PRESSURE-ENTHALPY CHARTS AND THEIR USE

By: Dr. Ralph C. Downing
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INTRODUCTION

The refrigerant in a refrigeration system, regardless of type, is present in two different states. It is present as liquid and as vapor (or gas). During the refrigeration cycle, it changes from one state to the other. You need to be familiar with the properties of both liquid and vapor in order to understand the refrigeration cycle. The five properties of vapor are:

- T, for temperature
- H, for enthalpy (or heat content)
- V, for volume
- S, for entropy
- P, for pressure

These properties are the key to understanding and using refrigerant pressure-enthalpy diagrams. Such diagrams chart the properties of gas.

Temperature is a measure of how hot or cold an object is. It does not tell how much heat an object will hold, nor does it tell how much heat it takes to change an object's temperature. There are two temperature scales that anyone in refrigeration work should understand. They are the Fahrenheit scale and the Celsius (formerly Centigrade) scale. The Fahrenheit scale is the most commonly used in the United States. The Celsius scale is used in most scientific work in the United States and almost universally in other countries. There is a movement to make the Celsius scale the standard of temperature measurement in the United States.

Volume is a measure of the space occupied by refrigerant vapor. In refrigeration work, cubic feet per pound (ft^3/lb) is the standard unit.

In refrigeration service work, gauge pressure (psig) is generally used. On a pressure-enthalpy diagram, however, the pressure is shown as absolute pressure (psia.) The difference between the two is about 14.7 pounds per square inch or about 30 inches of mercury (29.92 inches of mercury, to be more precise). This is the difference between atmospheric pressure at sea level and no pressure (14.7 psi less than atmospheric pressure, also referred to as a “perfect vacuum”).

There are three different kinds of pressure to consider in a study of refrigerants and how temperature affects them. Adding heat to gas when there is no liquid present will cause it to be superheated. The pressure of superheated gas increases very slowly when heated. If the gas is a saturated vapor (that is, if liquid is present), the pressure increases much more rapidly. This pressure is called vapor pressure (or saturated pressure). Remember that when liquid is present, any pressure value will be vapor pressure. There can be only one value for the pressure at a given temperature for a single refrigerant. (Refrigerant mixtures such as MP39 have a range of

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pressures for each temperature because of the differences in vapor pressures of the individual refrigerant components.)

If heat is added to a container full of liquid refrigerant, it will develop hydrostatic pressure. This pressure increases rapidly in response to a temperature increase. A small increase in temperature in a container full of liquid refrigerant will cause it to develop dangerously high pressure. Never completely fill a cylinder with liquid refrigerant. In dealing with liquid and gas pressures in the refrigeration cycle process, never do anything that might result in hydrostatic pressure.

Heat content is also known as enthalpy. It is a measure of how much heat a gas or liquid can hold and how much heat is needed to change the temperature. A gas can have the same heat content at different temperatures only if its other properties, such as pressure and volume, are different. When enthalpy is constant, the condition of the gas is called adiabatic. Entropy is harder to define. It is the ratio of heat content of a gas to its absolute temperature. It remains the same when a gas is compressed, if no heat is added or removed. When entropy is constant, the condition of the gas is called isentropic.

SAMPLE DIAGRAMS

The most common type of pressure-enthalpy diagram is shown in Figures 1A through 1H. They show all five properties for a specific refrigerant in present-day use. Pressure is listed along the right and left borders in psia. Horizontal lines are constant pressure lines. Enthalpy is listed along the bottom and top of the diagram. Vertical lines are constant enthalpy lines. Temperature, entropy, and volume values are shown in curves.

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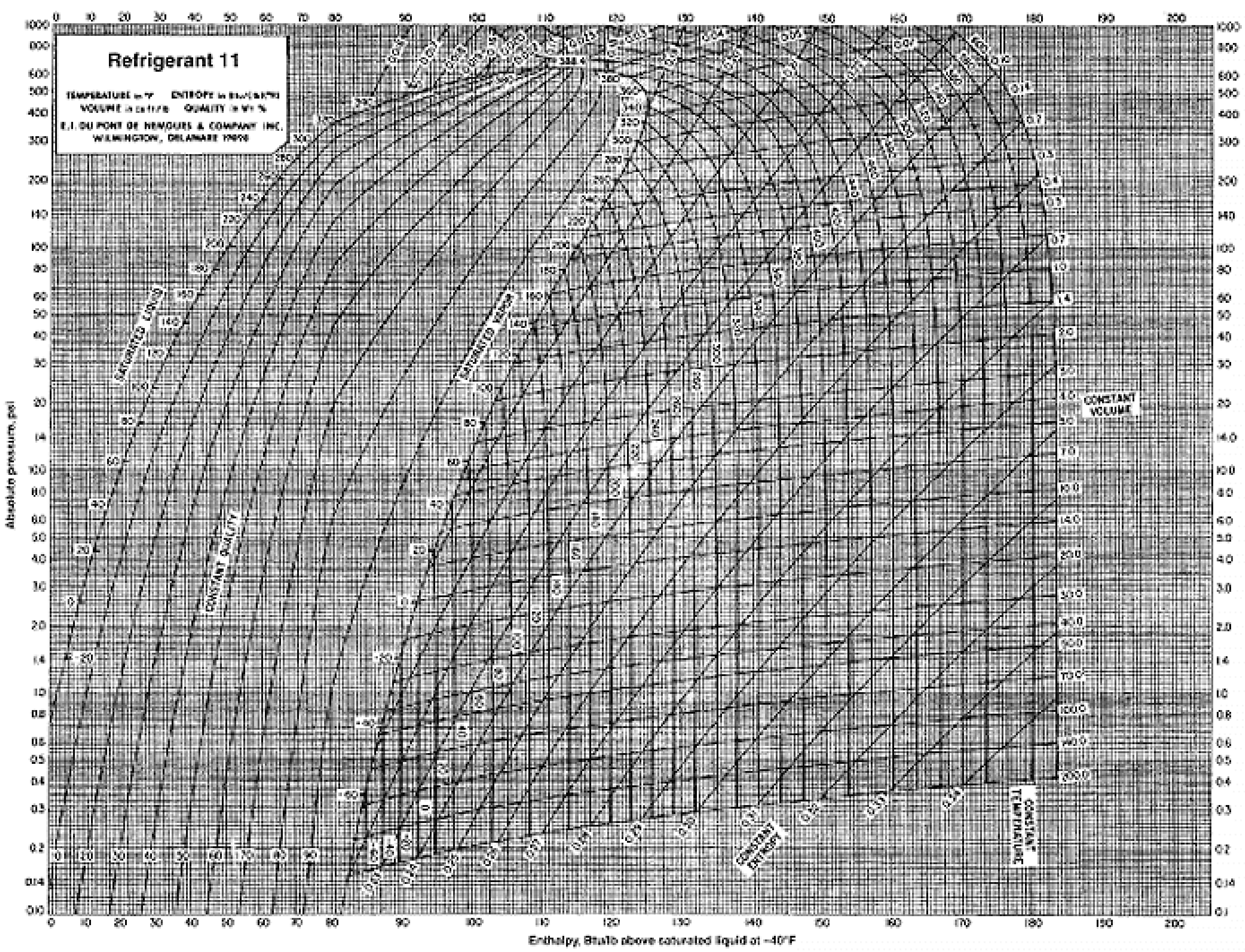

FIGURE 1A. Refrigerant 11.

