure the boiling temperature is 28 degrees below Fahrenheit zero or -28° F.

As heat flows from the substance being refrigerated to the ammonia, the liquid changes to vapor. As fast as this vapor forms it must be removed in order to make room for new vapor which is constantly forming. If the vapor is removed at the same rate as it forms, the pressure will remain constant or, if it is removed more rapidly than it forms, the pressure will decrease. The removal of this vapor is the function of the compressor.

5. Compressor. There is a relation between suction pressure and capacity of compressor that is not always appreciated. The amount of heat absorbed in the evaporator is dependent primarily on the number of pounds of refrigerant vaporized. A compressor operating at a constant speed will remove from the evaporator a certain number of cubic feet of vapor per minute. At low pressures a pound of refrigerant vapor occupies more space than at high pressure. If the suction pressure is lower than necessary the capacity is unnecessarily low because the number of pounds of vapor which can enter the machine is smaller.

Referring once more to the case where we desire to cool brine to -10° F., if the ammonia is evaporated at -20° F. the volume the machine must remove from the evaporator for each pound of ammonia evaporated is 14.67 cubic feet. (See Appendix A.) If the evaporating temperature can be increased to -15° F. the volume to be drawn in by the machine per pound of ammonia is 12.97 cubic feet. The capacity of the machine in the second case will be 1.13 times as great as in the first case on the basis of pounds of vapor pumped out of the evaporator.

The thoughtful operator will always operate his machine at as high a suction pressure as possible, consistent with the temperatures that must be carried, because he knows that in that way he can get more capacity and also use less power per unit of capacity.

6. Condenser. The vapor is pumped by the compressor into the condenser. The condensing pressure, also called the discharge or head
pressure, is determined by the temperature of the cooling medium. This medium is nearly always water except in the small household machine where air is used. With a good condenser properly operated, the condensing temperature of the refrigerant should not be more than two or three degrees warmer than the water leaving. For instance, if the water comes to the condenser at 65° F. and leaves at 75° F., the condensing temperature should not be greater than about 77° F. The ammonia pressure corresponding to this is 145.5 pounds per square inch absolute or 130.7 pounds per square inch gage. The head pressure cannot possibly be less than this with a 77° F. condensing temperature, and if noncondensable gases are present in the system it will be higher. In a well operated and maintained plant, however, it should not be more than two or three pounds per square inch greater. A great difference between suction pressure and head pressure requires expenditure of more power for a given compressor capacity than a small difference. Cold condensing water results in a lower condensing temperature and consequently a lower head pressure, hence an effort should always be made to get as cold condensing water as possible consistent with cost.

7. Illustrative calculations. Let us follow through the calculations for a given size of machine.

A certain machine is 6 inches in diameter by 6 inches stroke and operates at 200 revolutions per minute. There are two cylinders, each single acting. The evaporator temperature is -10° F. and cooling water comes to the condenser at 60° and leaves at 70° F. The condensing temperature should not be more than 72° F. The suction pressure will be 23.74 pounds per square inch absolute and discharge pressure 133.4 pounds per square inch absolute.

The volume of a cylinder 6 inches in diameter by 6 inches stroke is 169.6 cubic inches. The cylinder will not be completely filled with fresh vapor each suction stroke for the following reasons. At the completion of the discharge stroke the clearance space will be full of high pressure vapor. As the piston starts down this high pressure vapor expands and partly fills the cylinder. In order that the vapor from the evaporator may flow into the cylinder, the pressure in the cylinder must be slightly less than the evaporator pressure. This causes the incoming vapor to occupy more space in the cylinder than it would require if it were at the same pressure as the vapor in the evaporator. Furthermore, the cylinder is warmer than the incoming vapor and gives up its heat to the vapor causing it to expand. All these factors together reduce the effective displacement to something less than the actual volume swept through by the piston. A reasonable value for the case we are considering is 83 per cent of the actual displacement. This factor is called the volumetric efficiency and is estimated as follows: Divide the discharge pressure in pounds per square inch absolute by the suction pressure in pounds per square inch absolute. The result is the "compression ratio," and for the case under consideration is 133.4/23.74 or 5.62. Refer to the chart, Appendix B, and locate the horizontal distance 5.62. The intersection of a perpendicular at this point and the curve labeled "Vertical S. A." indicates the value of the volumetric efficiency which can be read on the vertical scale at the left and is about 83 per cent.

The effective displacement is then 169.6 cubic inches \times 0.83 or 141.4 cubic inches. In one minute there are 200 suction strokes per cylinder and since there are two cylinders the total effective displacement is
A Mollier Chart for Ammonia, Modified from the one in Circular 142, U. S. Bureau of Standards.
The volume of one pound of ammonia at \(-10^\circ\) F. is 11.5 cubic feet per pound. The number of pounds that the machine will remove from the evaporator per minute is 32.6/11.5 or 2.84.

