PRINCIPLES OF REFRIGERATION – PART 3

By: Paul Reed

INTRODUCTION

This article will cover the process by which the refrigerant vapor that is boiled off in the evaporator is salvaged and reconverted into a liquid, so that it can be used over again, rather than wasting it to the atmosphere.

A liquid boils and changes to a saturated vapor for either of two reasons:

1. Its temperature is raised.
2. Its pressure is reduced.

In either case, heat must be added to the liquid.

Conversely, a saturated vapor changes to a liquid, for either of two reasons:

1. Its temperature is lowered.
2. Its pressure is raised.

In either case, heat must be removed from the vapor.

The temperature at which a saturated vapor condenses to a liquid is exactly the same temperature as that at which that liquid boils to a saturated vapor provided that the pressure is the same in both cases.

Moreover, the amount of heat required to change the liquid to a vapor (boil) is exactly the same as that which the vapor releases when it changes to a liquid (condenses), provided that the temperatures and pressures are the same in both cases.

Therefore, we can convert the vapor from the evaporator back into a liquid, so that it can be re-used in the evaporator, in either of two ways.

RECONDENSING THE VAPOR AT EVAPORATOR TEMPERATURE

1. Chill the vapor to a temperature equal to that of the evaporator from where the vapor came. This could be done by allowing the vapor from the evaporator to pass through a coil cooled by ice, dry-ice or other means, whose temperature is lesser than that of the evaporator, so that heat will flow from the vapor to the cooling medium. In such an arrangement, the vapor would condense to a liquid at the same pressure as that of the vapor and as that of the boiling liquid in the evaporator.

For example, if the refrigerant is dichlorodifluoromethane (R-12) and it is boiling in the evaporator at 20°F, the pressure of the vapor being boiled off is 21 pounds per square inch gage, abbreviated psig (or 34.7 pounds per square inch absolute, abbreviated psia).

If this vapor were passed through a coil cooled to 20°F by cold air, salt and ice, or other means, the vapor would condense back to a liquid at 20°F and at the same pressure of 21 psig. The cooling air, salt and ice, etc. (called the cooling medium or condensing medium) would have to be somewhat colder than 20°F, say at 15°F or 10°F, in order for the heat to flow from the vapor into the cooling medium, for heat always flows from a higher to a lower temperature.

There would be no point in converting the vapor back to a liquid in this manner, for the ice, dry-ice, etc. could have been used instead of the evaporator in the first place.

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CONDENSING THE VAPOR AT HIGH TEMPERATURES

2. However, we can cause the 20°F vapor at 21 psig to condense back to a liquid at a temperature above 20°F, by raising the pressure of the vapor by compressing it in a compressor to a higher pressure, and then cooling it by means of a condensing medium which is at a higher temperature than that of the evaporator, but is somewhat lower than the condensing temperature of the vapor at the higher pressure.

For example: the compressor compresses the 21 psig vapor to a pressure of 136 psig, at which pressure the vapor then condenses if it is cooled to 110°. To do this, the compressed vapor (now usually called a gas) passes through a coil of some sort, over which flows cool air or cool water, comprising the condensing medium. In order for heat to flow from the 110° high pressure (136 psig) gas in the condenser, to the condensing medium, the latter must be at least 5° cooler, that is, 105°, but in practice, it is usually from 20° to 40° cooler.

If the cooling air available is itself 80°, then, allowing a difference of 30° between the condensing air and the gas in the condenser, the condensing temperature corresponding to a pressure of 136 psig, would be 110°F (80° + 30°).

From this, it can be seen that the pressure to which the vapor must be raised by the compressor, depends upon the temperature to which the condensing medium is able to cool the gas in the condenser; the higher the condensing temperature, the higher the condensing pressure, and vice versa.

By the method of compressing the low pressure vapor at 21 psig from the 20°F evaporator, to a pressure of 136 psig, the high pressure gas can be condensed at 110°F by means of air at 80°F. It is now a liquid at the same pressure as the gas in the condenser, and is again available for use in the evaporator.

In the meantime, surplus liquid is stored in a tank-like container, called a receiver, at the outlet of the condenser, and flows from there through the liquid line to the evaporator, as needed.

REDUCING HIGH PRESSURE LIQUID TO LOW PRESSURE LIQUID

The liquid refrigerant in the liquid line is at a high pressure (condensing pressure of 136 psig). After it gets into the evaporator, its pressure must be down to 91 psig in order that it will boil at 20°; so some means must be provided at the inlet of the evaporator to reduce the 136 psig to 21 psig.

There are several means of reducing the liquid from condensing pressure to evaporator pressure, but they all depend upon a small orifice in a valve known as an expansion valve, float valve or injector, or a length of tubing of small inside diameter, known as a capillary tube or restrictor tube. These restrict the flow of refrigerant and cause a pressure drop from condensing pressure to evaporator pressure.

THE COMPRESSION REFRIGERATION CYCLE

This process is called a Compression Refrigeration Cycle, and consists of boiling a liquid refrigerant to a vapor in an evaporator, during which heat is absorbed by the refrigerant, thus producing cooling (refrigeration); passing this heat-laden low pressure vapor to a compressor where its pressure is raised (and some heat added due to the work done on the vapor in compressing it); then passing this heat-laden, high pressure, compressed gas through a condenser where it is cooled and condensed to a high pressure liquid and in so doing, loses most of its heat to the cooling medium (air or water); and finally passing the high-pressure liquid, through a pressure-reducing device, back to the evaporator where it is again boiled, and in so doing again absorbs heat and performs refrigeration. This cycle is repeated as long as the compressor is in operation.

