COMPRESSOR DESIGN, APPLICATION, AND GENERAL SERVICE (PART 1)

By: John L. Zant and
By: Bernard A.Nagengast

FOREWORD

This article is confined to small compressors as used in commercial refrigeration and air conditioning applications, up to about 50 horsepower. It will be assumed that the reader has some knowledge and experience in refrigeration servicing. In other words, it is not intended that this article will make the reader an expert application engineer or refrigeration service man, if he has had no previous training.

An attempt will be made to avoid repetition of subjects previously covered in other articles on this disk. In this regard the sections previously issued on principles of refrigeration, two and three stage refrigeration systems, refrigerants, lubrication, electric motors, and in some instances other sections, have touched on matters pertaining to compressors and their application.

HISTORY

Vapor compression refrigeration was attempted as early as 1755 when William Cullen cooled water by drawing a vacuum over it. Oliver Evans proposed a closed circuit vapor compression system in 1805. However, it was Jacob Perkins who actually built a workable system in 1834. Several decades passed until commercial refrigeration became marketable, the success of which was helped by practical development of compressors and systems for ammonia, sulfur dioxide, and methyl chloride in the 1870s.

These early compressors were operated at low speeds, rarely in excess of 100 rpm, and, as a result, were large and heavy. Advancements in valve design, shaft seals, bearings, and lubrication systems resulted in a gradual increase in the design speed of the compressors. This allowed the compressors to become smaller for a given horsepower as increased displacement was obtained from higher speed operation.

The development of electric power systems in the 1880s, together with the invention of the electric motor opened up a large market for refrigeration, since steam plants were now unnecessary. The problem with shaft seal leaks prompted a search for a successful hermetically sealed refrigeration system. Although various systems were actually built prior to the twentieth century, the possibility of a practical hermetic system was evident in the marketing of the Audriffen-Singrun refrigeration machine (nicknamed the "dumbbell") beginning in 1907. The success of this system prompted much research, resulting in the first hermetic motor compressors as we know them, in the middle 1920's.

Advances in refrigeration technology resulted in a tremendous expansion of vapor compression refrigeration into the commercial and domestic fields in the 1920's. This was made possible by development of automatic temperature and refrigerant flow controls, as well as mechanical shaft seals in the early years of the twentieth century. A major hurdle was overcome in 1929 with the development of the "Freon" products by DuPont, which served as odorless, nonflammable replacements for sulfur dioxide and methyl chloride.

Although open-drive refrigeration equipment was being reduced in physical size due to increasing speeds, the hermetic and semi-hermetic motor compressor began to replace belt and direct drive equipment, first in domestic refrigeration, and then in the commercial area. After a slow start in the 30s, this trend accelerated after World War II, as sealed systems proved their reliability and lower cost. The size limit of hermetic and semi-hermetic compressors has gradually moved up, so that open-drive equipment has been replaced in all but the most strenuous applications up to about 50 horsepower. The trend has been towards the use of welded hermetic compressors for air conditioning units. However, the need for long life and serviceability still provides a market for semi-hermetic and open-drive equipment where the higher initial equipment cost is a secondary consideration. An upsurge in the popularity of rotary compressors is also evident, with recent applications in not only refrigeration but air conditioning as well.

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TYPES

The compressor is often called the heart of any refrigeration system. It is that important item of the system which provides the force to withdraw the gas from the evaporator, force it into the condenser, and keep this circulation going.

Many different types of compressors have been developed and produced in the history of refrigeration. Of those which have become popular, however, we might classify them into a few general types. The following will briefly cover the advantages and disadvantages of these general types.

OPEN TYPE VERSUS HERMETIC

By "open type," we mean a compressor driven by an external motor, either belt-driven or directly connected. Since this involves a shaft extending through the crankcase of the compressor, it requires a shaft seal. Opposed to this type compressor is the hermetically sealed compressor, in which the motor is incorporated within the same housing as the compressor. Thus the hermetically sealed compressor has no shaft extending through the crankcase and therefore requires no seal. Each type has some advantages over the other. An open type compressor, if of the belt-driven type, can be very flexible. In other words, its speed can be varied so that a single compressor can often be used for two or three different horsepower sizes of units. By merely changing the size of motor pulley in most instances this same compressor can be used not only with different sizes of motors, but can be used for high, medium, or low temperature applications. This is the most outstanding advantage of an open type compressor as compared to a hermetic type. Other advantages are that this type compressor can be used with motors available for odd voltages and frequencies for which hermetic type compressors are not produced. This is particularly true in the case of direct current. Hermetically sealed compressors as presently designed cannot be used with direct current, since direct current necessitates a commutator and brushes which could not operate in the atmosphere of oil and refrigerant. Where only direct current is available, therefore, open type compressors must be used unless a converter is installed to convert the direct current to alternating current of some standard voltage. Another factor in favor of the open type is its slow speed, which results in long life.

Open type compressors are always field serviceable, which is not true of all hermetically sealed compressors. In the event of a motor burnout, it is probably easier to replace the motor on an open type system than on a hermetic system where the motor is exposed to the refrigerant. In the event that a motor burns out in a hermetically sealed system, the entire compressor has to be replaced, returned to a repair shop or factory to be dismantled and reconditioned, a system cleaner be installed or the system flushed out with liquid refrigerant, etc., in order to eliminate all possibility of acid which might have resulted from the burning of the insulation being circulated through the system.

As compared to open type compressors, however, hermetically sealed compressors have many outstanding advantages. Perhaps the greatest of these is the elimination of the shaft seal. Shaft seals are vulnerable to dirt, temporary failure of lubrication, anything abrasive which might accumulate in the system, such as scale, and physical damage due to rough handling, etc. Although the seals produced today are much improved over those of 20 or 30 years ago, they are still a potential source of trouble, especially on a low temperature system where the low-side pressure might be under considerable vacuum. In such a case, a leak at the seal causes air and moisture vapor to enter the refrigeration system, which is more serious than the loss of refrigerant from the system. Another problem is that the seal can dry out and leak if the system is shut down for prolonged periods, such as air conditioning equipment shut off for the winter.

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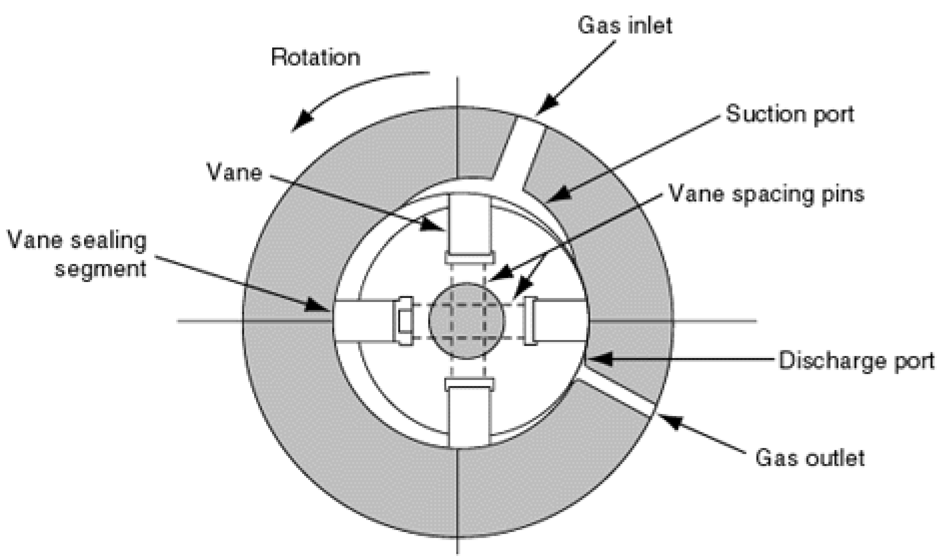
Other advantages of the hermetic type compressor are that it is smaller-more compact, more free of vibration, and has its motor continuously and positively lubricated. They have no belts which need adjusting and eventual replacement.

A design variation of the hermetic compressor is the accessible or semi-hermetic type, which combines the sealed system qualities of the hermetic and the serviceability and ruggedness of the open type compressor. The semi-hermetic compressor incorporates the motor and compressor in a bolted casting; thus the compressor can be disassembled for trouble-shooting and service. This feature also allows the compressor manufacturer to rebuild the compressor easily.