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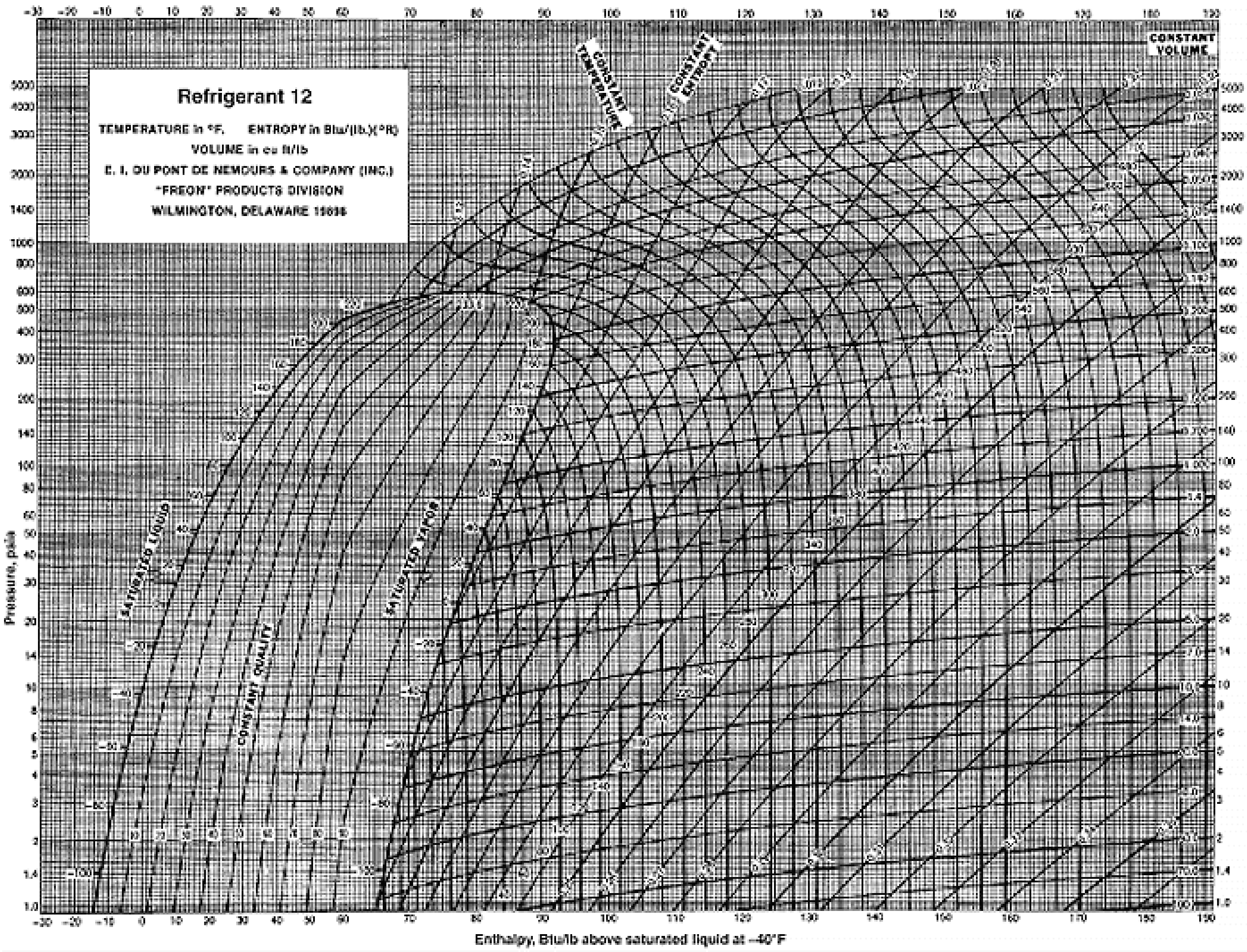

FIGURE 1B. Refrigerant 12.

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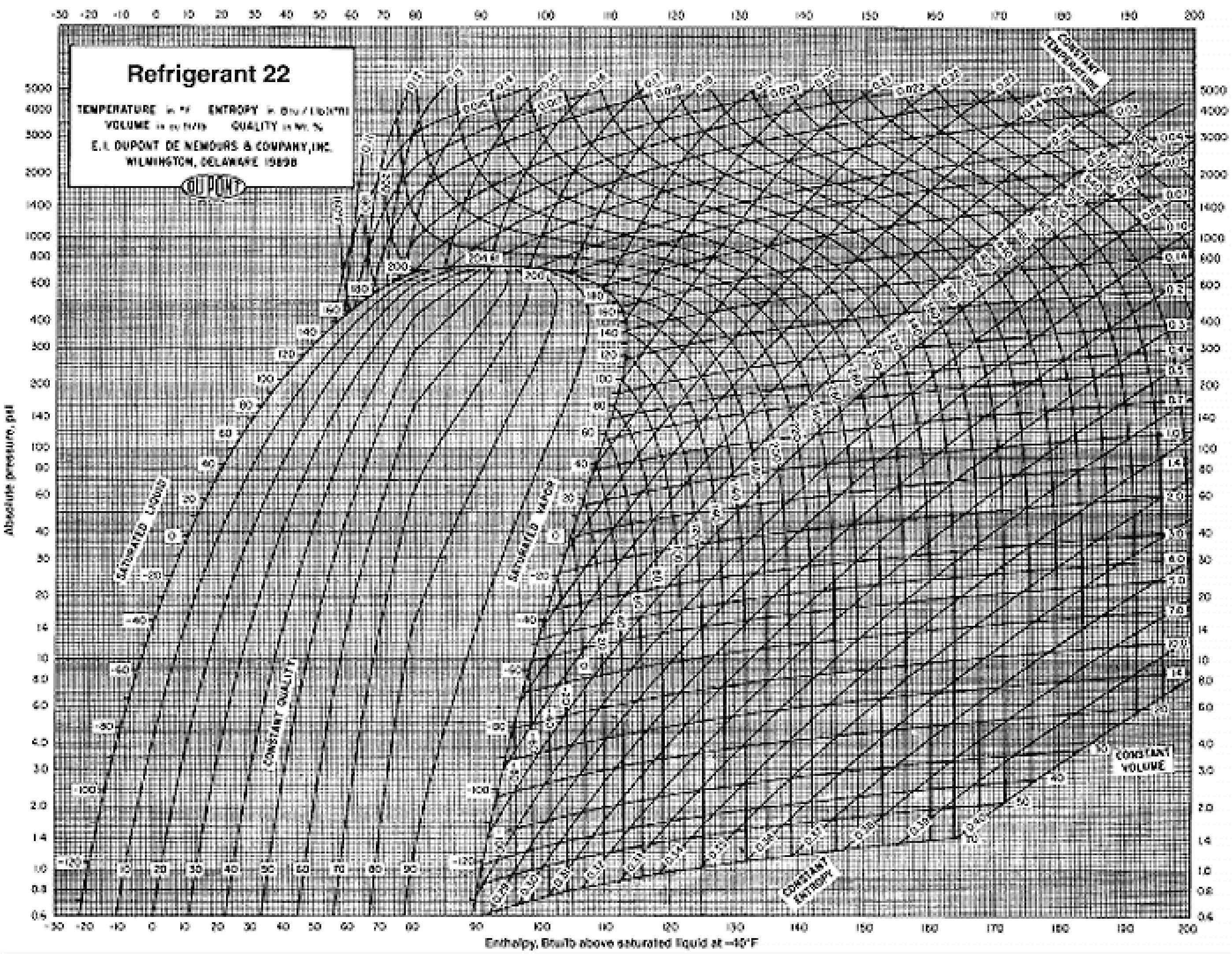

FIGURE 1C. Refrigerant 22.