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In this manner, heat is removed from the air in the refrigerator by means of the 20° evaporator; this heat is carried by the refrigerant to the condenser, where the heat is disposed of to the outside 80° air (or to water, in a water-cooled unit).

A refrigeration system is a means of transferring heat from one place at a low temperature to another place which is at a higher temperature.

The compressor and condenser (condensing unit) are means of salvaging refrigerant vapor from the evaporator and converting it back into usable liquid.

A refrigerating system may be thought of as a vehicle for transferring heat from one place to another. The action is purely physical, not chemical, and no heat is created nor destroyed. Every Btu of heat can be accounted for, as will be shown later.

The real process of producing the low temperature is done in the evaporator; the only function of the compressor and condenser is to salvage the vapor from the evaporator by reconverting it to a liquid so that it can again be used in the evaporator.

To illustrate this, let us take an example using R-12 as a liquid at 110°F and at a pressure of 136 psig, and pass it through a "liquid line" to the valve at the entrance to the evaporator. Thus it arrives at the valve at 110°F and 136 psig.

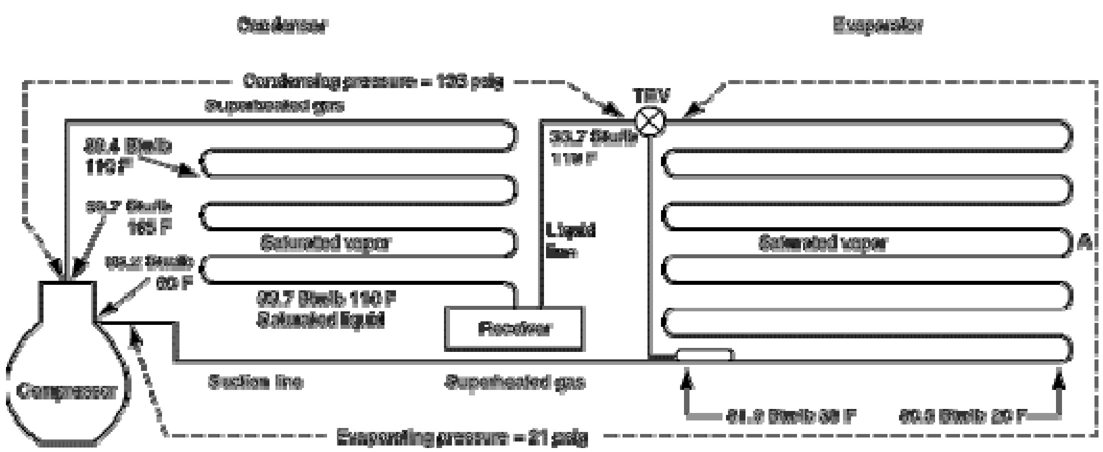

FIGURE 2F43 The Compression Refrigeration Cycle

NET REFRIGERATING EFFECT

The expansion valve is adjusted to reduce the pressure from 136 psig to 21 psig, so the pressure in the evaporator is 21 psig and the temperature 20°F, for at 21 psig, R-12 boils at 20°F.

Heat is required however, so the boiling refrigerant absorbs heat from the evaporator and from the air around the evaporator. We can even determine how much heat each pound of boiling refrigerant absorbs; that is, each pound that is circulated through the evaporator and the rest of the system.

The latent heat of R-12 at 20°F and 21 psig is 67.94 Btu per pound. Therefore each pound must absorb 67.94 Btu for it to boil to a saturated vapor, which is also at 20°F and 21 psig. This is its ability to perform refrigeration.

However the 110° liquid must be cooled down to 20° when it comes into the evaporator; that is, its sensible heat from 110° down to 20° must be removed. The tables show that at 110° the liquid contains

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33.65 Btu per pound and at 20°, 12.55 Btu per pound. So it takes 21.1 Btu per pound (33.65 - 12.55) cooling effect to cool the 110° liquid to a 20° liquid.

The only source of cooling is the latent heat (of vaporization) of 67.94 Btu per pound, so after the 21.1 Btu per pound is subtracted from the 67.94 Btu per pound (67.9 - 21.1), only 46.8 Btu per pound are left with which the refrigerant can cool the evaporator. This 46.8 Btu per pound is called the Net Refrigerating Effect.

Another and somewhat easier way to determine the net refrigerating effect is as follows:

When the 20° vapor leaves the evaporator, it has a total heat content (as shown in the tables in the "Vapor" column) of 80.49 Btu per pound. When the liquid came into the evaporator it had a heat content of 33.65 Btu per pound. The difference is 46.84 Btu per pound (80.49 - 33.65) which is the net refrigerating effect, for it is the amount of heat picked up by the refrigerant in passing through the evaporator, from 110°F liquid to 20°F vapor (saturated).

COOLING CAPACITY OF 12 CFM OF 20° VAPOR CIRCULATED (PUMPED)

Let us now study the process of compressing the gaseous refrigerant in the compressor. First, we will consider this with the vapor entering the compressor as a saturated vapor at 20°F and later as a superheated vapor (which is usually called a "gas").