The ammonia just before it enters the evaporator is a liquid at 72° F. and its heat content is 122.8 B.t.u. per pound. In the evaporator it absorbs heat and is changed to a vapor at \(-10^\circ\) F. At this condition its heat content is 608.5 B.t.u. per pound. The heat absorbed or refrigerating effect is the difference between these two values or 485.7 B.t.u. per pound. The total heat absorbed per minute is 2.84 \times 485.7 or 1,380 B.t.u.

The unit of capacity in refrigeration is the ton. This unit came into use because of the fact that ice was formerly used for refrigeration. It was purchased by the ton of 2,000 pounds and was delivered daily. It was easy to make the transformation from a ton of ice per 24 hours to a ton of ice melting capacity per 24 hours. The heat absorbed when one pound of ice melts is 144 B.t.u. The heat absorbed by the melting of 2,000 pounds is 2000 \times 144 B.t.u. or 288,000 B.t.u.

\[\frac{288,000}{24} = 12,000\text{ B.t.u. per hour} \text{ or } \frac{288,000}{24 \times 60} = 200\text{ B.t.u. per minute}.\]

The machine we are considering is then operating with a capacity of 1380/200 or 6.9 tons.

We next want to know what horsepower will be required to drive the machine. While the cylinders of compressors are water jacketed and some heat is removed by the water in the jackets, for the time being we will not introduce any serious error if we neglect this heat transfer and assume that there is no heat flow to or from the vapor while it is in the cylinder. This kind of a process is known in thermodynamics as a constant entropy adiabatic. We shall not attempt to explain the meaning of this term in all its aspects but only its use. In the ammonia tables, and tables for other refrigerants, there is a property designated “entropy” and its value for the case we are considering is 1.3558 for dry vapor. If the vapor is compressed and this property is maintained constant until a pressure of 133.4 pounds per square inch absolute is reached the heat content will be increased to 717 B.t.u. per pound. This value can readily be obtained from a chart such as accompanies nearly all refrigerant tables. Such a chart for ammonia follows this page. The chart is constructed with heat values plotted horizontally and pressures plotted vertically. Constant entropy lines and constant volume lines are indicated. By following the constant entropy line on the ammonia chart from a point representing dry vapor to a point representing a pressure of 133.4 pounds per square inch absolute, the value 717 can be read at the top or bottom of the chart. The heat content at entrance to the compressor is 608.5 B.t.u. per pound and since we have assumed that there is no heat transfer to or from the vapor in the cylinder the gain must be due to the work done on the vapor. This is 717 \(-\) 608.5 or 108.5 B.t.u. per pound. The work done per minute is 2.84 \times 108.5 B.t.u. or 308 B.t.u. A horsepower is 33,000 foot pounds per minute and since there are 778 foot pounds in one B.t.u. a horsepower expressed in B.t.u. is 42.4 B.t.u. per minute. The horsepower required for this case is 308/42.4 or 7.27.

The effect of the jacket cooling is to reduce the discharge temperature and the heat content at discharge somewhat, which results in a slight reduc-
tion in work done on the vapor. This is more than counterbalanced by the friction of the machine and other energy losses, such as the compression of non-condensible gas, a little of which is nearly always present, so that the actual horsepower that must be furnished by the motor is about 20 per cent greater than that represented by the work done on the vapor. The horsepower for this case is then \( 1.2 \times 7.27 \) or 8.72. A ten horsepower motor would probably be used.

Each pound of cooling water, since it warms 10 degrees, will remove 10 B.t.u. The heat content of the vapor leaving the compressor is 717 B.t.u. per pound and the heat content of the liquid leaving the condenser is 122.8 B.t.u. per pound. In condensing the vapor to liquid there must be removed from each pound 717—122.8 B.t.u. or 594.2 B.t.u. This will require 59.42 pounds of water per pound of ammonia. The water required per minute will be \( 2.84 \times 59.42 \) pounds or 169 pounds. There are 8.34 pounds in a gallon so there will be needed \( 169/8.34 = 20.25 \) gallons per minute.

Summarizing:

- Evaporator temperature: -10°F.
- Suction pressure: 23.74 pounds per square inch absolute
- Condensing temperature: 72°F.
- Discharge pressure: 133.4 pounds per square inch absolute
- Actual displacement: 169.6 cubic inches per stroke
- Volumetric efficiency: 83 per cent
- Effective displacement: \( 0.83 \times 169.6 = \frac{141 \times (2 \times 200)}{1728} = 141 \) cubic inches per stroke
- Horsepower: \( \frac{32.6 \times 11.5}{608.5} = 2.84 \times 108.5 = 308 \)
- Horsepower per ton: \( \frac{308}{42.4} = 7.27 \)
- Horsepower of motor to drive: \( 1.2 \times 7.27 = 8.72 \)

Heat removed in condenser, B.t.u. per pound

- Heat content at discharge from compressor, B.t.u. per pound: 717
- Work done on vapor in compressor, B.t.u. per pound
- Work done per minute, B.t.u
- Heat removed in condenser, B.t.u. per pound
- Pounds of water required per pound of ammonia
- Pounds of water per minute
- Gallons of water per minute
- Pounds of ammonia per ton
- Horsepower per ton
- Gallons per minute per ton
By following the same method of procedure results can be calculated for any other set of conditions or for any other refrigerant provided tables of properties of the other refrigerant are available.