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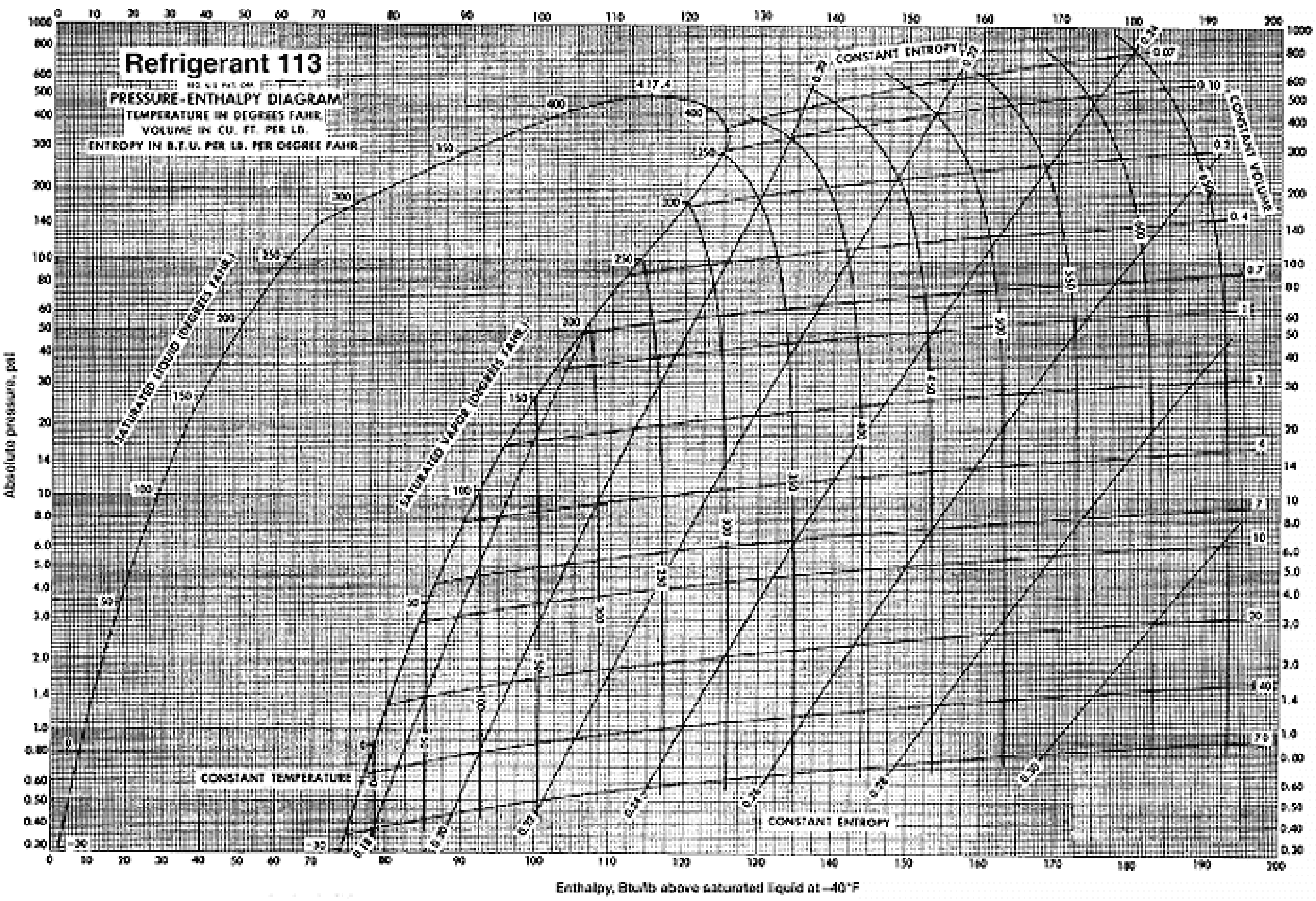

FIGURE 1D. Refrigerant 113.

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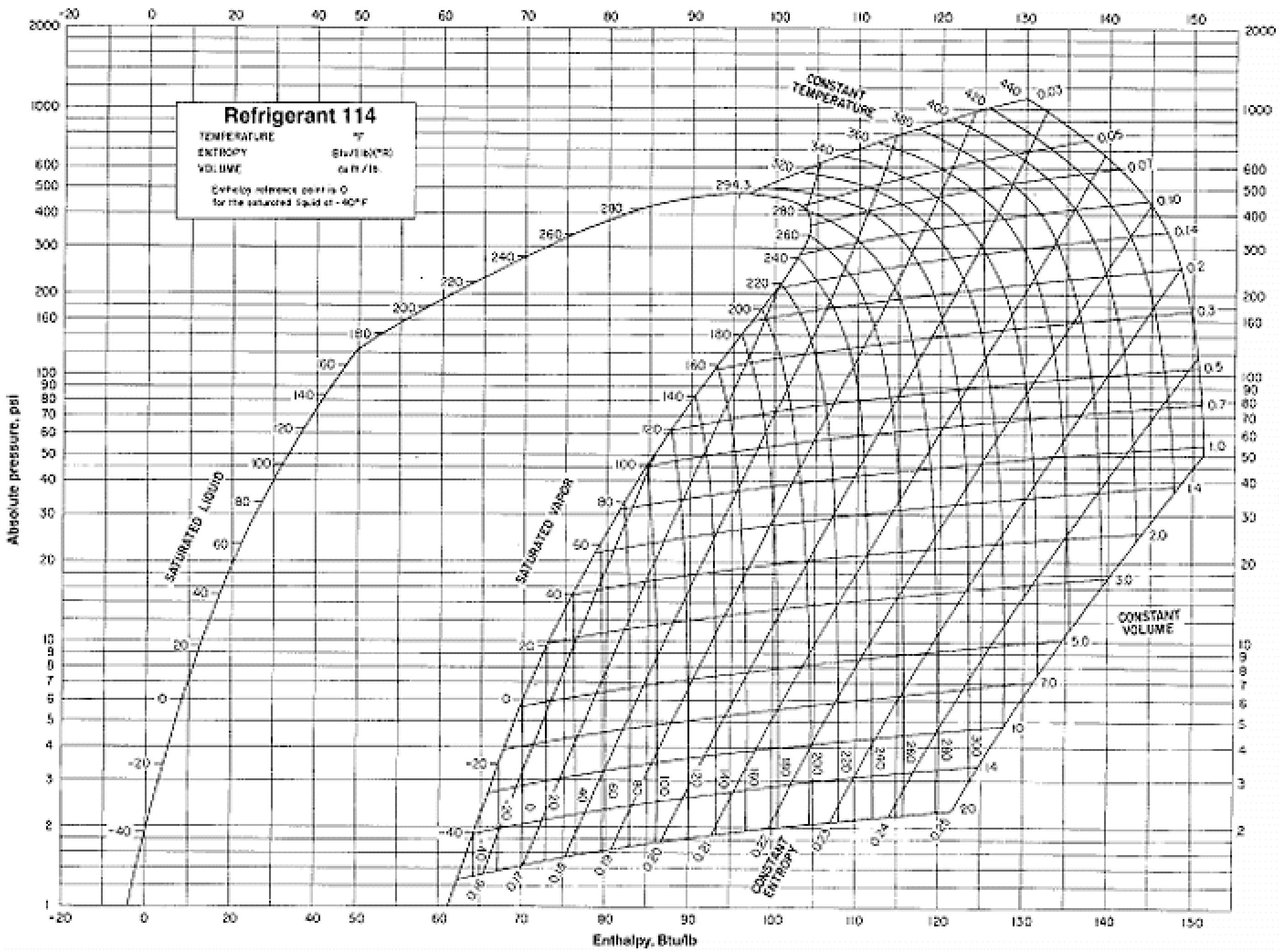

FIGURE 1E. Refrigerant 114.