Again referring to the tables, we find that the density of R-12 saturated vapor as it comes from the evaporator at 20°F, is .892 pounds per cubic foot (lb/ft.^{3}). If we assume that the pump or compressor as we will now call it, can pump 12 cubic feet per minute, it will pump 10.7 pounds of this vapor per minute (12 × .892) - provided that the vapor arrives at the suction part of the compressor saturated at 20°; that is, that the vapor is not superheated between the evaporator and the compressor.

The evaporator can only evaporate refrigerant as fast as the compressor can remove the vapor, so the evaporator capacity is 10.7 × 46.8, or 500 Btu per minute. Since the evaporator is where the refrigeration occurs, 500 Btu per minute is the capacity of this refrigerating system. We can also express this capacity in terms of Btu per hour, which would of course be 60 (minutes per hour) × 500 or 30,000 Btu per hour, or a capacity of 24 × 30,000 =720,000 Btu per day of 24 hours, if the compressor runs continuously for the 24 hours.

A TON OF REFRIGERATION

The smaller sizes of refrigerating systems are usually rated in Btu per hour, which in this example would be 30,000 Btu per hour. The larger sizes of refrigerating systems are usually rated in Tons of refrigeration. A ton of refrigeration is the refrigeration that is given by the melting of a ton of ice in 24 hours.

One pound of ice requires 144 Btu to melt from ice at 32° to water at 32°; that is, the latent heat of fusion is 144 Btu per pound. There are 2,000 pounds per ton of ice, so in melting, a ton of ice requires 288,000 Btu (2,000 × 144). If it melts uniformly over a 24 hour period, it is refrigerating at a rate of 288,000 Btu per 24 hours.

This is called a ton of refrigeration; that is, refrigeration at the rate of 288,000 Btu per 24 hours, or 12,000 Btu per hour (288 000 ÷ 24), or 200 Btu per minute (12,000 ÷ 60). A Standard Ton of mechanical refrigeration is based on a 5° evaporator and 86° condensing.

Therefore, since the refrigerating system in our example has a capacity of 500 Btu per minute, it could also be said to have a capacity of 2.5 (500 ÷ 200) tons of refrigeration.

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VOLUMETRIC EFFICIENCY OF THE COMPRESSOR

But to get back to our example. We assumed that the compressor would pump 12 cubic feet of vapor per minute. If the compressor were 100% efficient, which of course it cannot be, the volume of its cylinders multiplied by its revolutions per minute (RPM) would be 12 cubic feet per minute (cfm) or 20,736 cubic inches per minute (12 × 1,728). If it ran at 600 RPM, then the internal volume of the cylinders would be 34.56 cubic inches (20,736 ÷ 600). If it had four cylinders, the volume of each cylinder would have to be 8.64 cubic inches (34.56 ÷ 4). A cylinder 2-5/16" in diameter with a piston stroke of 2" would have a volume of 8.64 cu. in.

The total volume of a compressor multiplied by its RPM is called its Displacement in cfm. Thus the displacement of this 4 cylinder compressor, with a bore of 2-5/16" and a 2 inch stroke, running at 600 RPM, would be 12 cfm, and if it were 100% efficient it would pump 12 cfm of gas per minute. If this suction gas were saturated vapor at 20°F, the capacity of the compressor would be 500 Btu per minute or 2-1/2 tons of refrigeration—provided, of course, that the liquid to the evaporator was at 110°F and 136 psig.

However, no compressor is 100% efficient; that is, no compressor with a displacement of 12 cfm would actually pump 12 cfm. Compressors vary in efficiency according to their design, accuracy of manufacture, suction pressure and how much the suction gas is superheated, head pressure, the temperature of the compressor (how well it is ventilated) and whether or not there is any oil, air or other non-condensible gases mixed with the refrigerant.

This "pumping" efficiency is called the Volumetric Efficiency, and is expressed in percent. The Volumetric efficiency of a compressor is the percentage of its displacement that the compressor can actually pump, and is found by dividing the cfm of gas that it is actually pumping, by its displacement in cfm. For example, if a compressor having a displacement of 15 cfm were actually pumping 12 cfm, its volumetric efficiency would be 80% (12 ÷ 15).

If, in our example, we assume a volumetric efficiency of 80%, then in order to pump 12 cfm under the conditions given, the displacement of the compressor would have to be 15 cfm. To obtain 15 cfm, a larger compressor (greater bore and stroke) could be used. A 4-cylinder compressor, with a bore of 2-5/16" and a stroke of 2-1/2" running at the same speed of 600 rpm, would have a displacement of 15 cfm. Or the smaller compressor could be used, running at a speed of 750 rpm.

20° SATURATED VAPOR SUPERHEATED TO 60°

In the above example, we assumed that the vapor from the 20°F evaporator was saturated and still at 20°F and without any pressure-drop when it reached the suction port of the compressor. Now let us assume that this 20°F vapor warms up 40°F in the outlet portion of the evaporator and in the suction line, so when it reaches the compressor it is a superheated gas at a temperature of 60° instead of a 20°F saturated vapor. Assuming that there is still no pressure-drop, the suction gas arrives at the compressor at 21 psig, but at 60° instead of 20°F. Nevertheless, since the gas is warmed, it has expanded and is lighter; that is, its density will be less than the .892 pounds per cubic foot that it was when it was 20° and saturated. The superheat tables tell us that the density is now .82 lbs. per cu. ft.