Cross-Section of a Vane Type Rotary Compressor

RECIPROCATING VERSUS ROTARY

The next general design consideration is reciprocating vs. rotary, both of which have been used and are being used at the present time. Rotary type compressors are quite popular for household refrigerator applications and have recently been applied in air conditioners as well. For economy of manufacture, the rotary type must be produced in very large quantities. Considerable precision of machining is required to provide satisfactory performance and compressor efficiency. Rotary type compressors generally need a check valve in the suction line to prevent the high pressure gas in the condenser from leaking back through the compressor into the low side during the off cycle. Another disadvantage of the rotary type compressor is that it cannot tolerate any liquid refrigerant floodback since the suction line directly enters the compression chamber. The rotary type compressor is ideally suited to applications operating at a low ratio of compression, and where large volumes of gas must be circulated, such as is the case with some refrigerants.

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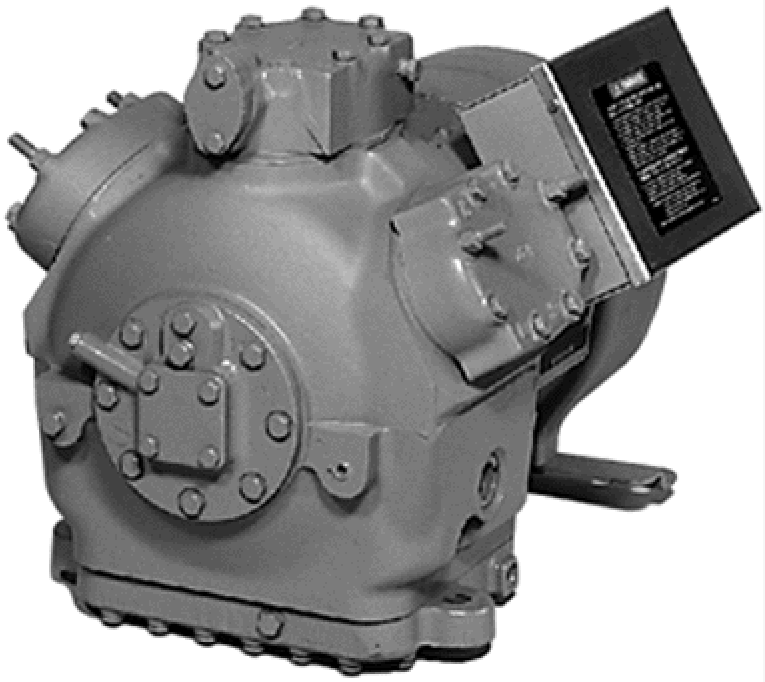

Semi Hermetic Reciprocating Compressor

The reciprocating type compressors are adaptable to small systems as well as very large ones, have good efficiency, and are known for their long life. Reciprocating type compressors are being built in both the open type and hermetic type. Rotary type compressors have been built for belt-drive, but most rotary compressors produced so far have been of the hermetic type.

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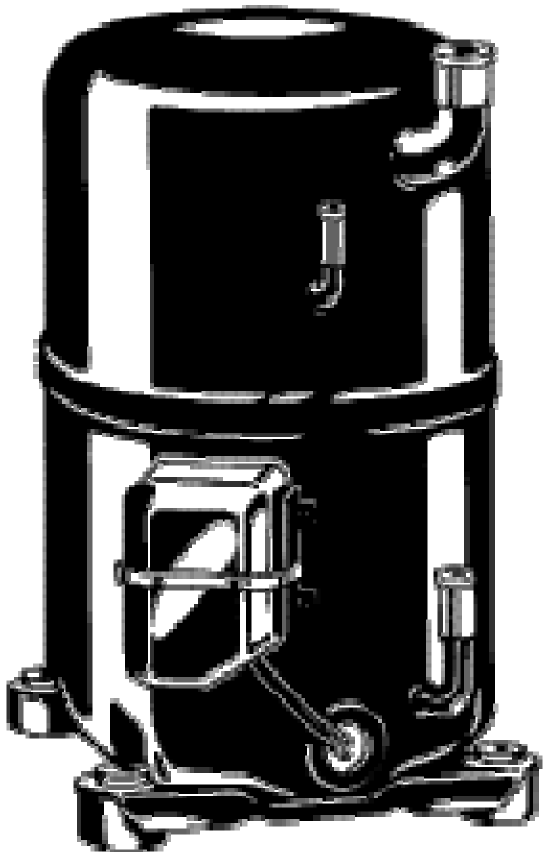

Suction Cooled Hermetic Compressor

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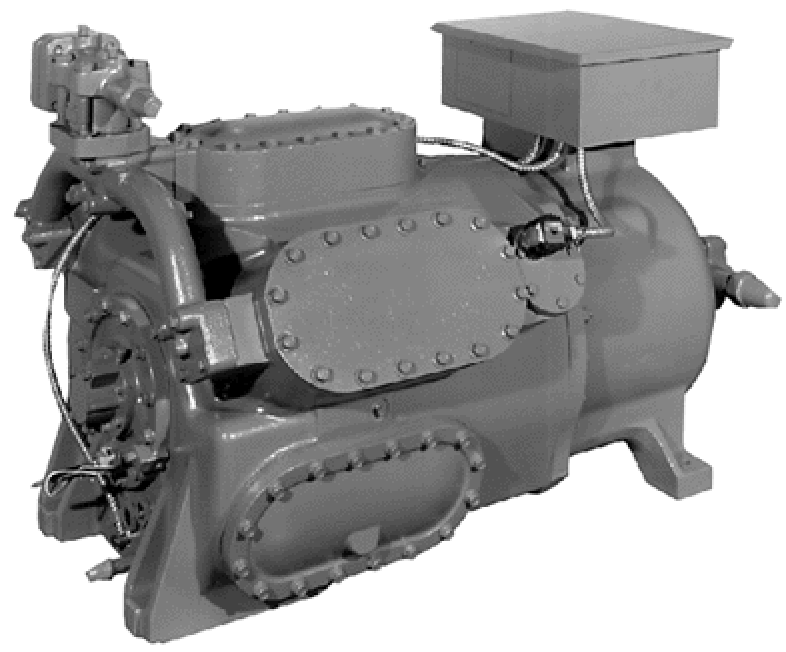
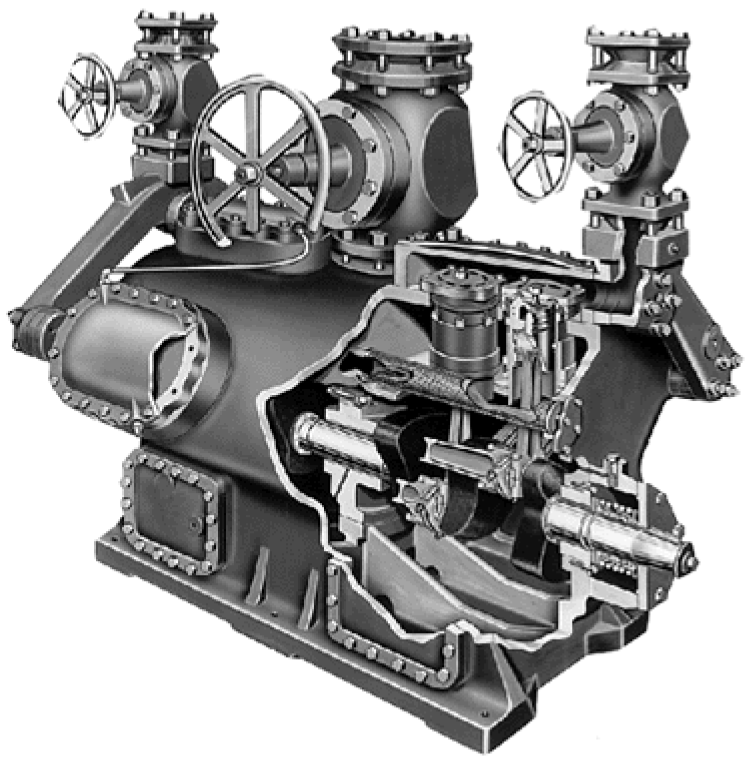


Combination Suction and Air-Cooled Semi Hermetic Compressor

SUCTION COOLED VERSUS AIR OR WATER

In regard to hermetic type compressors, some means has to be provided to dissipate the motor heat. With open type or belt-driven compressors, this motor heat is transferred to the surrounding air. With hermetic type compressors, however, the motor being enclosed in the crankcase, so to speak, some means has to be provided to pick up heat generated from motor losses. There are four ways this can be done:

A. Suction gas can be passed through or around the stator winding to pick up the motor heat, and the refrigerant thus carries this heat to the condenser, where it is dissipated.



Water Jacketed Semi Hermetic Compressor

B. Air can be blown over the motor housing, and the motor heat dissipated in this way.

C. A combination of A and B.

D. The motor can be water-jacketed, or a water coil applied to the motor housing, through which water can be circulated to pick up the motor heat.