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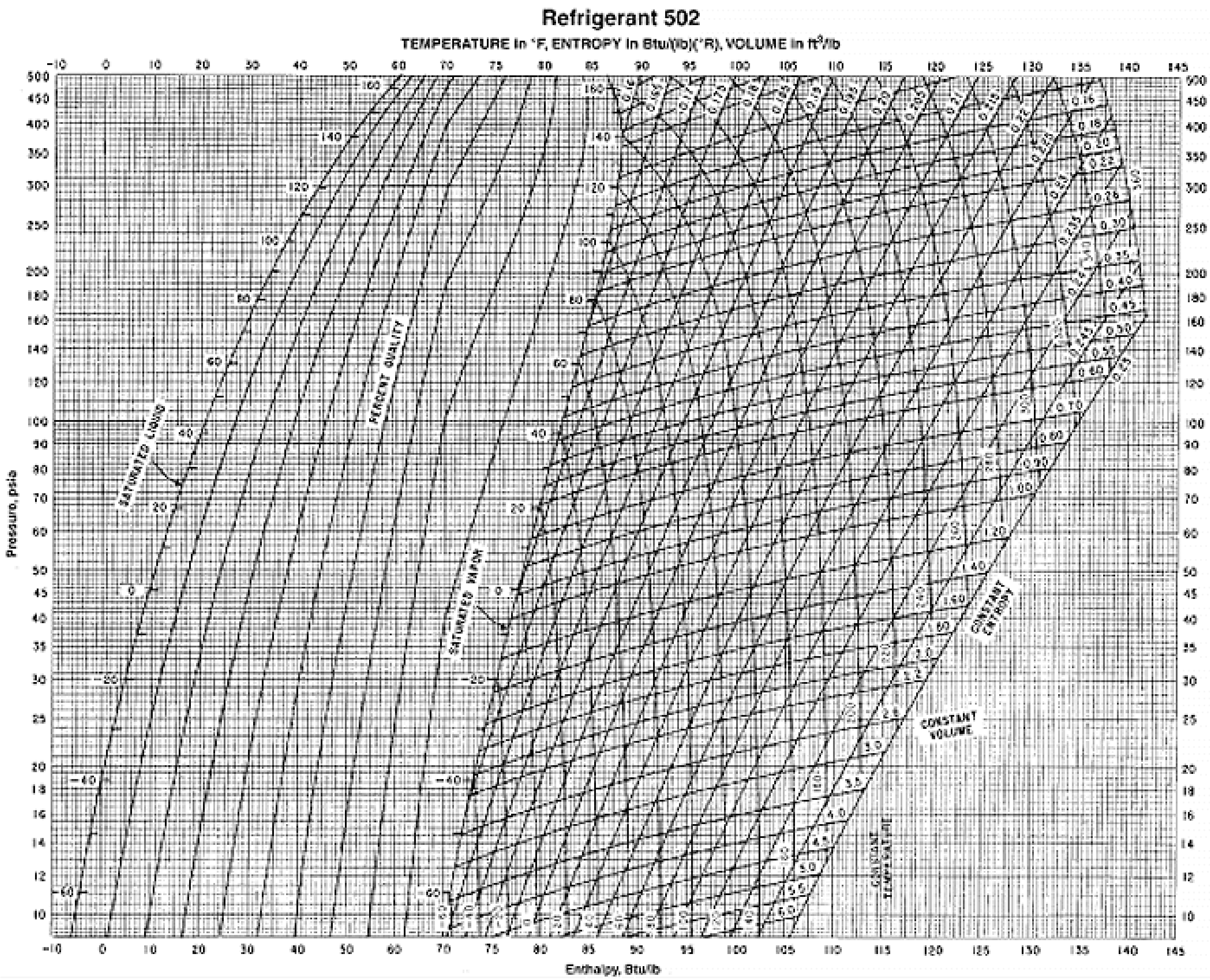

FIGURE 1F. Refrigerant 502.

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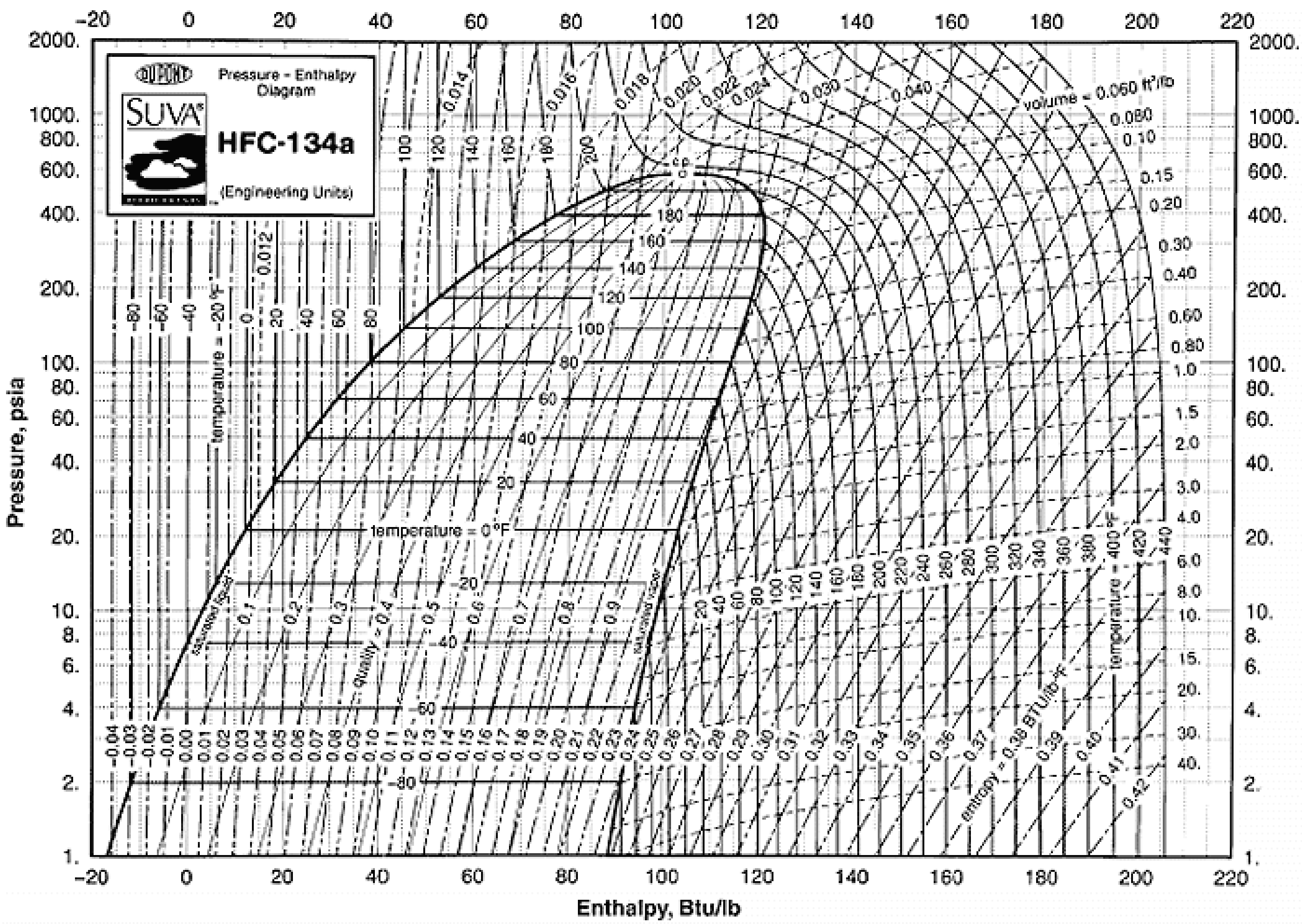