On the other hand there would be no difference in the net refrigerating effect in Btu per pound of refrigerant evaporated, if the temperature of the liquid entering the evaporator is still 110°F and the temperature of the evaporating refrigerant (20°) remains the same. So the net refrigerating effect is the same—namely, 46.8 Btu per pound.

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However, we are now pumping only .82 lbs. per cubic foot of 20° vapor, superheated to 60° by the time it reaches the compressor, instead of .892 lbs. per cu. ft. of vapor that arrived at the compressor saturated, that is, still at 20°F.

If we use the compressor having a displacement of 15 cfm, and it has a volumetric efficiency of 80%, we are actually pumping (or circulating is a better term) only 12 cfm. Therefore the weight of the superheated gas is 12 × .82 or 9.84 pounds per minute, whereas with the saturated vapor, it was 10.7 pounds circulated per minute.

LOSS DUE TO SUCTION SUPERHEATING

The capacity is now 9.84 × 46.8 or 460 Btu per minute. Before, with the saturated vapor, it was 500 Btu per minute, a loss of 40 Btu per minute, or 8% in capacity, just because the vapor warmed up from 20° to 60° before it entered the compressor.

From this, it is seen that there is a loss in compressor capacity as a result of suction superheat. So we might jump to the conclusion that the expansion valve should be adjusted to overfeed the refrigerant in order that the suction gas should be as nearly saturated as possible.

This has been done, particularly on ammonia systems. But ordinarily, it is good practice to have the suction gas somewhat superheated, for several reasons:

1. Some superheat is desirable in order to assure against liquid slugging to the compressor, which also aggravates oil pumping.
2. There is a loss of refrigeration from the cold suction line to the warm room, which may be a greater loss than that due to superheating.
3. A cold suction line, at a temperature below the dew point temperature of the room, causes water to drip from the suction line, which may be objectionable to the user.
4. Insulating the suction line can reduce the loss in (2) and prevent dripping in (3) but it is expensive in first cost and in maintenance.

It is usually good practice to keep the temperature of the suction line about 60° to 65° so that it is above normal room dew point temperatures. This is not always possible, especially on short suction lines, for in all cases the evaporator must be fully fed with refrigerant; that is, the vapor coming from the evaporator must be superheated no more than about 10 or 12 degrees. A properly installed and adjusted TX valve, with its bulb properly located, can maintain a superheat within about 10 to 12 degrees, unless there are other bad conditions over which the TX valve has no control, such as excessive pressure drop in the evaporator.

FACTORS AFFECTING VOLUMETRIC EFFICIENCY

What are the factors that affect the volumetric efficiency of a compressor? Some of the chief factors are:

THE "CLEARANCE VOLUME" OF THE CYLINDER

There are two "strokes" of the compressor piston: the intake or suction stroke, and the exhaust or compression stroke. During the compression stroke (the upward movement of the piston of a vertical compressor), the gas trapped in the cylinder is compressed to an increasingly higher pressure as the piston moves upward. Toward the end of the compression stroke, the pressure of the gas in the cylinder becomes great enough to overcome the high pressure in the condenser, so the discharge valve opens,

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allowing the compressed gas to flow into the condenser. Ideally, this continues until all of the gas is forced out of the cylinder into the condenser. In practice, this is not quite true.

The piston must stop before it hits the head of the cylinder or the discharge valve plate; otherwise the compressor would be damaged and would operate noisily. Also there may be a small amount of oil in the cylinder. Even at the very top of the stroke, there must still be a small clearance between the top of the piston and the valve plate. Perhaps it is only a few thousandths of an inch, but there must be some Piston Head Clearance.

There is also some other space in the cylinder head - the port in the valve plate to the discharge valve, around the suction valve, and around the top of the piston down to the top ring. The high pressure gas in these spaces, known as Clearance Volume, does not get pumped over. When the piston comes to the top of its compression stroke and starts back down on the suction stroke, this trapped, high pressure gas starts to expand and its pressure drops.

However, no suction gas can enter the cylinder until the trapped high pressure gas expands enough that its pressure becomes lower than suction pressure. So the first part of the suction stroke is not effective; for the suction gas cannot enter.

COMPRESSION RATIO

The greater difference there is between suction pressure and condensing pressure the longer it takes for the pressure of the gas trapped in the clearance volume to get down to suction pressure, and the more of the suction stroke that is "wasted".

The compression ratio is the measure of the difference between suction and condensing pressure. It is found by dividing condensing pressure by suction pressure, both expressed in pounds per square inch absolute (psia). Some tables give refrigerant pressures in both pounds per square inch gage (psig) and pounds per square inch absolute (psia), but if gage pressure is given, psia can be found by adding 14.7 to psig.