Suction cooling of the compressor motor is very effective where the compressor is working at high suction pressure. In this case a large amount of refrigerant is circulated for the amount of heat to be picked up, and thus very little superheating of the suction gas in passing through or around the motor results. At very low suction pressure, however, suction cooling has the disadvantage of excessively superheating the suction gas before it goes under compression. Upon further heating under compression, therefore, the discharge gas temperature sometimes reaches dangerous limits, thus leading to breakdown of oil, rapid chemical reactions in the case of air and moisture in the system, and an overall increase in the compressor temperature, thus thinning the oil and impairing lubrication. For low temperature applications suction cooled compressors are sometimes used with supplementary cooling provided by a fan blowing air over the compressor.

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Air circulated at high velocity over the motor housing is an effective way of picking up the motor heat. This can be done by placing the compressor in the path of the air off the condenser in the case of an air-cooled unit, or supplying an auxiliary fan to blow against the motor housing.

Water cooling of the motor is often incorporated in the case of water-cooled condensing units. The water on its way to the condenser can first be circulated around the motor housing, thus picking up the motor heat and dissipating this heat in the same way that the condenser heat is dissipated. The demands of the condenser are so much greater than the amount of water required to cool the motor that, as a result, the water passing around the motor has a very low temperature rise. In general, the motor heat amounts to only 5 to 10 % of the heat rejected by the condenser. Water-cooled compressors incorporate water jackets, or have water coils wrapped around the compressor body.

SELECTION FACTORS

The user of a compressor has to have in mind the following factors:

PERFORMANCE

In most instances, performance has to be considered. By performance, we mean capacity, efficiency, and economy of operation. For a given hp or a given size compressor for use with a given refrigerant it is desirable to have a compressor producing the maximum Btu per hour capacity. The term "efficiency" is a very broad one, but we generally mean by an efficient compressor, one that has good capacity even at very low suction pressure and at high ratio of compression. The efficiency of the compressor is often judged by the Btu per cubic foot displacement; in other words, a compressor is more efficient if it has a greater Btu per hour capacity, per cubic foot, per hour displacement, for a given set of conditions. The measure of economy of operation is generally judged by the brake hp per ton factor, or Btu per watt. The lower the brake hp per ton factor and the higher the Btu per watt factor, the more desirable the compressor would be and the greater would be its economy of operation. Brake hp per ton and Btu per watt are of similar nature, except that brake hp per ton involves the compressor only, whereas Btu per watt involves both the compressor and motor driving it. In the latter case, it also takes into account the efficiency of the drive.

DEPENDABILITY

In most installations, someone is concerned with the cost of operation and obtaining sufficient capacity for all conditions that may be encountered. Perhaps the factor that has the greatest appeal to the fixture manufacturer, the dealer, the user, and service persons, is dependability. Therefore, the designer and manufacturer have to leave no stone unturned in their efforts to avoid all possible weaknesses that might lead to failure in the field. It is not only the cost of repairing the compressor that must be considered, but the product loss due to spoilage, the loss of sales, and the loss of good-will, which might result from failure of a compressor.

COMPACTNESS

Especially in the case of self-contained fixtures and cabinets it is desirable to use a compressor which will occupy the least amount of space. This is becoming increasingly important in the field of air conditioning. In the case of self-contained merchandising fixtures, the smaller the compressor, the less space is occupied by the condensing unit. Therefore, there is an increase in the volume of products stored. The trend, therefore, in compressor design is toward higher speeds, which permit reduction in physical size.

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FLEXIBILITY

Flexibility is a factor which has a lot to do with the determination of how many compressor models are required to fill a certain range. In this respect, belt driven units can be used very often for two or three different HP sizes, providing they are not operated beyond the design maximum speed. The displacement in cubic feet per hour can be changed merely by varying the size of motor pulley or the diameter of the flywheel. On the other hand, a hermetic compressor is generally designed to operate at a given speed and therefore has a fixed displacement. Flexibility also involves the use of a given compressor for both air-cooled and water-cooled condensing units, use with different refrigerants, and applications involving variable degrees of suction gas superheat. Although being able to apply a given compressor at variable speeds has its advantages, a problem somewhat difficult to control at times is the variation in amount of oil circulated and obtaining satisfactory lubrication at all speeds.

Compressors are engineered by the manufacturer for specific refrigerants and evaporating-condensing temperature ranges. The compressor manufacturer should always be consulted first before a decision is made to use a compressor outside its design range, or with a different refrigerant. This is especially critical in the case of hermetic compressors.

LIFE

Long life is, of course, desirable in any product. Compressors built today are expected to provide many years of constant trouble-free and quiet operation. In many applications the compressors are called upon to run 24 hours per day and 365 days per year. Such continuous operation, however, is often not as hard on a compressor as cycling operation, which taxes electrical components and can result in oil return problems. In order to insure long life, it is necessary for the manufacturer to select the best possible materials, hold all tolerances to an absolute minimum, and engineer the compressor to withstand not only the normal operating conditions but some periodic abnormal conditions. At times some performance has to be sacrificed in order to assure the compressor being able to stand up under conditions of some liquid slugging, excessive discharge pressure, and for short periods of operation at suction pressures above, and sometimes considerably above, the normal operating range.

COST

Initial cost of the equipment is a factor in every job. First cost must be judged against the quality and features of the compressor and condensing unit, depending on the customer's needs. Where long life, flexibility or serviceability are important, higher first installation costs are justified. On the other hand, where these factors are not important, lower cost equipment is desirable. For instance, it would be wasteful to install an expensive, long lasting compressor in a building scheduled for demolition next year!

SERVICEABILITY

A compressor which can be serviced in the field is advantageous to the user and the service man. This is especially important in the case of compressors used in commercial refrigeration applications and commercial and industrial air conditioning, etc. where a failure of the compressor can be very costly. For example, in a supermarket the compressor operating the refrigerated cases cannot be shut down for long without creating a considerable loss of sales for the merchant. Likewise, in some air conditioning applications where it is essential that proper temperature and relative humidity be maintained, a cessation of operations or loss of business might result from the prolonged shutdown of the air conditioning system. Field serviceability is also of paramount consideration in any application where the equipment is installed in some remote spot where it is not an easy matter for a service person to go to a supply house to obtain an exchange compressor. Field-serviceable compressors are more expensive to manufacture than those that are not field serviceable. In some applications, field serviceability is sacrificed in order to hold down the cost of the product. Serviceability is of prime importance if the equipment is to be installed where replacement would be difficult and costly.

APPLICATION

No compressor can be built to provide good performance under all conceivable conditions and for all applications. Therefore, compressors are generally designed, manufactured, and sold for use in given applications, or within a given range of conditions. Some of the conditions which affect the compressor in this regard are as follows:

SUCTION PRESSURE

Compressors are generally designed to operate up to a certain maximum suction pressure. As the suction pressure increases, the motor load increases also, and at a certain point, the motor will be loaded to its maximum. In this regard compressors are generally designed for either high suction applications, medium suction applications, or low suction pressure range. A compressor designed for the high temperature range can sometimes be used for medium temperature or low temperature installations, but will not have the capacity that a compressor would have if it were designed for that particular medium temperature or low temperature range. It might be used for a medium or low suction range, but the motor would be lightly loaded, and a reduction in capacity would also result. Medium temperature compressors might also be used for low temperature, with reduced motor load and capacity resulting. In general, internally spring mounted hermetic compressors designed for high or medium temperature will run excessively hot in low temperature applications. Single stage compressors should not be applied below -40°F evaporating temperatures as high compression ratios, high discharge temperatures and impaired cylinder lubrication may occur. In general, the compressor manufacturer's allowable range of evaporating temperatures for the particular compressor model is the criterion to avoid premature failures.

DISCHARGE PRESSURE

Increased discharge pressure increases the load on the motor, reduces refrigerating capacity, and makes a compressor run hotter. In some cases the displacement selected permits the compressor to be operated not above a certain limiting discharge pressure. For example, sometimes hermetic compressors are designed for use in water-cooled units which normally operate at a relatively low discharge pressure. If such compressors are operated in connection with air-cooled condensers, an overloaded motor might result in high ambient.

In the case of belt-driven compressors, for the same suction pressure range and the same motor HP, the compressor is generally run at a slightly higher rpm on a water-cooled unit than in an air-cooled unit, when the same model of compressor is used in both air and water-cooled condensing units. This is permissible, since the lower discharge pressure of the water-cooled unit allows a higher compressor speed for the same motor load.