FIGURE 1G. Refrigerant HFC-134a.

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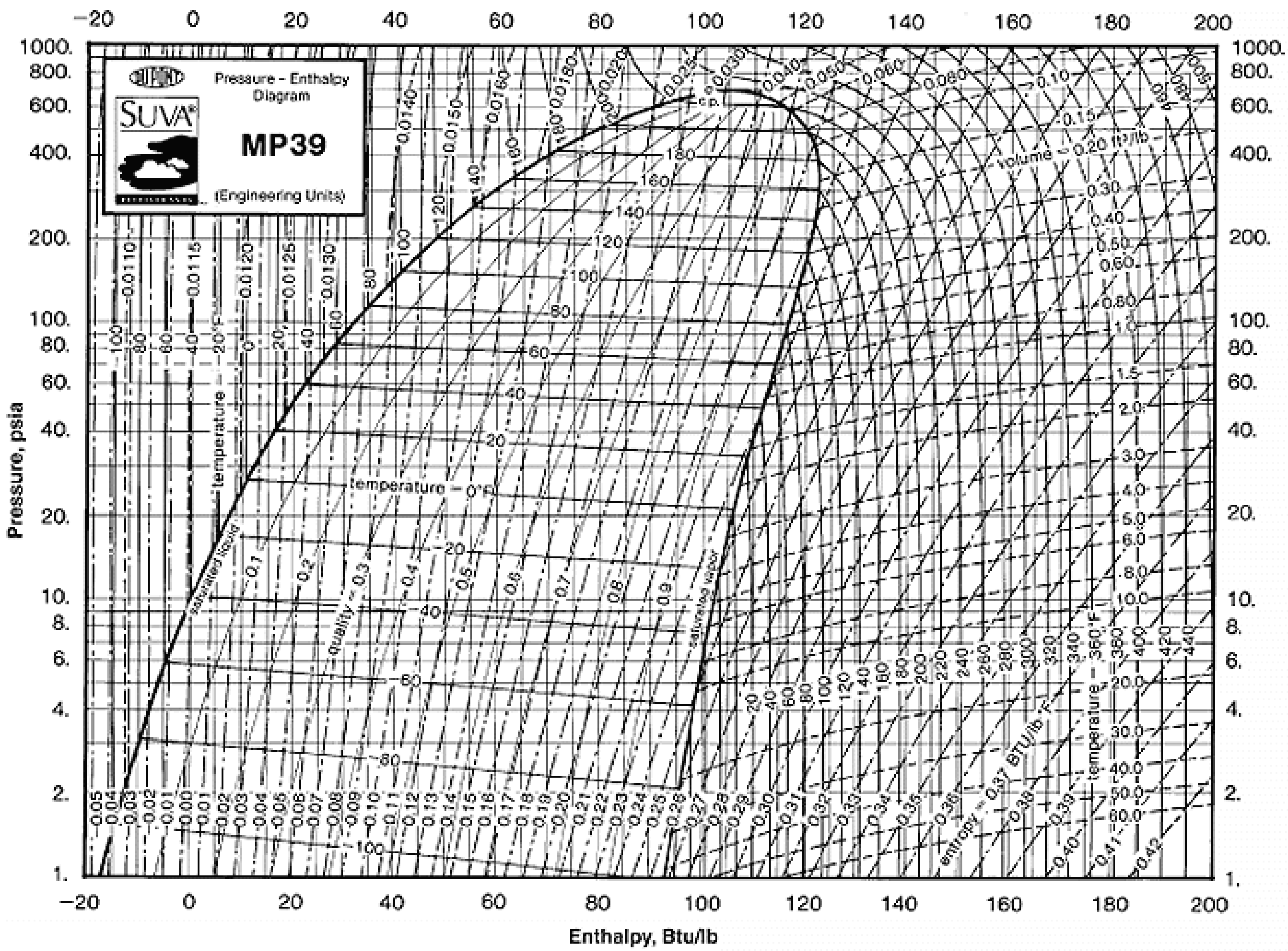

FIGURE 1H. Refrigerant MP39.

Figure 1G applies to HFC-134a, the subject of this text. Figures 2 through 7 use the HFC-134a pressure-enthalpy diagram to show the information available from such diagrams when their potential is understood. This means that some lines are omitted in Figures 2 through 7 (the complete diagram is shown in Figure 1G).

INTERPRETING THE DIAGRAM

The curved line at the left in Figure 2 represents liquid at saturation temperature. Absolute vapor pressure is read on the vertical scale along the left side. Enthalpy (heat content) is read on the bottom scale at a number of different temperatures. To the right is a curve representing saturated vapor. This is vapor associated with liquid, and the type of vapor found in a refrigerant cylinder, a condenser, or a flooded evaporator. All of the data to the right of this curve relate to superheated gas, where no liquid can be present. The space between the saturated vapor and the saturated liquid lines represents a mixture of saturated vapor and liquid.