In our example, the suction pressure is 21 psig or 35.7 psia corresponding to an evaporator temperature of 20°; and the condensing pressure is 136 psig, or 150.7 psia corresponding to a condensing temperature of 110° . Then the ratio of compression is 4.2 (150.7 ÷ 35.7). For the 21 psig suction gas to be able to enter the cylinder, the piston must go downward on the suction stroke enough to furnish a space above the piston at least 4.2 times the clearance volume; for the trapped gas must expand over 4.2 times before its pressure gets down to suction pressure. The higher the ratio of compression, the more important it is to keep the clearance volume of the compressor to the lowest possible amount. The lower the evaporator temperature and suction pressure and the higher the condensing temperature and head pressure, the higher is the ratio of compression and the greater the loss due to clearance volume; consequently the lower will be the volumetric efficiency. Compressors with low clearance volume (well designed and in good condition) are particularly necessary on low temperature systems, such as freezers, even more so than on high temperature systems such as room air conditioners.

DISCHARGE VALVE CLOSING

Ideally, the discharge valve should open as soon as the pressure in the cylinder gets above the condensing pressure. In practice, the discharge valve must have some spring tension, so the pressure in the cylinder must be equal to condenser pressure, plus the spring tension of the discharge valve, plus enough pressure difference to cause the gas in the cylinder to flow into the condenser.

Also important is the cut-off action of the discharge valve. Ideally, it should close the instant that cylinder pressure drops below condensing pressure, otherwise some of the high pressure gas from the condenser

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gets back into the cylinder and adds more gas to the trapped gas in the clearance volume. In practice, there is some delay in the closing of the discharge valve, but the cut-off should be as soon as possible, for the longer it delays closing, the lower becomes the volumetric efficiency.

SUCTION VALVE EFFECT

We have seen that for any given net refrigerating effect (depending upon liquid temperature and the evaporator temperature) the important factor in capacity is the weight of refrigerant circulated; that is, the pounds of refrigerant circulated per revolution, per minute and per hour. It is therefore important that the cylinders of the compressor get a full charge of suction gas, that they be as free of gas trapped from the previous compression stroke as possible, and that the cylinderful of suction gas itself be trapped during the compression stroke and not get pushed back out into the suction line as the pressure in the cylinder increases during the compression stroke.

This again calls for a check valve, which is what the suction valve is. It should stay open all during the suction stroke but close instantly at the end of the suction stroke and just before the compression stroke starts.

Thus, a compressor, for greatest efficiency, should have a suction valve, rather than simply a suction port. If, instead, a suction port is used, it must be large enough to completely fill the cylinder with gas at full suction pressure. Moreover, since a suction port is open during part of the suction stroke, it will also be open during part of the compression stroke; thus, in effect, reducing the length of the compression stroke and thereby reducing the volumetric efficiency of the compressor.

SUPERHEATING IN THE CYLINDERS

The gas in the cylinder can be superheated by the compressor as well as in the suction line. If the cylinders are hot, the suction gas is superheated by them and its density becomes less, thus reducing the amount of the gas in the cylinder. The design should be such as to enable the cylinder housing to remain fairly cool, and there should be normal ventilation around the compressor to enable it to stay reasonably cool.

ADIABATIC COMPRESSION

The term "adiabatic compression" is used to indicate that the suction gas going into the compressor is compressed and discharged into the condenser without either losing heat to or gaining heat from the compressor body. In most engineering problems, it is assumed that compression is adiabatic, but in practice that is not quite true. In a well designed system operating under normal conditions, compression is just about adiabatic however.

The fact that compression is adiabatic does not mean that the gas does not gain heat while it passes through the compressor, for it does. In being compressed, the gas has work done on it, and it takes energy to do work.

TRANSFORMATION OF ENERGY

Energy cannot be created out of nothing, nor can it be destroyed. Energy can be changed from one form to another, however. The chemical energy in coal is changed into heat energy in steam, which is changed into mechanical energy in the steam turbine, which is changed into electrical energy in the electric generator which is changed into mechanical energy again in the electric motor, again into mechanical energy in the compressor, and back to heat again in the compressed gas.

At each of these steps, we lose some of the energy when it is changed from one form to another. The "lost energy" is usually in the form of heat radiated from the boiler, engine, generator, motor or

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compressor, etc., so it is lost as usable energy as far as we are concerned, but it is not "destroyed". As far as nature is concerned, it is still in existence.

If, for example, we put a certain amount of electrical energy into an electric motor, most of it is changed into useful work energy that we measure in horsepower (HP), but some of the input electrical energy is changed into heat that warms the motor and is radiated into the air.

MECHANICAL EFFICIENCY OF THE COMPRESSOR

Now let us return to our example of the 4-cylinder compressor having a displacement of 15 cfm and a volumetric efficiency of 80%; so it can actually pump (circulate) 12 cfm. Please note this is "volumetric efficiency" and not mechanical efficiency. They are not the same thing, for volumetric efficiency relates only to volumes of gas, while mechanical efficiency relates to energy.

In our example, the suction pressure is 21 psig, corresponding to an evaporator temperature of 20°, but is superheated to 60° by the time it reaches the compressor suction service valve. The liquid enters the evaporator at 110°, so the net refrigerating effect is 46.8 Btu per pound.

Under these conditions the density of R-12 is .82 lbs. per cu. ft., and 9.84 pounds are circulated per minute, thus giving a capacity of 460 Btu per minute refrigerating capacity (46.8 × 9.84), which can also be expressed as 2.3 tons of refrigeration (460 ÷ 200).

But how much work had to be done on the gas in order to obtain refrigeration of 460 Btu per minute (or 2.3 tons)?