Excessive discharge pressures should be avoided whenever possible by adequately sizing the condenser, proper condenser maintenance, etc.

SUPERHEAT OF SUCTION GAS

Compressors and condensing units are usually rated at a given suction gas temperature. (This is not to be confused with "evaporating temperature"). Suction gas temperature is the actual temperature of the gas entering the compressor at the suction shut-off valve.) Too low a suction gas temperature or too low a superheat of the suction gas entering the compressor should be avoided. Under such conditions liquid refrigerant might enter the compressor, causing great stress on the compressor valves, wrist pins. rods, shaft, etc., and also causing excessive foaming of the oil in the crankcase, and numerous other damaging effects. In addition, the rated capacity of the compressor will not be obtained.

Too high a suction gas temperature will cause the compressor to run hot, and in the case of suction cooled compressors, may provide insufficient motor cooling. As a result, the motor may cycle on the protector, and its life considerably shortened.

For best overall performance, the suction gas temperature should be controlled at a superheat of not less than 15°F, and thus superheated either in the evaporator or by means of a suction-liquid line heat exchanger. Superheating of the suction gas in a suction line uninsulated and exposed to high ambient or solar heat causes a definite loss of refrigeration effect in the evaporator, which is where the refrigeration is wanted. Maximum capacity in the evaporator will be obtained by subcooling the liquid refrigerant and super-heating the suction gas in a heat exchanger. However, too high an actual suction gas temperature, especially in the case of suction-cooled hermetic compressors, must be avoided. For best performance the suction gas temperature should not exceed 80 to 90°F, but in the case of suction-cooled hermetic compressors operating in low temperature systems, it would be more desirable to avoid suction gas temperatures over 40 to 50°F.

DISCHARGE GAS TEMPERATURE

The discharge gas temperature is affected by each of the factors enumerated previously—suction pressure, discharge pressure, and superheat of the suction gas. In addition, it varies with the characteristics of the refrigerant itself.

We might sum this all up by stating that discharge gas temperature, for a given refrigerant, depends on suction gas temperature and ratio of compression. The higher the actual compression ratio or suction gas temperature becomes, the higher the discharge gas temperature. The ratio of compression is the absolute discharge pressure (psig plus 14.7) divided by the absolute suction pressure (psig plus 14.7). So one can readily see that for a given suction pressure, the ratio of compression and discharge gas temperatures will go up as the discharge pressure is increased. And by the same token, for a given discharge pressure, the ratio of compression and discharge gas temperature will go up as the suction pressure decreases. That is why the discharge gas temperature—and, in general, the entire compressor—will run considerably hotter in an air-cooled condensing unit operated at low suction pressure than it would in a water-cooled condensing unit operated at medium or high suction pressure.

The characteristic of a refrigerant which affects discharge gas temperature is the ratio of its specific heat at constant pressure to its specific heat at constant volume, known as the factor $\frac{C_p}{C_v}$.

The higher this factor is, the higher the gas will be heated under compression. Since R-502 has about the lowest value for this factor, all conditions being the same, a compressor operating with R-502 will have a lower discharge gas temperature than it would have operating with any other common refrigerant. For the same suction gas temperature, the discharge gas temperature will run higher with R-12, and higher yet with R-22.

The discharge line temperature should never be allowed to go above 250°F on commercial refrigeration systems, although the lower this temperature is, the better. High discharge gas temperatures cause thinning of the oil, and thus impairing lubrication, and promoting more rapid oxidation.

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CAPACITY VERSUS POWER INPUT

It is naturally desirable to obtain the maximum capacity from the compressor without overloading the motor. As stated previously, capacity varies with suction pressure and discharge pressure. The motor load also varies with these same two factors. This can best be illustrated by showing performance curves of a typical condensing unit.

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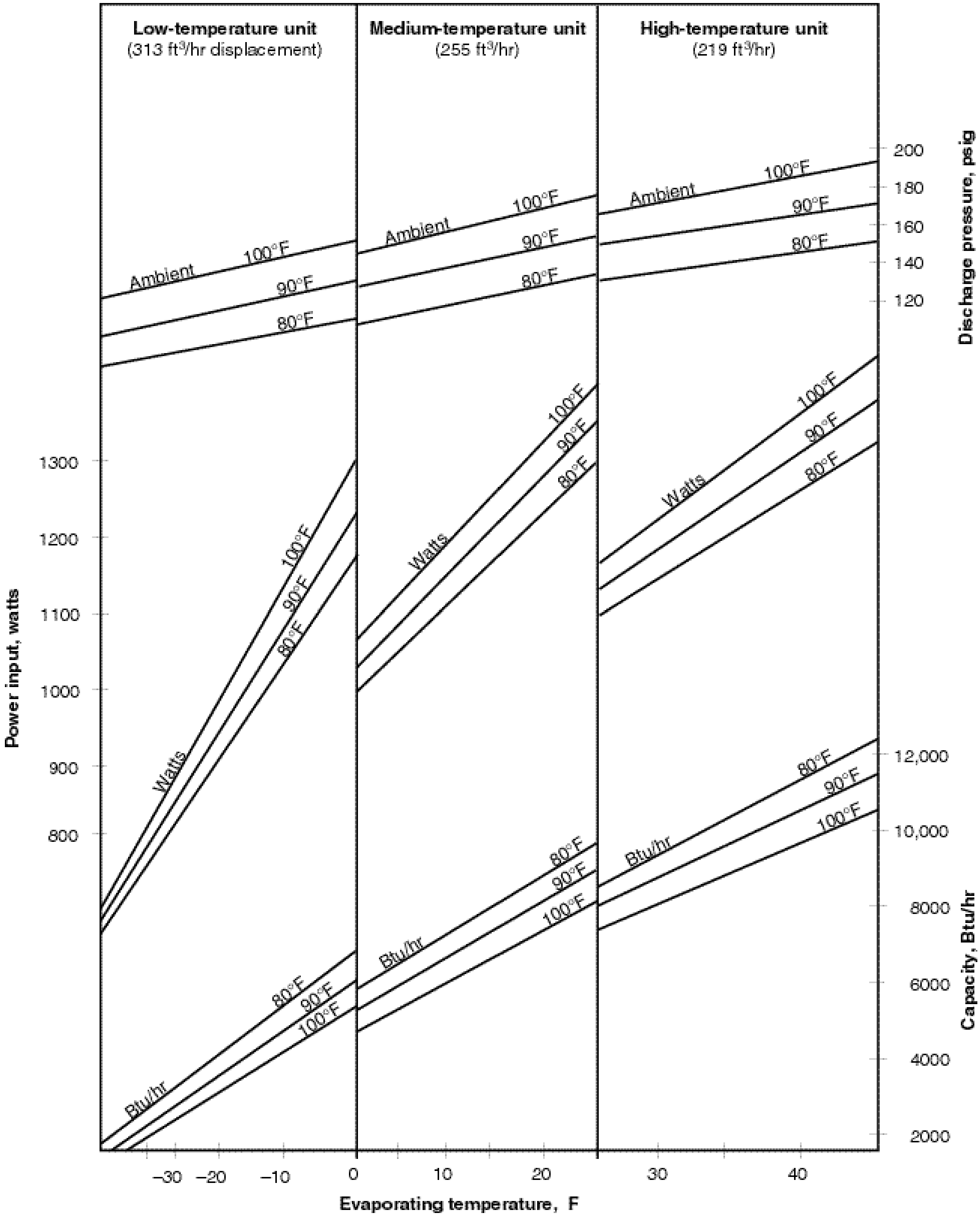

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Figure 11—Performance of 1 H.P. Air-Cooled Condensing Unit

The above is a graphical representation of capacity (Btu/hr), power input (Watts), and discharge pressure (psig) for three models of a 1 hp condensing unit, air-cooled: a low-temperature model, a medium-temperature model, and a high-temperature model. The capacity, power input, and discharge pressure are plotted against evaporating temperature for ambient temperatures of 80, 90, and 100°F. You will note the power input curves are quite similar for all three models, but the capacity curves show quite a considerable difference from one model or suction range to another. The displacements of these compressors were selected to provide the motor loads shown. In spite of the increasing displacement as we go from high to medium to low temperature models, the capacity actually decreases. The reason for this is that as we go down in suction pressure, the ratio of compression increases, and as the ratio of compression increases, the volumetric efficiency of the compressor becomes less.