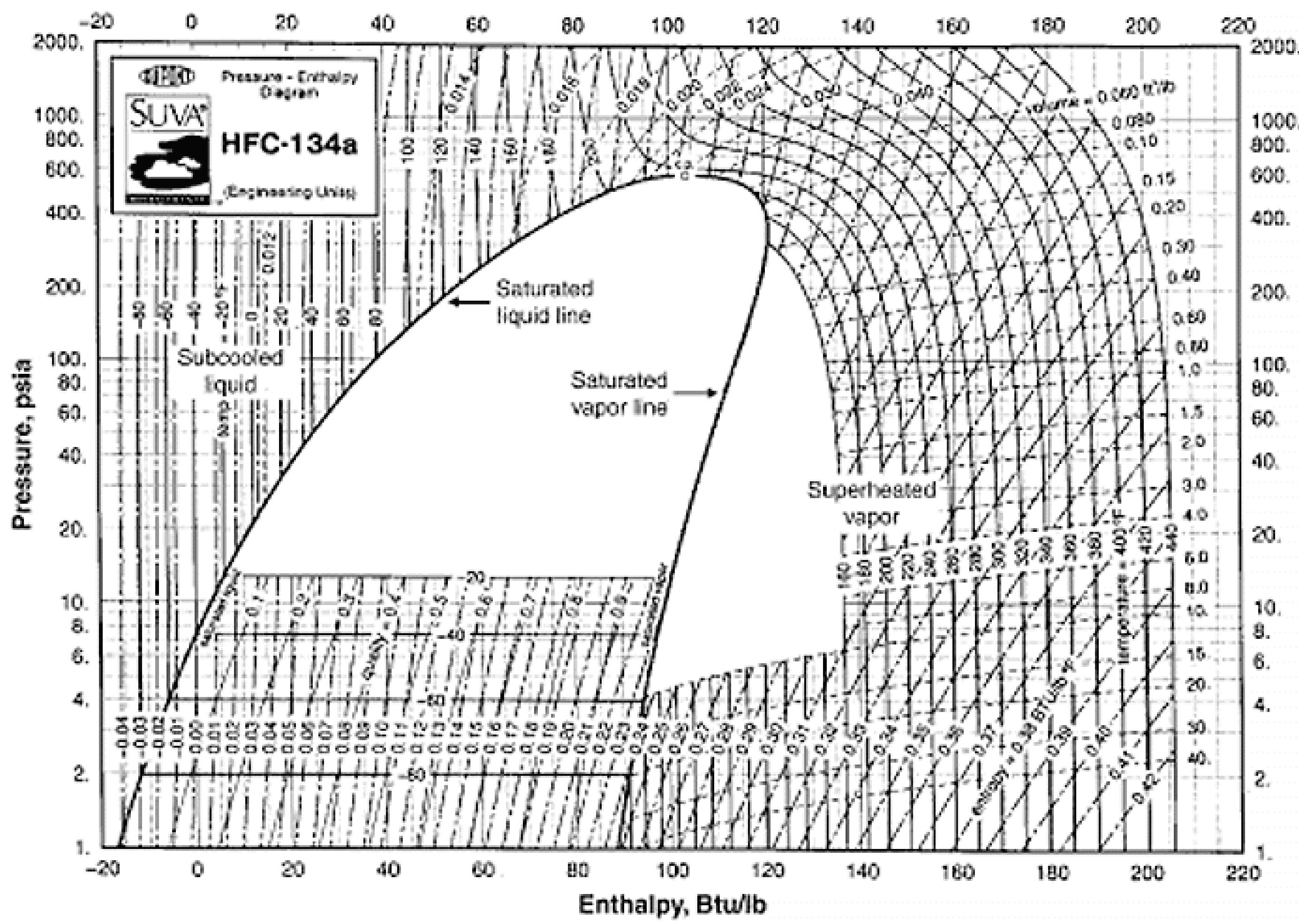

FIGURE 2. Liquid and vapor lines.

PUTTING THE PRESSURE-ENTHALPY DIAGRAM TO WORK

Compression ratio is a factor that is easily found using the pressure-enthalpy diagram, as shown in Figure 3. For example, the evaporating temperature of HFC-134a in a system is 0°F. The condensing temperature is 120°F. Pressures corresponding to these temperatures are read on the left-hand scale. To find the compression ratio, divide the high-pressure value by the low-pressure value.

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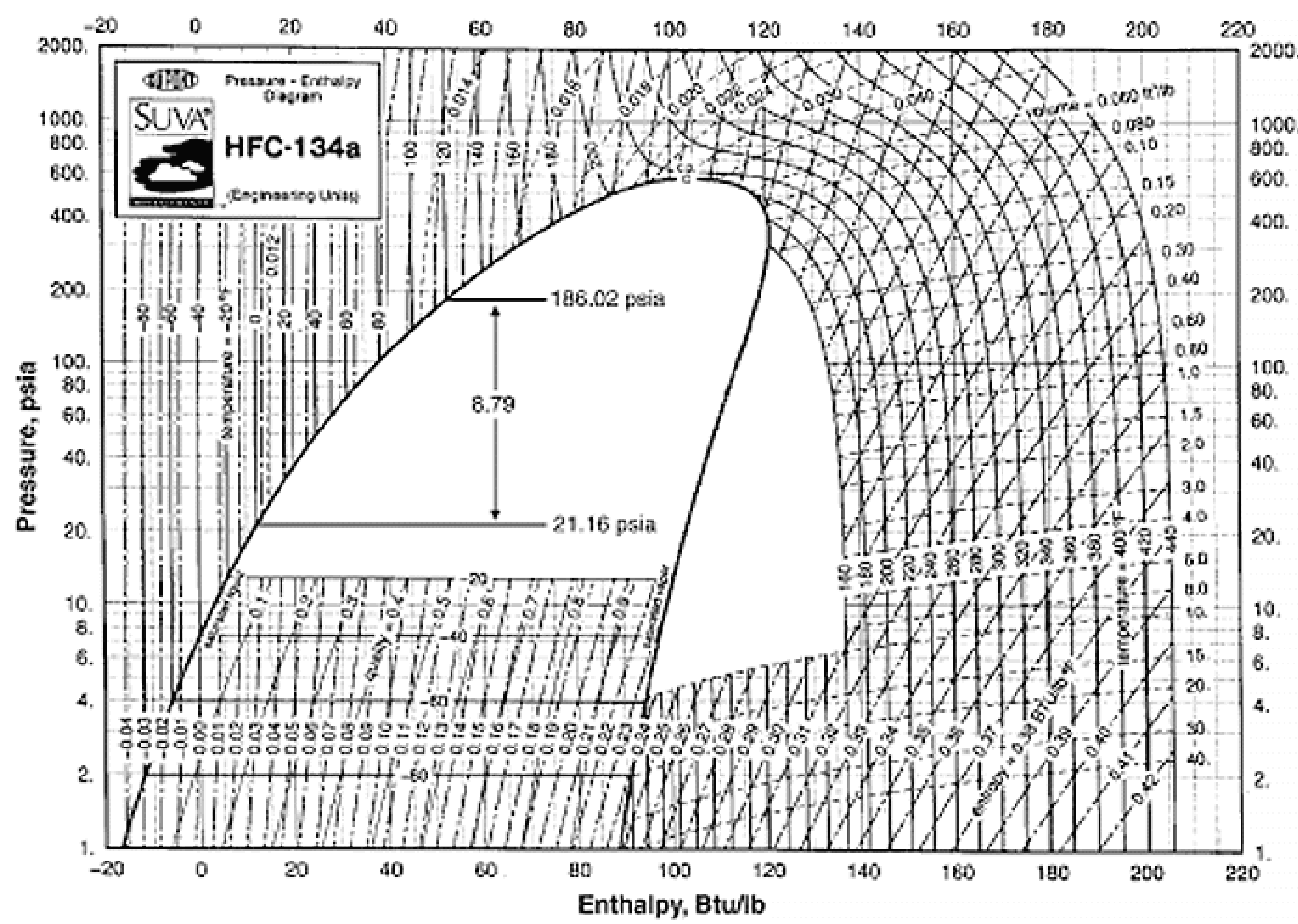

FIGURE 3. Compression ratio.

The exact compression ratio of 8.790010868 would be difficult to use in basic calculations. So a rounded-off figure of 8.79 is used in this example. The pressures at this point must be in psia. Correct values can be taken directly from the chart. A compression ratio of 8.79 means that discharge pressure is slightly less than nine times that of suction pressure. In other words, the compressor must compress the refrigerant gas almost nine times to function efficiently under these conditions. This poses no problem to ordinary reciprocating compressors. They can usually handle compression ratios up to 15. When the ratio exceeds 10, however, a second-stage compressor is recommended. Rotary compressors can deal with compression ratios up to 5, centrifugal compressors up to about 3. The average compression ratio in large-capacity centrifugal units is only about 1.5 or less in each stage.