If we refer to superheat tables or to a Mollier diagram for R-12, we find that the gas entering the compressor at 60°F, superheated from 20°F saturation, has a heat content of 86.2 Btu per pound.

The tables or diagram also show that the gas does not come out of the compressor at 110°F, for it is greatly superheated above the condensing temperature of 110°F. Actually, the gas comes out of the compressor (under the conditions in this example) at 165°F, which is 55° above the condensing temperature of 110F°.

Also the tables or diagram tell us that this 165°F "hot gas" has a heat content of 98.7 Btu per pound. (This is not strictly true in practice, for it assumes "constant entropy", but it is approximately true.)

Since the gas enters the compressor with a heat content of 86.2 Btu per pound and is discharged with a heat content of 98.7 Btu per pound, each pound circulated gains 12.5 Btu (98.7 - 86.2). This 12.5 Btu per pound represents the work that is done by the compressor in compressing the gas, so it is called the Heat of Compression. Since 9.84 pounds are circulated per minute, 123 Btu of energy are expended on the gas per minute (9.84 × 12.5), or 7,380 Btu per hour (123 × 60), which represents the energy required to circulate and compress the gas at the conditions in the example.

This 7,380 Btu per hour is expended by the compressor on the gas, but the compressor itself is not 100% efficient. Let us assume that it is 75% efficient. Then, since its output (work on the gas) is energy at the rate of 7,380 Btu per hour, the work input to the compressor must be energy at the rate of 9,840 Btu per hour (7,380 ÷ 75%). Thus, this compressor has a mechanical efficiency of 75%, although its volumetric efficiency is 80%.

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CONVERSION FACTORS

In an electrically driven compression refrigeration system, energy exists in several forms—heat, electricity, mechanical energy- each of which has its own unit of measurement: the Btu for heat, the watt for electricity, and the horsepower for mechanical energy. There are some other forms of energy and other units of measurement; but these are the main ones in common use by HVACR technicians.

Since energy can exist in different forms and be measured by different units, there are equivalent units of one form of energy for units of another form of energy. That is, we can express one form of energy in units of another form of energy. For example: 1 watt of electricity is equivalent to 3.412 Btu of heat, and 1 Btu of heat is equivalent to .293 watts: 1 horsepower of mechanical energy (power) is equivalent to 746 watts, and 1 kilowatt (1,000 watts) is equivalent to 1.34 horsepower; 1 horsepower is also equivalent to 2,545 Btu per hour (746 × 3.412) .

Therefore, instead of expressing the input to the compressor as 9,840 Btu per hour, we can express it as 3.87 horsepower (9,840 ÷ 2,545). In other words, it requires 3.87 hp to drive this compressor under the conditions given in the example.

THE DRIVE EFFICIENCY

If the compressor is driven directly by the motor, as in a hermetic motor-compressor, the loss between the motor and the compressor is negligible. If, however, the compressor is gear or belt driven, there is a loss in power and energy between the motor and compressor, and this loss is in the form of heat radiated and conducted from the gears, belts and pulleys.

If the belt slips in the grooves of the pulley or does not properly fit the grooves, or the belt tension is not properly adjusted, the loss in the drive may run quite high. An inefficient drive will make itself evident by overheating of the belts and pulleys, for the loss of power is converted into heat.

A well designed V belt drive, with the belts and pulleys clean, free of oil, the belts properly adjusted, and the belts in good condition, may have an efficiency of about 90%, that is, the loss in the belt drive is about 10%.

Therefore, if the compressor in our example is V-belt driven, and the efficiency of the drive is 90%, the motor output must be 4.3 hp (3.87 ÷ 0.9). There is no standard motor between 3 hp and 5 hp, so a 5 hp motor would have to be used. This allows at least .7 hp for times when the suction or condensing pressures may be above those in our example. Also, it allows for some voltage fluctuations, and for other conditions that may temporarily impose more than a 4.3 hp load on the motor.

MECHANICAL EFFICIENCY OF THE MOTOR

Electric motors vary a great deal in efficiency, depending upon the type and design of motor, horsepower, load, temperature, whether or not the voltage and frequency of the electrical supply are those for which the motor was designed as shown on the motor name plate, and, of course, its condition.

A small fan motor may have an efficiency as low as 55%, while a large three-phase squirrel-cage motor may have an efficiency as high as 92%. If we use a 3 hp, 3-phase, squirrel-cage motor of good design and in good condition, and if the voltage and frequency are normal, its efficiency should be about 75% or possibly as high as 80%.

The 4.3 hp is equivalent to 3,208 watts (4.3 × 746), so if we assume a mechanical efficiency for the motor of 75%, then the electrical input to the motor must be 4,280 watts (4.28 kW) (3,208 ÷ .75). This is a ratio of 1 kW input to 1 hp output, which is about normal for this size and type of motor.

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By: Paul Reed

HORSEPOWER PER TON

In our example, the refrigerating capacity under the conditions given is 460 Btu per minute, with the 20°F suction gas superheated to 60°F. We can also express this capacity at 27,600 Btu per hour (460 × 60), or as 2.3 tons of refrigeration (460 ÷ 200, or 27,600 ÷ 12,000).

To produce this 2.3 tons of refrigeration, we must expend 4.3 horsepower, so refrigeration is being produced at the rate of 1.87 hp per ton (4.3 ÷ 2.3).