One will note also from these performance curves that the discharge pressure runs higher for the high-temperature model than for the medium- or low-temperature models, and the discharge pressure runs higher for the medium-temperature model than for the low.-temperature model. This is because the condenser load varies with the capacity of the condensing unit-the higher the refrigerating capacity, the greater amount of heat the condenser has to dissipate. Therefore, with the same ambient, the same amount of condenser surface, and the same volume of condenser air circulated, the condenser must operate at a greater temperature differential, and discharge pressure goes up as the capacity becomes greater.

In spite of the condenser load being less for the low temperature model, it is customary for manufacturers to supply the same condenser for low, medium, and high temperature models. It is desirable to hold the discharge pressure down as much as can be done practically, for low temperature models, in order to hold the ratio of compression as low as possible.

Referring to the performance curves in the above graphic, we arrive at the following conclusions:

A. As suction pressure increases, for a given model of condensing unit, the capacity, power input, and condenser load all increase (which accounts for most condensing units being rated up to a certain limiting suction pressure or evaporating temperature).

B. For a given model of condensing unit, as the discharge pressure increases, the power input increases, and the capacity decreases. Therefore, the economy factor "Btu/Watt" decreases as the discharge pressure is elevated.

C. For a given evaporating temperature (or suction pressure), the medium temperature model will have more capacity, but a greater motor and condenser load as well, as compared to the high temperature model. Similarly, for a given evaporating temperature, the low temperature model will have a greater capacity and greater motor and condenser load than the medium temperature model. The reason for this is the greater displacement of the low temperature model with respect to the medium temperature model, and the greater displacement of the medium temperature model, as compared to the high temperature model. Although the increased displacement results in greater capacity, the motor load must be considered when selecting equipment, and, in some cases, the model with less displacement might have to be used to permit the motor remaining on the line during severe operating conditions, if frequent "pull-down" conditions have to be encountered, etc.

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SELECTION OF A CONDENSING UNIT

The more important factors to be considered in the selection of a compressor or condensing unit will now be taken up.

TYPES OF UNITS

The first step in the selection of a compressor or condensing unit is to decide the type of unit best suited to the application, i.e., whether it is to be air-cooled, water-cooled, combination air-water, evaporative condenser type, or some other. We will not go into the relative merits of the various types in this section, but many factors must be considered to assure the best performance, economy of operation, and freedom from trouble. When the type of condensing unit has been selected, the condensing medium temperature can be determined, as it governs the discharge pressure of the compressor, thus affecting its capacity.

For an air-cooled condensing unit the average maximum ambient temperature would have to be known. For a water-cooled unit the maximum temperature of the water supply must be determined. For a water-cooled unit connected to a cooling tower, or an evaporative condenser unit, the average maximum wet bulb temperature would have to be known.

As shown in the performance curve illustration (Figure 11), air-cooled units are rated at given evaporating temperatures and ambient temperatures. Water-cooled units and evaporative condenser units are usually rated at given evaporating temperatures and given discharge pressures, or corresponding condensing temperatures. The latter applies also to ratings of compressors only.

CAPACITY REQUIREMENT

It is, of course, essential that the compressor or condensing unit selected have sufficient capacity to handle the load at average maximum conditions. For best overall performance both the condensing unit and evaporator capacities should be approximately equal to the Btu/hr load at design conditions. Obviously, the load must be known and also the desired hours of operation per day. Actual Btu/hr load divided by desired hours of operation per day, multiplied by 24, equals the capacity requirement for the condensing unit and evaporator.

$$\frac{\text{Btu}/\text{hr. Load} \times 24}{\text{Hrs. Operation per Day}} = \text{Condensing Unit Capacity Req'd}$$

After determining Btu/hr capacity required, the next step is to determine evaporating temperature and suction pressure. This will depend considerably on the differential for which the evaporator is selected. For example, if a cooler is to be operated at 38°F, and the coil is selected for 15° differential, the evaporating temperature in the coil will be 38 minus 15°, or 23°F. For 23°, R-12 has a saturation pressure of 23.2 psig. From this evaporating pressure must be subtracted the pressure drop through the suction line. Let us assume 2.0 psig in this case. The compressor will therefore be operating at a suction pressure of 23.2 minus 2.0, or 21.2 psig, which corresponds to 20°F evaporating temperature. So, although we have a design evaporating temperature of 23°F, the compressor or condensing unit must have the required Btu/hr capacity at 20° evaporating temperature, since it is the pressure corresponding to this evaporating temperature at which the compressor will actually be operating, in view of the suction line pressure drop. With the conditions of evaporating temperature and ambient or condensing temperature determined (as outlined in previous paragraph), the condensing unit can be selected, although in a few cases there are some other factors, such as actual suction gas temperature, which would have some bearing on the capacity requirement.

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It is seldom possible to select a condensing unit having just the exact capacity desired. It is often necessary, or more practical, to use a condensing unit having somewhat more, or less, capacity than the calculated Btu/hr requirement. This simply results in slightly more or less running time in most instances. For example, supposing the load is 12,000 Btu/hr, and a running time of 16 hours per day is desired. Then the actual condensing unit capacity required would be:

$$\frac{12{,}000 \times 24}{16} = 18{,}000 \text{ Btu/hr.}$$

Now let us assume that one model of condensing unit has a rated capacity, at the conditions specified, of 15,000 Btu/hr., and the next larger size has 24,000 Btu/hr capacity at the design conditions. In the first instance, the running time would be:

$$\frac{\text{Btu/hr. Load} \times 24}{\text{Cond. Unit Capacity}} = \text{Hours per Day Operating Time}$$

$$\frac{12{,}000 \times 24}{15{,}000} = 19.2 \text{ Hours per Day}$$

In the other case, with a unit having 24,000 Btu/hr capacity, the running time would be:

$$\frac{12{,}000 \times 24}{24{,}000} = 12.0 \text{ Hours per Day}$$

It would probably be more practical to select the unit with 15,000 Btu/hr. capacity to operate 19. 2 hours per day, assuming the load is based on average maximum conditions.

As for the exact model of condensing unit to select, here the design evaporating temperature must be considered. In general, for evaporating temperature of 0°F, or less, low temperature models are used, above 0° but less than 25°F, medium temperature models, and for operation at evaporating temperatures above 25°F, high temperature models are necessary.

Some manufacturers rate machines for two ranges—such as "high and medium" or "medium and low". In such cases, the condensing units or compressors are actually designed for the higher suction range and simply operate under a more lightly loaded condition in the lower suction range.

There are times, however, when a unit is applied at a suction pressure or evaporating temperature below its normal design range. Supposing, for example, an air-cooled condensing unit is desired having a capacity of 7,200 Btu/hr at 20°F evaporating temperature in a 90°F ambient temperature. Let us assume that a 3/4 hp unit would not have sufficient capacity. Referring to Figure 11, it is evident that the 1 hp medium temperature model has approximately 8,500 Btu/hr at the prescribed condition, which is well above the required capacity. However, the high temperature model operating at 20° evaporating temperature would have almost exactly the desired capacity- approximately 7,400 Btu/hr. It might therefore be preferable to use the high temperature model, even though the application is in the medium temperature range.

Again, it is not always recommended, however, to apply high or medium temperature model compressors for low temperature usage. This applies to hermetic and semi-hermetic compressors of the suction cooled type, as inadequate motor cooling and high discharge gas temperature may result. The compressor manufacturer's recommendations should be adhered to in this regard.

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With belt-driven units it is sometimes practical to reduce the motor pulley diameter, thus reducing the compressor speed and resulting capacity, in order to obtain some exact capacity for some critical application. Precautions are necessary in this regard, however, as most compressors have a recommended minimum speed for proper lubrication, and too small a pulley diameter may cause excessive flexing of the belts, and/or insufficient contact between the belts and pulley to transmit the power of the motor, especially when starting.

As will be pointed out in the next section, motor loading is another important factor that sometimes governs the exact model of condensing unit or compressor best suited for a given application.

MOTOR LOADING

As illustrated in Figure 11, the motor load increases as the suction pressure increases. As a consequence, in a low temperature application, if a prolonged "pull down" must be frequently encountered, it might be advisable to use a condensing unit actually designed for medium temperature range, provided the compressor manufacturer approves the medium temperature model for low temperature operation. As mentioned previously, low temperature units are not usually designed for continuous operation at evaporating temperatures above 0°F, which corresponds to 9.2 psig suction pressure for R-12. On some automatic defrosting systems, however, the compressor may be called upon to operate at 30 psig suction pressure, or even higher, either during the defrosting period (with hot gas defrosting systems), or immediately after the defrosting period. Depending on the evaporator design and various controls that may be incorporated, the compressor motor may or may not be seriously overloaded during these periods. If the motor of a low temperature unit does become overloaded frequently in this way, it is generally a question of whether to select a low temperature unit and protect it with a "crankcase pressure regulating valve" installed in the suction line, oversize the motor, or select a medium or high temperature model.