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CHARTING A REFRIGERATION CYCLE

Figure 4 is a pressure-enthalpy diagram of a typical refrigeration cycle in a system with one pound of HFC-134a. It uses (for this example) evaporating and condensing temperatures of 0°F and 120°F. Points on the diagram are labeled to correspond to locations of equipment in the system. Each step of the cycle can be approached separately.

At Point 1, the refrigerant leaves the evaporator and enters the suction valve of the compressor. In an actual system, there will be some superheat at this point, but to simplify the system none is shown here.

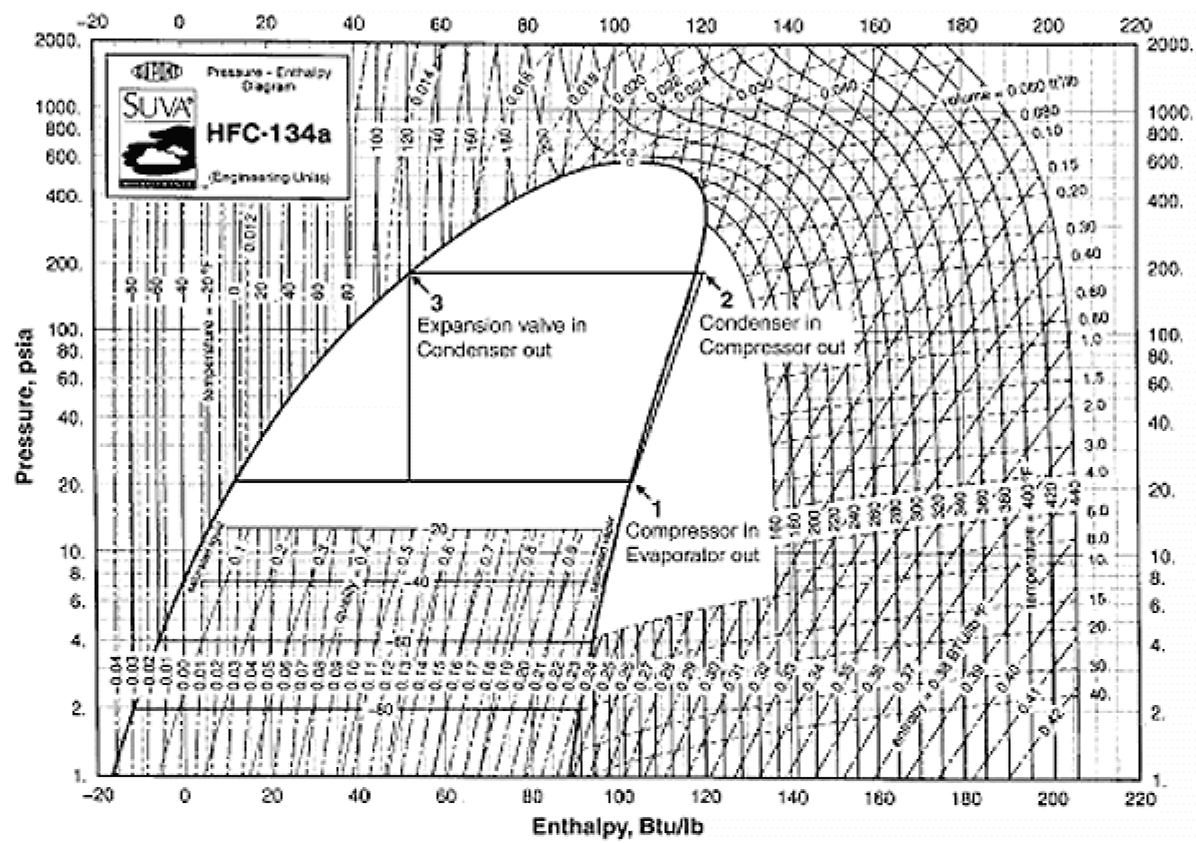

FIGURE 4. Charting a refrigeration cycle.

Point 2 indicates the condition of the refrigerant as it leaves the compressor discharge valve and enters the condenser. Note that the refrigerant is a superheated gas at this point. Point 3 represents the condition of refrigerant as it leaves the condenser and enters the metering device. In this example, the refrigerant is a saturated liquid. By necessity, Figure 4 illustrates the simplest conditions possible. Several refinements would be needed to fit any actual refrigeration system problem. In this and subsequent examples, temperature refers to actual refrigerant temperature, not the temperatures in the vicinity of the condenser or of the product being cooled.

A major phase of the refrigeration cycle is changing liquid refrigerant to a gas at the same temperature. This is done by absorbing enough heat to bring about the change. The heat is called latent heat or heat of vaporization. In the refrigeration cycle, this change takes place in the evaporator. It is the only place where any useful cooling occurs. The other system equipment returns the gas to liquid form and routes it back to the evaporator.

Latent heat can be calculated by use of the diagram in Figure 5. Latent heat is shown by the distance between the saturated liquid line, to the left of the curve, and the saturated vapor line on the right. The difference between the enthalpy of the saturated vapor and that of the saturated liquid is the latent heat of vaporization. In this example, the latent heat of vaporization for HFC-134a at 0° is 91.0 Btu/lb.