8. Interrelation of suction pressure, size of compressor, capacity, and power. The curves in Appendix C show how, for a machine with a fixed effective displacement and a fixed discharge pressure, the capacity, horsepower, and horsepower per ton, vary when the suction pressure is changed. The capacity in tons increases in almost direct proportion to the increase in suction pressure. The actual power required to drive the machine increases rapidly at first but for the higher suction pressures it is practically constant. The horsepower per ton is very high for low suction pressures, but drops rapidly as the suction pressure is raised. This emphasizes the desirability of operating with as high a suction pressure as possible consistent with other requirements in the plant.

There are many installations where several different temperatures must be carried. For instance, a plant in which milk and cream are cooled and ice cream is made, requires a very low temperature for hardening ice cream and moderate temperatures for storage and cooling of milk and cream.

When only one compressor is used the suction pressure must be as low as that corresponding to the lowest temperature. For instance, a hardening room may be held at -5° F. Refrigerant in the coils will have to be at a temperature lower than this, perhaps at -15° F. This will necessitate a pressure, if ammonia is the refrigerant, of 20.88 pounds per square inch absolute. If ice is made in the same plant, the ammonia temperature in the evaporating coils need not be lower than 8° F. which corresponds to a suction pressure of 36.77 pounds per square inch absolute. If the same machine is compressing the vapor for both, however, the suction pressure throughout has to be 20.88 pounds per square inch absolute.

Referring now to Appendix C, the horsepower per ton for a suction pressure of 20.88 pounds per square inch absolute is about 0.86 while for a suction pressure of 36.77 pounds per square inch absolute, the horsepower per ton is only 0.62, or a reduction of about 28 per cent. Looking at the situation from the standpoint of capacity the higher suction pressure will make it possible for a machine of 100 cubic feet per minute effective displacement to produce about 44 tons of refrigeration as against about 30.5 at the lower suction pressure. In other words, the capacity at the higher suction pressure is about 44 per cent greater than at the low suction pressure.

If a plant has only one machine, there is no remedy for this situation but in laying out new plants or remodeling old ones, it is often possible to install two compressors. When this can be done, they should be so connected that one can be used for low-temperature work and the other for high-temperature work. In one remodeled installation where this was done, two machines had formerly been operated in parallel, that is both at the same suction pressure. During warm weather in summer it was often necessary to stop making ice in order to carry the ice cream load. Ice had to be purchased during this period. After making the change the low temperature load was carried and ice was made every day throughout the summer.
III. HEAT TRANSFER

9. Film concept. The generally accepted theory of heat transfer is known as the "film theory." Let us suppose that brine is to be cooled by evaporating refrigerant in a coil immersed in the brine. In contact with the surface of the pipe coil, there is a film of brine that may be thought of as being stationary. This film offers resistance to heat flow through it from the brine to be cooled to the pipe surface. The pipe wall offers additional resistance. Inside of the pipe there is a film of refrigerant vapor which may also be thought of as being stationary. It offers additional resistance. The total resistance to heat flow from the brine to the refrigerant is then the brine film, the pipe wall, and the vapor film. If either surface of the pipe has any scale, dirt, oil, or other coating on it, the effect will be to offer additional resistance.

Obviously the resistance of any film will depend on the substance of which the film consists and the thickness of the film. A film of liquid will offer less resistance than a film of gas. All gases do not offer the same resistance. Rapid movement of the substance over the surface has a tendency to scrub the film off, thus reducing its thickness and reducing the resistance. Investigation of heat transfer through pipe walls has shown that the film resistance has a far greater effect than the resistance of the pipe wall itself. Since so many factors have an influence on the resistance to heat flow, it is evident that great care must be used in estimating the heat flow in any given case. Whole books have been written on the subject of heat transfer and although much progress has been made, information is still far from exact.

10. Coefficient of heat transfer. We have already referred to "Coefficient of Heat Transfer." By this we mean the B.t.u. that are transmitted per hour from the substance from which heat is flowing, to the substance to which heat is flowing, through one square foot of separating wall, when the difference between the temperatures is 1°F. Using the brine cooler example again, if the temperature of the brine is 1°F higher than the temperature of the evaporating refrigerant in the coil, the coefficient of heat transfer is numerically equal to the B.t.u. transmitted from the brine to the ammonia per hour through each square foot of pipe surface.

Coefficients for various kinds of conditions have been determined from experience and when used with care and judgment make possible a proper proportioning of surface for a given requirement.

In refrigeration heat transfer must be regarded in two lights. In the evaporator, condenser, liquid coolers, and similar apparatus, rapid heat transfer is desired and all improvements in design and construction have for the ultimate object more rapid heat flow.

On the other hand, walls of rooms are designed and built to retard as much as possible, consistent with cost, the heat flow through them. Materials used for cold storage walls or for covering cold pipes or ducts are usually called insulators. The same film theory, however, applies and the same methods are used to obtain the coefficient of heat transfer (K).