In some instances, to provide a motor reserve for conditions of extremely high ambient temperature, poor ventilation, low-voltage situations, etc., with belt-driven units, the motor hp is increased without changing the compressor speed. The same thing can be accomplished with hermetic units by selecting a high temperature model for a medium temperature application. In some instances, a compressor designed for a lower specific volume refrigerant can be substituted to provide motor reserve—for example, a compressor designed for R-22 in a R-12 system.

In the selection of a condensing unit or compressor, the manufacturer's recommendations relative to motor cooling should be followed. In the case of air-cooled equipment, adequate volume and velocity of air over the motor should be provided. In the case of water-cooled equipment, an adequate quantity of water should be circulated to insure proper dissipation of the motor heat. In the case of suction cooled hermetic compressors, some are limited too high, or high and medium temperature applications only. And in some cases, the suction cooling must be supplemented by forced convection air cooling.

Unless one is sure of the manufacturer's recommendations, it is wise to inquire rather than gamble on misapplication.

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SERVICING GENERAL

When servicing HVAC equipment, you should avoid using carbon tetrachloride for any purpose, and refrain from breathing the fumes of a halide torch. Refrigerant oil which has been discolored should not be allowed to touch your skin, and you should never use oxygen to blow out refrigerant lines (contact of oxygen with oil can cause a major explosion). Always install a discharge pressure gauge before pumping down a system. Charge refrigerant into the low side only as a vapor. To avoid damage to property and equipment, be aware of fire hazards—never use a torch where there are combustible vapors in the air or volatile fluids exposed.

CHECKING COMPRESSOR PERFORMANCE

Most refrigeration compressors are precision-built machines designed to pump vapor only, and are designed for high efficiency and long life when properly applied and maintained. In some cases, however, due to defective material or workmanship, normal wear, misapplication, air and moisture, dirt, stress of some extreme condition, or malfunction of some other part of the system, the compressor is found to be operating improperly. There are several checks which can be made on a compressor or condensing unit to determine whether or not it is doing what it is supposed to do, and, if not, where the cause of trouble might be. Some of the checks which can be made on the job are as follows:

CHECKING CAPACITY

With some systems it is a simple matter to check and determine whether or not a condensing unit is developing its rated capacity. For example, a water cooler or water chiller system can be easily checked by measuring the water flow and temperature drop of the water in passing through the evaporator. The capacity can be calculated by multiplying the gallons of water circulated per hour by 8.33 (to convert to lbs per hour), and then multiplying this by the temperature drop (°F). The result is Btu/hr condensing unit capacity. To get the gallons per hour flow, you can time the flow of one gallon (in seconds), and divide the time into 3,600 (seconds in one hour). This will determine the total gallons per hour. If there is a fluctuation in flow, several readings should be taken, and the average determined. Another method would be to discharge the flow into a container for a given time—5 minutes, for example. This amount of water can then be weighed, and it multiplied by 12 (60 minutes divided by 5 minutes) to determine lbs per hour directly.

This same method can be used for checking the capacity of a system cooling some fluid other than water. Instead of the factor 8.33, however, you would use the actual weight of one gallon of the fluid—or 8.33 multiplied by the specific gravity of the fluid. The result would also have to be multiplied by the specific heat applying to the particular fluid being cooled.

Ice makers can be checked for approximate capacity by weighing the total lbs of ice produced in one hour, and multiplying this by the heat removed from the water supplied. The latter factor is 1 Btu/°F differential between water temperature and 32°F, plus 144 Btu/lb (heat of fusion), plus 1/2 Btu/°F differential between 32°F and the actual temperature of the ice harvested.

In comfort air conditioning, or any air-cooling application reasonably accurate capacity figures can be obtained if the required instruments are available. This involves measuring accurately the wet bulb temperature of the air entering and leaving the air handling unit, or evaporator. Knowing the wet bulb temperature drop of the air in passing over the evaporator, one can calculate from tables or a psychrometric chart the Btu removed per lb of air circulated. From readings of wet bulb and dry bulb temperatures of the air leaving the coil, the specific volume (cubic feet per lb) of the air leaving the coil can be calculated from the psychrometric chart also. Then with a "Velometer," or some other device for reading velocity, the average velocity across the face area of the coil or through a discharge duct can be determined. From these data, the condensing unit capacity can be calculated as follows:

$$\frac{\text{Cu. ft. per min.}}{\text{Air Circulated}} = \text{Average Velocity (ft./min.)} \times \text{Area(sq. ft.)}$$

$$\frac{\text{Lbs. per hr. Air}}{\text{Circulated}} = \text{Cu. ft. per min.} \times 60 \div \text{Specific Volume}$$

$$\text{Btu/hr. Capacity} = \text{Lbs./hr. Air Circulated} \times \text{Reduction in Heat Content (Btu/lb.)}$$

As stated previously, the specific volume (cu ft per lb) and heat content of air entering and leaving the coil can be obtained from a psychrometric chart, if you know the wet and dry bulb temperatures of the air entering and leaving.

The foregoing are just a few examples of how the actual capacity delivered can sometimes be checked. Another way of checking capacity, which is only approximate, is by checking the heat rejected by the condenser where a water-cooled condenser is incorporated. This involves determining the Heat Rejection Factor from the chart (see Figure below), knowing the evaporating and condensing temperatures, and applying it to the formula along with the inlet and outlet water temperatures and time for one gallon flow.

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To determine capacity of water cooled condensing unit

Capacity (BTU/HR) = $(T_2 - T_1) \times 30{,}000/S \times F$

T_2 = Temp of water leaving condensor (°F)

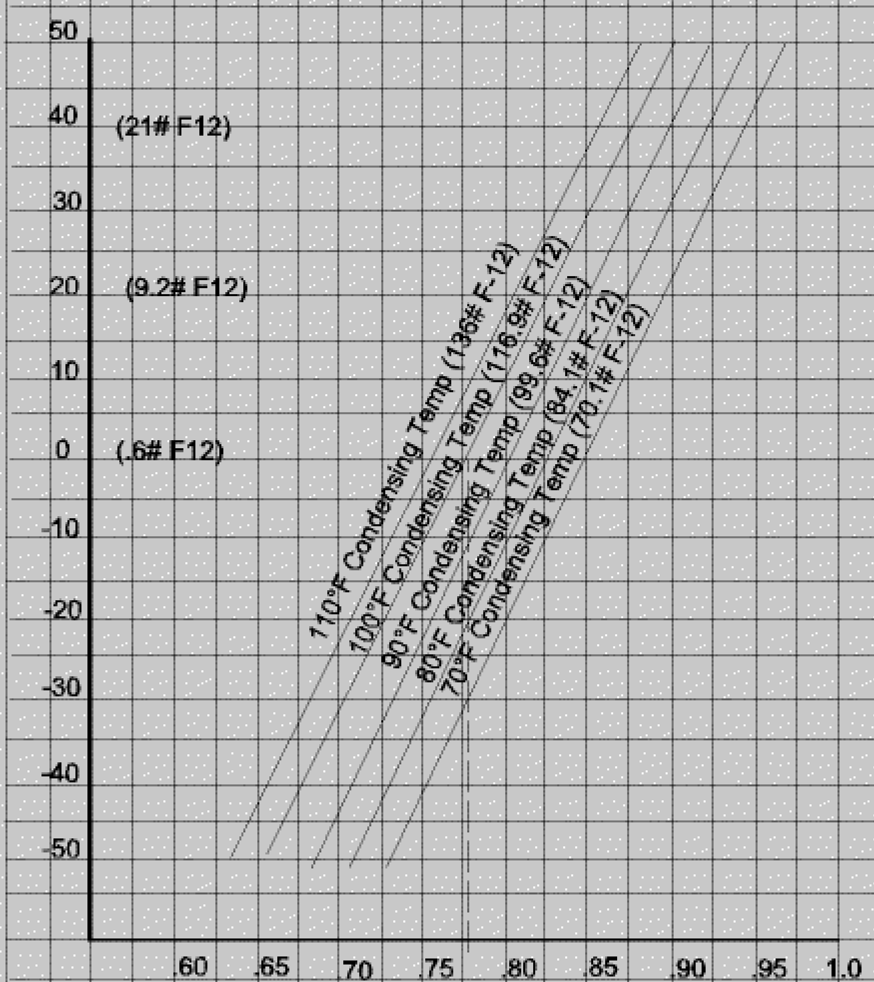
T_1 = Temp of water leaving condensor (°F)

S = Time of one gallon of water to flow through the condenser. (Seconds)

F = Heat rejection for given evaporating and condensing temp.