BTU PER WATT

Horsepower per ton is used in connection with large equipment. For small systems, we usually refer to the "Btu per watt". In our case, we produce 27,600 Btu per hour with a wattage (which is based on 1 hour) of 4,280 watts. Therefore we are producing refrigeration at the rate of 6.45 Btu per watt (27,600 ÷ 4,280).

EFFECT OF SUCTION SUPERHEAT

At first, we assumed that we were bringing the 20°F vapor from the evaporator without superheating; that is, still at 20°F when it entered the compressor. We found that, due to its greater density, we could circulate 10.7 pounds of saturated R-12 per minute, instead of 9.84 lbs. per min., with the suction gas superheated to 60°.

We found that this enabled us to develop 500 Btu per minute, instead of 460 Btu per minute; 500 Btu per minute is equivalent to 2.5 tons of refrigeration (500 ÷ 200). Then with the suction gas saturated at 20°F, the hp per ton would have been 1.72 instead of 1.87. (Actually, it would have been somewhat lower than 1.72 for there would have been some increase in compressor efficiency with the 20° saturated suction, over the 60° suction, superheated from 20°.)

Also, with the suction gas saturated at 20° the capacity would have been 500 Btu per minute or 30,000 Btu per hour (60 × 500). Therefore, instead of 6.45 Btu per watt, we would have produced refrigeration at the rate of 7 Btu per watts (30,000 ÷ 4,280), a gain of 8.5% on a Btu per watt basis.

EFFECT OF LIQUID SUB-COOLING

In our example, we allowed the 20° saturated vapor to warm to 60° before it reached the compressor. At 20° saturation, the heat contained in the vapor was 80.5 Btu per pound. Superheated to 60°, its heat content was 86.2 Btu per pound, so it gained 5.7 Btu per pound.

Presumably, most of this 5.7 Btu per pound was gained by the suction line from the room air. If we could have used this cooling effect to cool the 110° liquid to the expansion valve, there would have been a gain in net refrigerating effect and consequently in refrigerating capacity.

By using a liquid to suction line heat exchanger (called a "heat interchanger") we could have cooled the liquid by the cool suction gas. We could not have taken advantage of the entire 5.7 Btu per pound, but let us assume that we could have taken advantage of 4 Btu per pound.

The tables tell us that the heat content of 110°F liquid is 33.7 Btu per pound, so a cooling of 4 Btu per pound would reduce the heat content of the liquid to 29.7 Btu per pound. The tables also tell us that 29.7 Btu per pound is the heat content of 94°F liquid, so the liquid could be cooled from 110°F to 94° by using the cooling effect in the suction line, which would be lost anyway.

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Then the net refrigerating effect would have been 50.8 Btu per pound (80.5 - 29.7), a gain of 4 Btu, and the refrigeration produced would have been 500 Btu per minute (9.84 × 50.8), thus offsetting the loss of refrigeration due to the superheating of the suction gas from 20° saturation to 60°, and effecting a capacity gain of 40 Btu per minute or a gain of 8.5%.

NOTE:

This is based on the supposition that we could have "saved" all of the refrigeration lost in the suction line. In practice, we probably could not have taken advantage of that much, so the percentage of saving would have been less than 8.5%.

COEFFICIENT OF PERFORMANCE (COP)

Let us again return to our original example, with liquid entering the evaporator at 110°F, boiling in the evaporator at 20°F, and being superheated to 60°F by the time it enters the compressor. No heat interchanger is used, so the cooling effect of the returning vapor is not being taken advantage of. Thus, the system has a capacity of 460 Btu per minute or 27,600 Btu per hour.

In order to produce this refrigeration at the rate of 27,600 Btu per hour, 4,280 watts is used by the electric motor. We can convert the 4,280 watts into Btu by multiplying by 3.412, so the input energy to the electric motor can be expressed as 14,603 Btu per hour instead of 4,280 watts.

Since we put energy into the motor at the rate of 14,603 Btu per hour and produce refrigeration at the rate of 27,600 Btu per hour, we produce refrigeration at a ratio of 1.89 times as much as the energy input (27,600 ÷ 14,603).

This may sound as if we get out 1.89 times as much as we put in, but this would be an efficiency of 189%, and even ideally, efficiency cannot be higher than 100%. In practice, it is never even that high.

We use input energy equivalent to 14,603 Btu per hour, but we do not get 27,600 Btu of energy out. We use 14,603 Btu (heat equivalent of 4,280 watts) to transfer 27,600 Btu from the cold evaporator, through the compressor to the condenser and then to the air (or cooling water) outdoors. It is as if we burn 14,603 gallons of gasoline in the engine of a tank truck, to transport 27,600 gallons of gasoline from one city to another.

The ratio of refrigeration produced per hour, to the input to the motor per hour, both expressed in the same units of energy, is called the Coefficient of Performance, abbreviated COP. In our example, the COP is 1.89 (27,600 ÷ 14,603).

TEMPERATURES AND HEAT CONTENT OF THE REFRIGERANT IN THE CONDENSER

When the gas leaves the compressor it is hot, 165°, which is known as its Discharge Temperature, and it has a heat content of 98.7 Btu per pound. We assume that the air in the room is 80° and that the condenser is 30° higher than room temperature, that is, 110°, which is the temperature at which the gas condenses into a liquid, and is therefore the condensing temperature. But 165° is the temperature at which the superheated gas is discharged from the compressor. so it is the Discharge Temperature. Thus the Discharge Temperature of 165° is not the same as the Condensing Temperature of 110°, although the pressure is the same, 136 psig.