The coefficient for building walls is more often calculated, however, by taking into consideration the various film resistances and materials of the wall, than in the case of pipe coils.
11. **Method of calculation.** To illustrate, let us suppose that we have a wall made up of 6 inches of concrete and 4 inches of cork. What is the coefficient? The outside of the concrete wall is presumably an outside wall exposed to outside air circulation. Experience has shown that for average conditions the heat transfer through the air film next to the wall is about 4.2 B.t.u. per square foot per hour for each degree difference. The resistance is the reciprocal of this or \( \frac{1}{4.2} = 0.24 \). The coefficient for concrete is about 8 B.t.u. per square foot, per hour, per degree difference, for each inch thickness or the resistance is \( \frac{1}{8} \). The resistance for a 6-inch wall is 6 times as much or \( 6 \times \frac{1}{8} = 0.75 \). The coefficient for cork is about 0.31 B.t.u. per square foot, per hour, per degree difference, per inch thickness, or the resistance is \( \frac{1}{0.31} \). The resistance for a 4-inch wall is 4 \( \times \frac{1}{0.31} \) or \( \frac{4}{0.31} = 12.9 \). The inside is in contact with still air and experience has shown that an average value of the coefficient for the film is about 1.4. The resistance then is \( \frac{1}{1.4} \) or 0.77. The total resistance for this wall is then \( 0.24 + 0.75 + 12.9 + 0.77 = 14.66 \) and the over-all coefficient is \( \frac{1}{14.66} = 0.0682 \). It is apparent that the principal resistance is offered by the cork. By following the same procedure over-all coefficients for any type of wall can be obtained with a fair degree of accuracy.

12. **Heat leakage.** Let us now suppose that this room is 20 \( \times \) 30 feet by 10 feet high. The distance around the room is 100 feet. The total wall surface is 10 \( \times \) 100 = 1,000 square feet. With an outside temperature of 70° F. and an inside temperature of 30° F. the heat leakage through the wall is \( (70 - 30) \times 1,000 \times 0.0682 \) or 2730 B.t.u. per hour. The leakage through the floor and ceiling can be computed in the same way and thus the heat leakage for the room obtained. Any additional refrigeration due to cooling of goods in the room will have to be added to the heat leakage to get the total load.

13. **Water-proofing.** In installing insulating material of any kind it is very necessary that special precautions be taken to keep out moisture. The insulating value of practically all materials in common use is due to the minute air spaces contained. If these air spaces become filled with water the insulating value of the wall is destroyed. All air contains some water vapor. If air passes through a cold-storage wall, it carries water vapor with it. The average temperature of the wall is lower than that of the outside air. When the air cools, some of the water vapor condenses and collects in the wall. If the temperature in any part of the wall is low enough to cause the moisture to freeze, the insulating value will not only be destroyed but the wall will disintegrate. Evidently then, a cold storage wall should be made as nearly airtight as possible. This is accomplished in the case of many insulators by coating the surfaces with a water-proof paint. In the case of walls built up of wood and such materials as kiln dried planer shavings, felt paper is used in the wall to retard air leakage. A good construction consists of two thicknesses of tongued and grooved lumber on each side of the wall. The space between is filled with planer shavings. Between the planer shavings and the lumber there should be a layer of a good grade of felt paper with the edges well lapped over each other.

**IV. DAIRY REFRIGERATION**

14. **Requirements.** The dairy manufacturing industry requires refrigeration in connection with all of its processes and the varied nature of the application introduces many problems.
Milk and milk products must be cooled, butter must be stored, ice cream is frozen and hardened, and usually ice is made.

15. **Milk cooling.** The refrigerating load depends on the weight of product being cooled, on its specific heat, and on the temperature change. As an example, let us suppose that it is desired to cool 1,000 gallons of milk from pasteurizing temperature, 140° F., to 40° F. Specific gravity of milk is about 1.032. The weight per gallon is about 8.6 pounds. The specific heat is about 0.95. The amount of heat to be removed can be calculated by use of the formula:

\[ H = WC (t_1 - t_2) \]

Where:
- \( H \) = heat to be removed
- \( W \) = weight of substance being cooled
- \( C \) = specific heat, B.t.u. per pound degree
- \( t_1 \) = high temperature
- \( t_2 \) = low temperature

Applying this to the case we have assumed for illustration:

\[ H = 1000 \times 8.6 \times 0.95 (140 - 40) \]
\[ = 816,000 \text{ B.t.u.} \]

As much of this cooling should be done with water as possible. In most plants, it is entirely feasible to cool to 80° F. with water. The heat to be removed by water is:

\[ H = 1000 \times 8.6 \times 0.95 (140 - 80) \]
\[ = 490,000 \text{ B.t.u.} \]

If the water used for cooling is allowed to rise 10 degrees, the amount of water necessary is

\[ \frac{490,000}{10} \text{ or } 49,000 \text{ pounds} \]

Since water weighs 8.34 pounds per gallon, the number of gallons required is 49,000/8.34 = 5,880. If a four-hour period is allowed for the cooling, the amount of water per hour is 5,880/4 = 1,470. About 20 per cent extra should be added for losses. The requirement then is 1764 gallons per hour or 29.4 gallons per minute.