Example: Water enters condenser at 75 °F, leaves condenser at 90 °F.
Flow is one gallon in 20 seconds. Suction pressure is 4.5 # GA.
(-10°F). Head pressure is 100 # GA.

Approximate capacity = (90-75) x 30,000/20 x .78 = 17,560 BTU/HR.



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CHECKING EFFICIENCY

If it is known or suspected that a compressor is not pumping efficiently, some checks can be made to determine the cause of trouble.

There are a number of things which can cause inefficiency. One of the first points to check is the condition of the refrigerant entering the compressor. If any amount of liquid refrigerant is entering the compressor, the efficiency and resulting capacity will be very seriously affected. To obtain rated capacity of a compressor, the suction gas must be superheated to the temperature specified for the rating (usually 65°F for water-cooled units and 80°F for air-cooled). Besides the physical damage that can result from liquid slugging, this is a common cause of shortage in capacity. To determine actual degree of superheat accurately, measure the temperature of the suction line approximately 2 feet from the suction shut-off valve, and compare this reading with the saturated temperature corresponding to the actual suction pressure at the compressor. If less than 20°F superheat is indicated, steps should be taken to increase the superheat—adjustment of expansion valve, adjustment of refrigerant charge (if a capillary tube system), addition of a heat exchanger, etc.

Another cause of inefficiency is that of leaking compressor valves. The discharge valves can be easily checked by installing a suction pressure gauge, and simultaneously closing the compressor suction shut-off valve as the compressor is stopped. A leaking discharge valve will cause the crankcase pressure to climb rapidly. A rise of over 3 psig per minute indicates excessive discharge valve leakage and warrants the repair or replacement of the valve plate. However, one should make sure the leakage is not a leaking cylinder head gasket or oil separator float valve, as the symptoms are the same as for a leaking discharge valve. If an oil separator is incorporated, the possibility of its leaking can be ascertained by closing off or disconnecting the oil return line. A blown or leaking cylinder head gasket can usually be found by close inspection of the cylinder head and gasket after removal of the head. Observe particularly the web and gasket section separating the high from the low pressure chambers of the cylinder head.

Leaking suction valves seriously affect compressor efficiency (and capacity), especially of lower temperature applications. A test of suction valve operation can be made by gradually closing the suction shut-off valve and blocking the low pressure control, or putting a jumper across the control terminals. With suction and discharge pressure gauges installed, one can operate the compressor to determine the degree of vacuum that can be attained.

WARNING: **Hermetic compressors should not be operated over an 18" Hg vacuum to avoid possible damage to the motor winding. Also, some compressors tend to slug oil excessively if the suction shut-off valve is closed too quickly. One should be prepared to stop the compressor or open the suction shut-off valve if the compressor begins to knock severely. The degree of vacuum that should be attained varies with the make and model of compressor, with discharge pressure, and with atmospheric pressure.**

As an average, a compressor should pull a 20" to 22" vacuum against a 100 psig discharge pressure. When pumping against atmospheric pressure (with discharge shut-off valve closed and gauge port open to atmosphere), it should pull a 24" to 27" vacuum. Against discharge pressures of over 100 psig, the degree of vacuum that can be pulled will become less, and may be no more than 15" at very high discharge pressures. As stated above, the degree of vacuum varies with make and model. In general, compressors with a high ratio of stroke to bore should pull slightly greater vacuum. Compressors designed for low temperature application will usually pull a greater vacuum than those designed for air conditioning and such high temperature applications. In the latter case, compressors are often designed with high clearance volume, and, even when in perfect condition, will not pull over a 17" or 18" vacuum against 100 psig discharge pressure. It is important to keep this in mind when checking compressors in high suction applications.

The other variable, atmospheric (barometric) pressure, affects the suction gauge reading, since the suction pressure gauge reads the difference between the actual suction or crankcase pressure and atmospheric. The lower the barometer reads, therefore, the less vacuum the suction gauge will indicate. The foregoing figures for vacuum, inches of mercury (Hg) below atmosphere, are based on standard sea level atmospheric pressure, which is 29.92" Hg. If the barometer should be reading 28.92", the vacuum indicated on the gauge would be 1" less than when the barometer reads 29.92". If one does not know the barometer reading, as an approximation one can subtract 1" Hg for each 1,000 ft. elevation. For example, if a compressor will pull 22" vacuum at sea level, with suction shut-off valve closed, and operating against 100 psig discharge pressure, it would pull an indicated vacuum on the same gauge of only 20", approximately, at an elevation of 2,000 feet. This is an important factor which must be taken into consideration-especially in such high elevation cities as Denver, Salt Lake City, etc.

A compressor having two or more cylinders can sometimes pull a good vacuum using just one cylinder. With a belt-driven compressor you can determine if both cylinders (and all suction valves) are operating normally by manually turning over the fly-wheel. If equal resistance is felt on all cylinders, and the compressor does pull a satisfactory vacuum, normal operation is indicated. With hermetic type compressors having two or more cylinders, a good vacuum might be indicated when only one cylinder is operating properly. Therefore, if poor compressor capacity and efficiency is a symptom, the only positive check of a multiple-cylinder hermetic type compressor is to remove the valve plate and carefully inspect the suction valve reeds or discs and the valve seats. A properly seating valve will normally show a uniform ring marking on the suction reeds or discs. A small particle of dirt, scale, or any foreign matter on a valve seat can seriously affect the capacity of a compressor. Obviously, any burrs, pitting, flaws in the metal, or fracture of the valve seat will have the same effect.

If a compressor is proved to be inefficient, will not pull a satisfactory vacuum, and yet the valve plate and reeds are in perfect condition, the trouble is usually that of loose pistons or worn bearings. Loose pistons cause excessive blow-by and lack of compression. This condition can usually be observed by allowable sideways movement of the piston in the cylinder. Loose pistons will cause a more pronounced lack of efficiency with ringless pistons (no piston rings incorporated).

Worn bearing—especially loose connecting rods or wrist pins—prevent the pistons from coming up as far as they should on the compression stroke. This has the effect of increasing the clearance volume, and resulting in excessive re-expansion. It is especially detrimental in low temperature applications and is usually accompanied with a knocking sound.

CHECKING MOTOR LOAD

When a compressor is not performing satisfactorily, in addition to checking of capacity and efficiency as outlined above, checking of the motor load is sometimes revealing. Either an exceptionally high or exceptionally low motor load is an indication of improper operation.

Loose pistons, improper suction valve operation, or excessive clearance volume usually lead to a reduction in motor load. (A reduction in condenser heat rejected, as outlined in the previous graphic, will also result.) In addition, a common cause of poor compressor performance, which may not be indicated by abnormal suction or discharge pressures, is a restricted suction chamber screen. The result is a much lower actual pressure in the cylinders at the end of the suction stroke than the pressure in the suction line as registered on the suction gauge. If such is the cause, an abnormally low motor load will also result.

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Improper discharge valve operation, partially restricted ports in the valve plate (which will not show up on the discharge pressure gauge), and tight pistons will usually be accompanied by high motor load.

It may be difficult or impossible for the service man to recognize exceptionally high or low motor loads without exact compressor or condensing unit performance data. If one has at hand the compressor manufacturer's specifications and performance data, which usually lists the wattage input to the motor for various conditions, one can apply a wattmeter and compare the actual wattage of the compressor motor with that listed for the prevailing conditions of suction and discharge pressures. If one does not have available the manufacturer's performance data, the motor current (amperes) will probably be the only comparison. Most motors and some hermetic type compressors have a nameplate amperage stamped on them. This may be indicated as "full load current", in which case the motor is usually good for at least 25% over this value. Therefore, the fact that a motor is drawing a current greater than that stamped on its nameplate does not necessarily mean that it is operating abnormally. And at low temperature or extra low temperature operation the motor may be operating at only 75% of its full load rating. One can also see, by referring to the motor input curves, that the wattage varies with actual suction and discharge pressures for any given model. The motor current (amperes) will vary much in the same way.

It is impossible to indicate here what the motor wattage or current should be. However, the wattage should be reasonably close to that shown in the manufacturer's published data, corresponding to the prevailing conditions. As for motor current, if the compressor is operating at the lower end of its suction range, the current will usually be at, or less than, its nameplate rating. If operated at the upper end of its rated suction range, it will usually be between its rated nameplate current and 40% above this value. A current in excess of 140% of nameplate rating indicates a misapplication or something wrong with the motor, compressor, or some part of the system. The only time the current might exceed 140% of motor nameplate rating, without something being wrong, would be in the case of a low temperature system during the "pull-down" period. The time it takes to pull-down to safe operating conditions is usually a matter of seconds, if the system is properly engineered .