Before the hot discharge gas can condense, it must first cool from the discharge temperature of 165°F down to the condensing temperature of 110°F. This means that from 165°F to 110°F, the gas must lose sensible heat; then at 110°F it must lose latent heat (latent heat of condensation, which is the same as the latent heat of vaporization) to enable it to change from a vapor at 110°F to a liquid at 110°F.

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The tables show that saturated R-12 vapor at 110°F has a heat content of 89.4 Btu per pound. At 165°F, its heat content is 98.7 Btu, so it must lose 9.3 Btu per pound (98.7 - 89.4) of sensible heat (superheat) to allow it to cool to 110°F, and before it can start to condense. Since we are circulating 9.84 lbs. per minute, the hot-gas must lose 91.5 Btu per minute (9.84 × 9.3) of sensible heat (superheat) in the first part of the condenser, in cooling from a 165°F gas to a 110°F saturated vapor.

Although the discharge gas is at 165°F when it enters the condenser, its pressure is 136 psig, for the condensing temperature 110°F is what determines the pressure of 136 psig, which is the saturation pressure corresponding to 110°.

The saturation tables also show that the latent heat of vaporization of saturated R-12 at 110° is 55.7 Btu per pound. This is the amount of heat that the 110°F saturated vapor must lose before it can condense to a 110°F liquid. Since 9.84 pounds of refrigerant are being circulated per minute, the total amount of latent heat lost by the gas in the condenser is 548.1 Btu per minute (55.7 x 9.84).

Therefore, the total amount of heat that 165°F hot discharge gas must lose, first to cool to a saturated vapor at 110°F, and then to condense into a liquid at 110°F, is 639.6 Btu per minute (91.5 + 548.1) or 38,376 Btu per hour (639.6 × 60).

CONDENSER HEAT REJECTION FACTOR

This 639.6 Btu per minute or 38,376 Btu per hour is called the Heat Rejection of the condenser, for it is the total amount of heat that the condenser must reject to the air (or water) per hour.

The refrigerating system of our example is producing refrigeration at a rate of 27,600 Btu per hour, representing heat removed from the evaporator. This latent heat is disposed of (rejected) by the condenser to the air or water, but the total amount of heat rejected by the condenser is 38,376 Btu per hour.

Of this total, 27,600 Btu represents the useful refrigeration performed by the system, which is 72% of the total heat rejection (27,600 ÷ 38,376). This 72% is known as the Heat Rejection Factor of the condenser.

The difference between 38,376 Btu per hour rejected by the condenser, and 27,600 Btu per hour, the refrigeration capacity of the system, is 10,776 Btu per hour, which represents the work done on the gas by the compressor of 7,380 Btu per hour, plus the superheat of the gas from the outlet of the evaporator at 20°F to the inlet of the compressor at 60°F, amounting to 3,365.3 Btu per hour.

(More accurately the refrigeration produced in the evaporator, is 27,630.7 Btu per hour, or 46.8 Btu per pound net refrigerating effect 9.84 pounds per minute × 60 minutes = 27,630.7, but we have used 460 Btu per minute and 27,600 Btu per hour for simplicity.)

If we add these three values of heat per hour, we find that they exactly equal the amount of heat in Btu per hour rejected by the condenser.

Latent heat to evaporator	=	27,630.7
Superheat in suction line	=	3,365.3
Heat of compression	=	7,380.0
TOTAL		38,376.0

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While in the condenser:

Sensible heat, 165° to 110°	=	5,490.7
Latent heat at 110°	=	32,885.3
TOTAL		38,376.0

Thus, we find that, as predicted earlier, every Btu can be accounted for theoretically. This would even be true in practice if we could use enough and sufficiently accurate instruments.

The heat of compression, plus losses, accounts for the power used to drive the motor of 4,280 watts or 14,603 Btu per hour.

Heat of compression	7,380
Loss in compressor 75% Eff.	2,460
Loss in drive 90% Eff.	1,107
Loss in motor 75% Eff.	3,656
TOTAL	14,603

Thus the combined mechanical efficiency of the motor, drive and compressor, is 50.5% (7,380 ÷ 14,603) (or 75% × 90% × 75%).

It may be of some further interest to trace the heat content of the refrigerant in Btu per pound, throughout the cycle.

Starting with liquid entering the expansion valve at 136 psig:

Liquid at 110°F	33.7	Btu/lb.
Saturated vapor in evaporator at 21 psig	80.5	Btu/lb.
Difference is Net Refrigerating Effect of 46.8 Btu/lb.		
Superheated gas at 60°F entering compressor, still at 21 psig	86.2	Btu/lb.
Superheated gas at 165° leaving the compressor at 136 psig	98.7	Btu/lb.
Difference is Heat of Compression of 12.5 Btu/lb.		
Saturated vapor at 110° at 136 psig in condenser	89.4	Btu/lb.
Difference is Sensible Heat of 9.3 Btu/lb.		
Back to 110°F liquid at 136 psig in condenser	33.7	Btu/lb.
Difference is Latent Heat of Condensation of 55.7 Btu/lb.		