To cool from 80° F. to 40° F. the amount of heat to be removed, calculated in the same manner is 326,000 B.t.u. The cooling substance is usually brine. The brine for this purpose should enter the cooling coil at a temperature of about 20° F. and leave at about 30° F. Its specific heat is dependent on the amount of salt in it, but a reasonable value is 0.86. The amount of heat which the brine will remove can be calculated by use of the same expression used before:

\[ W = \frac{H}{C (t_1 - t_2)} \]
\[ = \frac{326,000}{0.86 (30 - 20)} \]
\[ = 37,900 \text{ pounds} \]

The weight per gallon varies considerably, depending on the salt content, but nine pounds per gallon is a reasonable value for brine suited to the temperature range we are considering.

The number of gallons required is 37,900/9 = 4,211
If a four-hour period is allowed for the cooling, the number of gallons per hour is \( 4,211/4 \) or 1,053, which is \( 1,053/60 = 17.55 \) gallons per minute. There will be some heat losses, so a pump capacity of about 20 gallons per minute should be provided.

While we are considering the milk cooler let us use it as an illustration to calculate the amount of surface required. For the part of the cooler in which water is the cooling medium the amount of heat to be removed per hour is \( 490,000/4 = 122,500 \) B.t.u. About 20 per cent should be added to provide for losses.

\[
1.2 \times 122,500 = 147,000 \text{ B.t.u.}
\]

The milk comes to the cooler at 140° F. and leaves the water-cooled section at 80 ° F. The water temperature of course depends on local conditions, but we shall assume 60° F. entering and 70° F. leaving. The milk enters the top of the cooler and leaves at the bottom, while the water enters at the bottom of the section and leaves at the top. This is characterized as counter flow and whenever it is obtainable in heat transfer apparatus it is usually used. It may be noted that the coolest milk is being cooled by the coldest water and the warmest milk by the warmest water. The temperature difference at the warm end is 140 — 70 = 70° F. and at the cold end 80 — 60 = 20. The arithmetic mean difference is \( (70 + 20)/2 = 45° \) F.

\[
H = K S t_d
\]

\[
H = \text{heat transfer B.t.u. per hour}
K = \text{coefficient of heat transfer}
S = \text{surface in square feet}
t_d = \text{mean temperature difference}
S = H/K t_d
\]

A reasonable value of \( K \) is 60 (See Table in Appendix D).

\[
147,000
60 \times 45
= 54.4 \text{ square feet.}
\]

The surface required for the brine cooled section can be calculated in the same manner. The amount of heat to be removed by the brine is 326,000 B.t.u. Allowing 20 per cent additional for losses the total is \( 326,000 \times 1.2 \) or 391,200. Let us assume that the compressor is operating eight hours per day. During the cooling period the evaporating ammonia will remove half of the 391,200 B.t.u. and the other half will have to be absorbed by the brine to be removed by evaporating ammonia later.

Using equation (2) again.

\[
H = W C (t_1 - t_2)
\]

\[
H = \text{heat to be absorbed by brine}
W = \text{weight of brine in system}
C = \text{specific heat of brine}
t_1 = \text{temperature of brine at end of milk cooling period}
t_2 = \text{temperature of brine at beginning of milk cooling period}
\]

Allow the brine to warm 5°F. during the cooling period, \( t_1 - t_2 = 5 \)

\[
W = \frac{H}{C (t_1 - t_2)} = \frac{391,200}{2 \times 0.86 \times 5}
= 45,450 \text{ pounds}
\]
At 9 pounds per gallon this is 5,050 gallons. On the basis of an operating time of 8 hours the refrigerating equipment must remove heat at the rate of 391,200/8 or 48,900 B.t.u. per hour. The tonnage capacity of the machine must be 48,900/12,000 or 4.075. The amount of evaporating surface in the brine tank is calculated by use of equation (3).

\[ H = K S t_d \]
\[ S = H/K t_d \]

A reasonable value for \( K \) for this case is about 15 since there is some circulation of the brine. The average brine temperature is 30° F. The evaporating ammonia can be taken at 5° F.

\[ S = \frac{48,900}{15 \times (30 - 5)} = 130.5 \text{ square feet} \]

If 1\( \frac{1}{4} \) inch pipe is used there is needed 130.5 \times 2.3 or 300 lineal feet.