Hermetic motor-compressors, especially those used in air conditioners, are now being produced with a motor current rating other than "full load". Many of these compressors are being stamped with a current rating which is approximately 80% of protector trip point.

Hermetic compressors, especially welded compressors incorporated in air conditioners, frequently have no "full load" or nameplate amps stamped on the serial plates. They may have only locked rotor amps indicated. In such cases the condensing unit or the complete package should have a nameplate or serial plate on which is stamped the normal operating amps which will be the total amps drawn by the entire unit (includes compressor amps, fan motors, crankcase heaters etc.).

CHECKING NOISY OPERATION

A noisy compressor usually indicates something wrong. It might be due to some abnormal condition outside the compressor or something defective or badly worn within the compressor. Obviously, if it is due to some cause outside the compressor, nothing will be gained by changing the compressor. Therefore, before changing a compressor one should first check for these possible causes:

A. Liquid Slugging. Make sure that only superheated vapor is entering the compressor.

B. Oil Slugging. Possibly oil is being trapped in the evaporator or suction line and coming back intermittently in slugs to the compressor.

C. Loose Flywheel. On belt-driven units the flywheel might be loose on the compressor shaft.

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D. Improperly Adjusted Compressor Mounting. With externally mounted hermetic type compressors the feet of the compressor may be bumping the studs, the hold-down nuts may not be backed off sufficiently, or the springs may be too weak allowing the compressor to bump against the base.

Compressor noises coming from internal sources could be any of the following:

A. Insufficient Lubrication. The oil level may be too low for adequate lubrication of all bearings. If an oil pump is incorporated, it may not be operating properly or have failed entirely. Oil ports may be plugged by foreign matter or oil sludged from moisture and acid in system.

B. Excessive Oil Level. The oil level may be so high as to cause excessive oil pumping or slugging.

C. Tight Piston Or Bearing. A tight piston or bearing can cause another bearing to knock—even though it has proper clearance. Sometimes with a new compressor such a condition will "wear-in" after a few hours of running. If a compressor has been in operation for some time, a tight piston or bearing may be due to copper plating, resulting from moisture in the system.

D. Defective Internal Mounting. In an internally spring-mounted compressor the mountings may be bent, causing the compressor body to bump against the shell, or interference result in some way.

E. Loose Bearings. A loose connecting rod, wrist pin, or main bearing will naturally create excessive noise. Misalignment of main bearings, shaft to crank pins or eccentrics, main bearings to cylinder walls, etc. can also cause noise and rapid wear.

F. Broken Valves. A broken suction or discharge valve may lodge in top of a piston and hit the valve plate at the end of each compression stroke. Chips, scale, or any foreign material lying on a piston head can cause the same result.

G. Loose Rotor Or Eccentric. In hermetic type compressors a rotor loose on the shaft can cause play between the key and key-way, resulting in noisy operation. If the shaft and eccentric are not integral, a loose locking device can be the cause of knocking.

H. Vibrating Discharge Valves. Some compressors, under certain conditions, especially at low suction pressure, have an inherent noise which is due to vibration of the discharge reed or disc on the compression stroke. No damage will result, but if the noise is very objectionable, some modification of the discharge valve may be available from the compressor manufacturer.

I. Gas Pulsation. Under certain conditions noise may be emitted from the evaporator, condenser, or suction line. Seemingly a knock and/or a whistling noise is being transmitted and amplified through the suction line or discharge tube. Actually, this may be due to no mechanical knock, but merely a pulsation created from the intermittent suction and compression strokes, coupled with certain phenomena associated with the size and length of refrigerant lines, number of bends, etc. The remedy is usually the installation of an accumulator. If the noise is coming from the evaporator, install the accumulator in the suction line near the compressor. Another possible remedy that often accomplishes the same result is merely to install a perforated disc between the suction shut-off valve and compressor body. The size and number of holes in this disc or plate must usually be determined by "cut and try". Just enough holes should be drilled or punched to silence the pulsation and yet not cause excessive pressure drop. If the noise is coming from the condenser, the remedy is the installation of a muffler in the discharge tube, near the compressor. The outlet should be at the bottom to avoid oil trapping.

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TESTING FOR LEAKS

Before starting up a new system, the compressor, as well as all other components and refrigeration lines, should he carefully tested for leaks. The time required to do so will be well spent and will pay dividends in avoidance of future trouble—especially where the normal operating suction pressure will be at a vacuum and a leak resulting in air (with consequent moisture) being drawn into the system.

Since the compressor or complete condensing unit in the case of a factory assembled package has been thoroughly dehydrated previous to shipment, it is recommended that the compressor shut-off and receiver outlet valves be left closed until the remainder of the system has been leak tested and evacuated.

The compressor or condensing unit can be leak tested separately, introducing refrigerant vapor through the gauge port of the suction shut-off valve if the unit does not already contain sufficient refrigerant vapor pressure to assure detection of any possible leak.

The first step in leak testing is to build up a refrigerant pressure of 50 to 150 psig.

WARNING: Under no conditions should the low side pressure, or crankcase pressure, be allowed to exceed 175 psig for leak testing. If it becomes necessary to mix nitrogen or carbon dioxide with the refrigerant to build up a satisfactory test pressure (such as in low ambient temperature), the gas cylinder must be equipped, not only with a pressure regulator, but the charging line must incorporate a pressure relief valve set to relieve at 175 psig maximum.

All joints should be carefully tested with a good leak detector. The common halide detector, when in good operating condition and properly used, should pick up any leak of any consequence. However, for leak testing systems having a small and critical charge, one of the modern electronic leak detectors is recommended.

As has been repeated many times, caution should be exercised in the use of any halide torch. The room should be well-ventilated to preclude any fire hazard. Also, the fumes discharged from the torch should not be inhaled, as these gases are extremely injurious when even a trace of refrigerant is being picked up by the sampling tube.

In order to make an accurate leak test, it is necessary, of course, to have the surrounding atmosphere free of refrigerant vapors and solvent vapors or any gases that might react as refrigerant would and create false indications by the detector.

In regard to leak testing compressors, all joints—both welded and gasketed—should be tested. With belt-driven or open type compressors, the shaft seal should be tested with special care. If a leak at the shaft seal is indicated, the compressor should be turned several revolutions and again tested. (The seal of a compressor that has been setting idle for a long time may have lost its oil film, which is necessary for a vapor tight seal.)

If any leaks are found, they should be repaired, and the leak testing procedure repeated.

After the system is found to be entirely free of leaks, it should be completely evacuated with a good vacuum pump. The compressor of the system should never be used to evacuate a system. No refrigeration compressor makes a satisfactory vacuum pump, and, in the case of hermetic compressors, failure of the motor winding can result from operating under a deep vacuum.

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In order to evacuate a system satisfactorily, the vacuum pump used should be capable of evacuating to a positive pressure of not over 500 microns of Hg absolute. When you consider that 1/ 1,000 in. equals approximately 25 microns, you will realize how inadequate any refrigeration compressor is as a substitute for a vacuum pump. Very few refrigeration compressors will pull a vacuum of 28" Hg, even when the barometer reads 30" Hg—or an absolute pressure of 2" Hg. Thus, no compressor capable of evacuating to only 2" should be used when a limit of 2/100" Hg is required (500 microns Hg absolute pressure).

To thoroughly evacuate a system, connect evacuating lines to both the high side and low side of the system. These lines should be 3/8" or larger. Run the vacuum pump until 1500 microns Hg absolute pressure is reached. Then stop the pump and break the vacuum with refrigerant. Repeat this procedure twice, evacuating to 500 microns Hg or less absolute pressure the third time.

The vacuum can then be broken by charging in vapor of the refrigerant to be used in the system. Remove vacuum pump and proceed to charge the system. The recommended procedure for field charging is to connect the charging line between the refrigerant cylinder and gauge port of the suction shut-off valve or process connection, with suction gauge also connected. If factory filled cylinders are not being used when charging refrigerant into a system, charge the refrigerant into the system through a drier. Most service cylinders contain objectionable amounts of moisture. The compressor can then be operated, and the valve on the refrigerant cylinder throttled to control the suction pressure at approximately normal operating pressure.

WARNING: **Never tilt the cylinder to allow liquid refrigerant to enter the charging line. Severe damage to the compressor may result if such is attempted.**