16. **Milk storage.** After cooling, the milk is placed in a storage room where a temperature of about 35° F. is maintained. This room is usually also used for storage of other products such as cream and butter. The amount varies so much that it is not usually very satisfactory to proportion the piping on the basis of the goods stored, but experience has shown that if a certain amount of refrigeration is provided per cubic foot of room space, satisfactory conditions can be maintained. For a room having a volume of 2,200 cubic feet to be held at 35° F. about one ton of refrigeration should be provided. For a 1,000-cubic-foot room about 0.55 tons should be provided. For more detailed data on this point, see Macintire:° “Principles of Mechanical Refrigeration” 2nd edition, p. 292. The room is usually cooled by direct expansion and a reasonable evaporating temperature is 5° F. The heat transfer coefficient for direct expansion pipe is about 2. To remove 12,000 B.t.u. per hour, the surface is calculated as follows:

\[ H = K S (t - t_r) \]
\[ H = \text{heat to be removed per hour} = 12,000 \]
\[ K = \text{coefficient of heat transfer} = 2 \]
\[ S = \text{surface of coil in square feet} \]
\[ t = \text{temperature of room} = 35° F. \]
\[ t_r = \text{temperature of evaporating refrigerant} = 5° F. \]

\[ S = \frac{H}{K (t - t_r)} \]
\[ S = \frac{12,000}{2 (35 - 5)} = 200 \]

If one inch pipe is used the number of lineal feet required is 200 \times 2.9 = 580. For one and one-fourth inch pipe the lineal feet \( = 200 \times 2.3 = 460 \). The pipe should be located near the top of the room and a room with a ceiling height of the order of 10 feet should be used. A bunker as illustrated in Figure 2 facilitates better air circulation.

A brine tank is often located in the milk storage room. When so located, it should be placed near the ceiling. The refrigeration stored in it during the time the machine is in operation will help to hold the room at the proper temperature during the shutdown period and the fact that
it is in an insulated room eliminates the necessity of insulating the tank itself. If it cannot be placed near the ceiling, it should not be placed in

![Diagram showing bunker system for cooling rooms of six feet or less outside width. W=inside width of cooler. For wider rooms double bunker is used. (Frigidaire Sales Engineering Manual. Chap. VI., p 7.)](image)

the room at all because it occupies valuable space, does not help to hold the room temperature very well, and renders it much more difficult to keep the room neat and sanitary.

In a dairy manufacturing plant, two compressors, each operating at a different suction pressure, are desirable. The machine which cools the brine for freezing and cools the hardening room can be operated with an evaporating temperature of -20° F, which corresponds to a pressure of 18.3 pounds absolute or 3.6 pounds gage. The machine which serves the milk storage room, the brine tank for milk cooling, and the mix-holding vat should have an evaporating temperature of about 5° F, which requires a suction pressure of 34.3 pounds absolute or 19.6 pounds gage. If ice is made the ice tank should also be connected to this machine.

**V. ICE CREAM**

**17. Process.** In the manufacture of ice cream the mix must be cooled after pasteurization and then held at a temperature of 40° F, or less, in aging vats for two or three days. It is then frozen in a brine or direct expansion type of freezer after which it is placed in a hardening room for several hours for the purpose of removing more heat.
18. **Cooling of mix.** The refrigerating load for cooling the mix from pasteurizing temperature to the temperature at which the mix is put into the aging tank is calculated in exactly the same way as for milk cooling. The specific heat varies for different varieties of mix, but a reasonable value is 0.74. The same brine which is used for milk cooling can be used for mix cooling.

19. **Aging.** The vat in which the mix is stored for aging is built with a jacket surrounding it in which brine of the same strength as that used for mix cooling is placed. A comparatively small amount of evaporating coil is placed in this brine to maintain it at a constant temperature.

20. **Freezing.** According to Kennedy, the amount of heat to be removed in the freezer per pound of ice cream is 48 B.t.u. A gallon weighs about 5 pounds. A ten gallon freezer requires the removal of $5 \times 48 \times 10 = 2400$ B.t.u. during the freezing process. These values are all subject to some variation with different varieties of product. If we allow for the freezing of two batches an hour the heat that must be removed per hour is $2 \times 2400 = 4800$ B.t.u. During the peak season provision should be made for operating at this rate for eight hours per day. The capacity required is $4800/12000 = 0.4$ tons. If proper allowance is made for losses and for requirements for different kinds of product 0.5 tons should be provided. Since it is only in the larger plants that direct expansion freezers are found, we shall consider only the brine cooled freezer.

The research committee of the International Association of Ice Cream Manufacturers recommends the circulation of 40 to 50 gallons of brine per minute for each 40-quart freezer capacity. Since there are 7.48 gallons in one cubic foot, the cubic feet per minute required are $50/7.48 = 6.7$. Brine velocity should not exceed 150 feet per minute. The area of pipe to carry the brine to and from the freezer is then $6.7/150 = 0.0445$ square feet or 6.4 square inches. A 3" pipe has an internal area of 7.39 square inches, and should be used, since a 2½" pipe has an area of only 4.79 square inches. If the smaller size pipe were used, the power consumption of the brine pump would be excessive on account of the greater friction loss. The successful operation of brine freezers is largely a matter of securing an ample supply of cold brine during the freezing process and money spent for large circulating pipes is well spent.

21. **Brine for freezing.** The freezing process which we have chosen for illustration requires the removal of 4,800 B.t.u. per hour. Since the plant may be called on to operate at this rate for several hours, provision should be made for removing the heat from the brine at the same rate. To provide for losses and variations in demand on the freezers, removal of heat at the rate of 6,000 B.t.u. per hour is recommended. A reasonable brine temperature is $-5^\circ$ F. This should be a calcium chloride brine of about 1.2 specific gravity. Its specific heat is about 0.72. Ammonia evaporating temperature at $-20^\circ$ F. provides for a temperature difference of 15 degrees.

The surface required in the evaporating coils is calculated by use of equation (3).

\[
H = KS_t_d
\]
\[
S = H/K_t_d
\]
A reasonable value for the coefficient of heat transfer between comparatively still brine and evaporating coils is 10.

\[ S = \frac{6,000}{10 \times 15} = 40 \text{ square feet.} \]

If one inch pipe is used the amount required is \( 40 \times 2.9 \) or 116 lineal feet. The heat is removed from the brine very rapidly at the beginning of the freezing process and sufficient brine should be provided to prevent too rapid a rise in temperature. A reasonable condition is the removal of heat in the freezer at a rate three times as fast for a short period as the removal rate in the brine cooler. The 3,000 B.t.u. which the evaporator absorbs in thirty minutes must be absorbed by the brine in about ten minutes. This means that at the end of the first ten minutes, the brine contains about 2,000 more B.t.u. than when at its lowest temperature \(-5^\circ F\). If the temperature rise is limited to two degrees, the weight can be calculated:

\[
H = CW (t_2 - t_1)
\]

\[ H = \text{Heat absorbed, 2,000 B.t.u.} \]

\[ C = \text{specific heat; about 0.72.} \]

\[ W = \text{weight of brine in pounds} \]

\[ t_2 = \text{high temperature; } -3^\circ F. \]

\[ t_1 = \text{low temperature; } -5^\circ F. \]

\[ W = \frac{C}{H} (t_2 - t_1) \]

\[ = \frac{2,000}{0.72 (2)} = 1,390 \text{ pounds} \]

At ten pounds per gallon this is 139 gallons. The tank must be large enough to hold 139 gallons of brine and 116 lineal feet of one-inch pipe or about 150 gallons total. A somewhat larger tank would be desirable because of its greater capacity for storing refrigeration. The tank should be insulated with about eight inches of cork or the equivalent, and the brine should not be used for any other purpose.

22. **Hardening.** The frozen cream when placed in the hardening room requires the removal of about 66 B.t.u. per pound, according to Kennedy, to reduce its temperature to \(-10^\circ F\). Cooling of 800 pounds per day will require the removal of \( 800 \times 66 = 52,800 \) B.t.u. or \( 52,800/288,000 = 0.18 \) tons. The hardening room should be insulated with eight inches of cork board, it should be maintained at a temperature of about \(-10^\circ F\) and should have in it about two lineal feet of 2" pipe or three lineal feet of 14" pipe per cubic foot of space. One and one-half to two cubic feet of space are recommended per gallon of daily production. Hardening rooms with fairly high ceilings of the order of 10 feet with a large part of the cooling surface near the top are more satisfactory than rooms with low ceilings. The air circulation is much better and the temperature throughout the room is more uniform. Fan circulation also is very effective in speeding up the hardening process.

VI. **ICE MANUFACTURE**

23. **Process.** Water for ice making will probably be at a temperature of about 70° F. and the brine at about 14° F. The process then will be:

- Cooling the water from 70° F. to 32° F. during which 38 B.t.u. are removed,
- Freezing during which 144 B.t.u. are removed, and cooling of the ice to
brine temperature which requires the removal of \((32 - 14) \times 0.5\) or 9 B.t.u. The total heat removed then is \(38 + 144 + 9\) or 191 B.t.u. for each pound of ice produced. If 20% is added to compensate for losses, the total refrigeration to be provided is 191 + 38 or 229 B.t.u. per pound of ice. To make one ton of ice in twenty-four hours, \(2,000 \times 229\) or 458,000 B.t.u. must be removed. This is 458,000/288,000 or 1.59 tons of refrigeration. The evaporating coils in ice tanks are of several forms and the heat transfer coefficient varies from 15 to 80 depending on the type of coil.

With a coefficient of 15 the evaporator surface required is calculated as follows:

\[
H = \frac{KS (t_b - t_r)}{H} = \frac{1.59 \times 12,000}{15 (14-5)} = 141.5 \text{ square feet}
\]

If \(\frac{1}{2}\) inch pipe is used, the number of lineal feet required is \(2.3 \times 141.5\) or 325. With more efficient coils, the surface can be made less, but care should be exercised to avoid using too little surface. It is not good economy to reduce the evaporating surface too much.

VII. GENERAL

24. Check valves. In many plants the coils of two or more evaporators are placed in series. A hardening room and a milk storage room may be so piped that the refrigerant first enters the hardening room coils and then passes through the coils in the milk storage room. In this case a valve should be placed in the line where it enters the milk storage room. This valve should be closed when the machine is shut down. Unless this is done when the compressor is stopped, the higher temperature in the milk storage will cause enough evaporation to cause the pressure to build up in the coils. The temperature in the coils will rise correspondingly and may easily reach a higher level than that of the hardening room. In this case the hardening room will be warmed instead of cooled.

25. Corrosion. Wherever brine is used there is danger of corrosion. Either sodium chloride or calcium chloride brine can be made practically noncorrosive by treatment to maintain an alkalinity of slightly over 7 on the pH scale. If there are no surfaces except iron with which the brine makes contact the pH may be carried at about 8, but in ice tanks where the brine makes contact with galvanized cans it should be very nearly neutral.

26. Charge. It is very necessary that sufficient refrigerant be kept in the system at all times. In small plants the receiver should be large enough to hold the entire refrigerant supply and during normal operation there should be enough surplus refrigerant to keep the receiver about half full.

27. Flooded operation. There is a considerable tendency toward the use of "flooded" operation. By this is meant that the evaporating coils are filled with liquid. As the liquid evaporates, the gas returns to an
"accumulator." Some unevaporated refrigerant is usually carried out of the evaporator with the vapor. The accumulator is fairly large and serves to separate the liquid from the vapor. The vapor returns to the compressor and the liquid goes to the bottom of the accumulator and returns to the evaporator. The level of liquid in the accumulator is often maintained constant by a float-controlled valve.

Flooded operation with float control has several advantages. Regulation is automatic. The amount of refrigerant admitted is equal at all times to the amount being evaporated. The condition of the vapor returning to the machine is practically dry; that is, not superheated or not wet. The heat transfer in the evaporator is greater because a large part of the surface is in contact with liquid refrigerant rather than gas.

VIII. ACKNOWLEDGMENTS

The author has drawn rather freely on such publications as those of the National Association of Practical Refrigerating Engineers, the Ice Cream Trade Journal, and others. He has endeavored to list those publications from which he has obtained helpful material in the bibliography and expresses his thanks to the authors of them. Thanks are also due Professor S. H. Graf, Head of the Department of Mechanical Engineering and Director of Engineering Research, for his cooperation, and to William H. Paul, Assistant Professor of Mechanical Engineering, for drawing the sketches and curves.

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### Appendix A

**SATURATED AMMONIA: TEMPERATURE TABLE**

<table>
<thead>
<tr>
<th>Degrees F.</th>
<th>Pressure per square inch absolute</th>
<th>Specific volume per pound</th>
<th>Latent heat of liquid</th>
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<td></td>
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Appendix B

Volumetric efficiency vs. compression ratio for vertical single-acting and horizontal double-acting compressors.
Appendix C

Compressor characteristics for various suction pressures with fixed effective displacement and discharge pressure.
### HEAT TRANSFER COEFFICIENTS

**Can Ice Making Piping:**
- Old style feed hand controlled, non-flooded: 12 to 15
- Float controlled, flooded: 20 to 30

**Ammonia Condensers:**
- Atmospheric, parallel flow type: 50 to 60
- Atmospheric, counter flow, bleeder type: 125 to 200
- Double pipe type
  - Velocity of water, feet per minute: 100, 200, 300, 400
  - K: 110, 180, 225, 250
- Shell and tube (vertical single pass type)
  - Gallons per minute per square foot of tube surface:
    - 0.1: 70, 120, 150, 170, 190, 200, 210, 220
    - 0.2: 70
    - 0.3: 70
    - 0.4: 70
    - 0.5: 70
    - 0.6: 70
    - 0.7: 70
    - 0.8: 70
- Shell and tube (multipass type): 150 to 300

**Coolers (Baudelot type)**
- Direct expansion:
  - Water: 60
  - Milk: 60
  - Cream: 50
  - Brine
    - Water: 80
    - Milk: 80

**Brine Coolers:**
- Shell and tube type
  - Velocity of brine, feet per minute: 50, 100, 150, 200, 250
  - K: 55, 80, 100, 115, 125
- Double pipe type
  - Velocity of brine, feet per minute: 100, 150, 200, 250, 300, 350, 400
  - K: 75, 85, 97.5, 110, 120, 127.5, 135
- Brine coils submerged in still liquid: 12
- Direct expansion coils in still air: 2
- Direct expansion coils in still liquid: 10 to 13
- Direct expansion coils in circulated liquid
  - Velocity of liquid in feet per minute:
    - Non-flooded: 13.4, 15.0, 16.4, 17.8, 19.0
    - Flooded: 17.9, 20.0, 21.9, 23.7, 25.3

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SCHOOL OF HOME ECONOMICS (B.A., B.S., M.A., M.S. degrees)

SCHOOL OF PHARMACY (B.S., M.S. degrees)

SECRETARIAL SCIENCE (B.S.S. degree)

The Graduate Division (M.A., M.S., Ch.E., C.E., E.E., F.E., M.E., Ph.D. degrees)

The Summer Session

The Short Courses

RESEARCH AND EXPERIMENTATION

The General Research Council

The Agricultural Experiment Station—

The Central Station, Corvallis

The Branch Stations at Union, Moro, Hermiston, Talent, Burns, Astoria, Hood River, Pendleton, and Medford

The Engineering Experiment Station, Corvallis

EXTENSION

Federal Cooperative Extension (Agriculture and Home Economics)

General Extension Division.