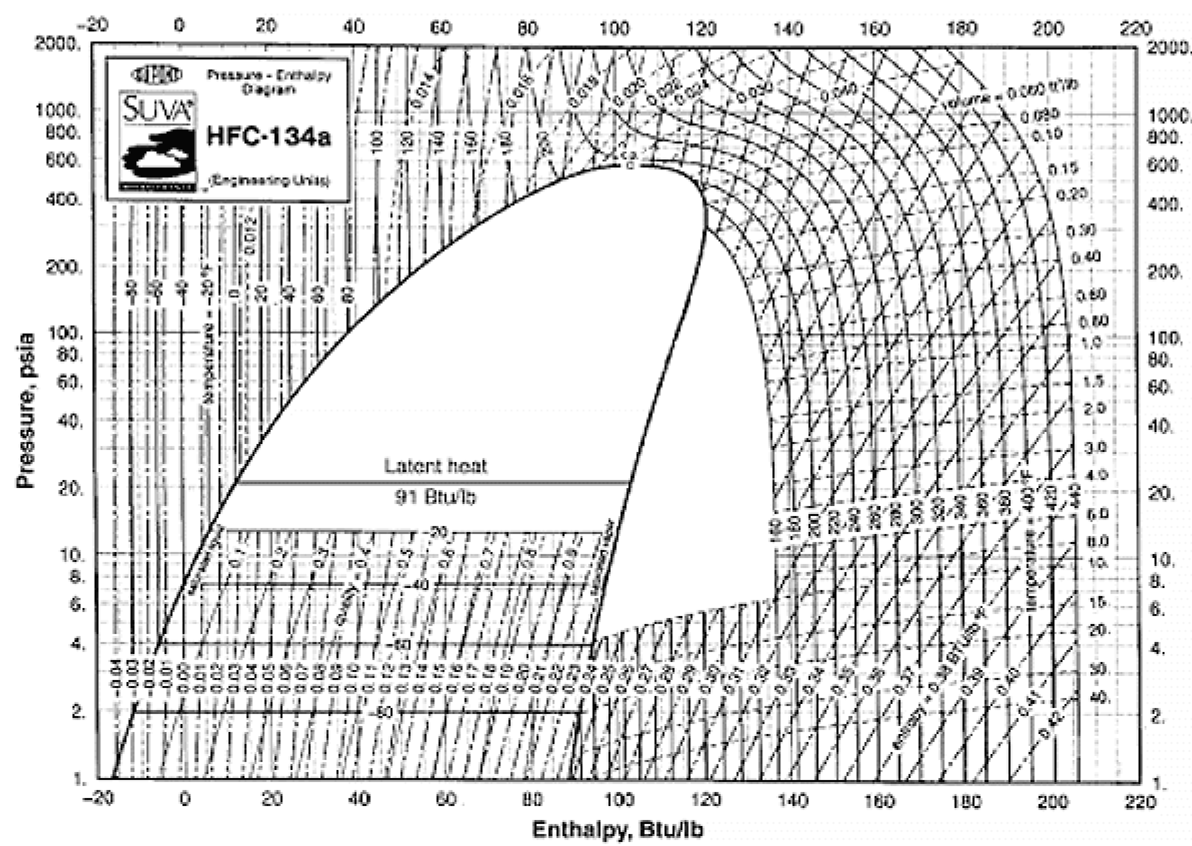

FIGURE 5. Latent heat.

The values for latent heat at various temperatures become smaller as the temperature rises. The critical temperature is the highest temperature at which liquid can exist. At this point, latent heat value is 0 Btu/lb, and there is no difference between liquid and vapor. The critical temperature for HFC-134a is 213.9°F.

Another important factor in analyzing refrigeration cycle performance is net refrigerating effect. You can also get this from a pressure-enthalpy diagram. The heat required to evaporate liquid refrigerant depends on the material being cooled. Not all of this latent heat can be used to produce product cooling. A portion must be used to cool the refrigerant itself from condensing temperature to evaporating temperature. The pressure-enthalpy diagram in Figure 6 provides a worksheet for calculating net cooling effect of the example cycle. Recall the established condensing temperature of 120° and the evaporating temperature of 0°. The enthalpy values at these temperatures can be read from the diagram easily. The difference in enthalpy value at 120° and enthalpy value at 0° is the amount of heat needed to cool the refrigerant. If subtracted from the previously calculated latent heat, the result is the actual amount of product cooling that can be produced by a pound of refrigerant—or, in other words, net refrigerating effect. Figure 6 shows net refrigerating effect to be 50.7 Btu/lb.

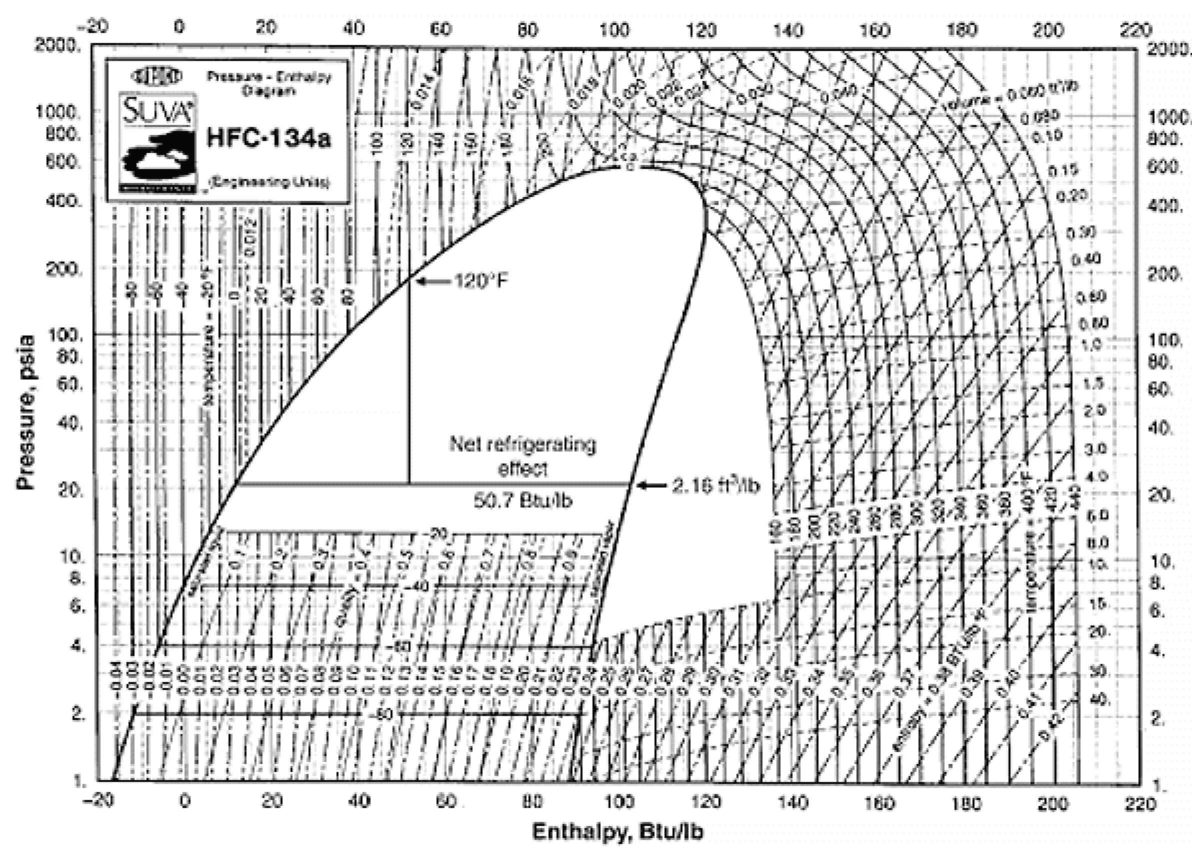

FIGURE 6. Net refrigerating effect and refrigerant volume.

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REFRIGERANT VOLUME

Refrigerant volume is the measure of space occupied by refrigerant vapor. It is found easily in the pressure-enthalpy diagram. Lines of constant volume are roughly horizontal lines on a pressure-enthalpy diagram. In Figure 6, the volume for the saturated vapor in the example cycle at 0° can be calculated as 2.16 ft^3/lb.

REFRIGERANT CIRCULATED

Calculation established a net cooling effect of the example cycle from evaporation of one pound of HFC-134a as 50.7 Btu/lb. This was based on data provided by the pressure-enthalpy diagram in Figure 6. Total capacity of a system depends on how often this one pound of refrigerant can be circulated through the evaporator. Capacity is generally expressed in either Btu per minute, Btu per hour, or in tons of refrigeration. One ton of refrigeration is the amount of heat required to melt one ton of ice in 24 hours. This amounts to 200 Btu per minute, or 12,000 Btu per hour. The capacity requirement of the example system will be established as one ton.

Now we want to figure the amount of refrigerant that must be circulated every minute to produce one ton of cooling under the conditions shown. This is easily calculated by using the net refrigerating effect value from the pressure-enthalpy diagram in Figure 6. The formula is:

$$\begin{aligned}\text{Refrigerant circulated} &= \frac{\text{load, Btu/min}}{\text{net refrigeration effect, Btu/lb}}\\ &= \frac{200\ \text{Btu/min}}{50.7\ \text{Btu/lb}}\\ &= 3.94\ \text{lb/min}\end{aligned}$$

The answer is that 3.94 pounds of refrigerant must be circulated each minute for each ton of refrigeration in this system.

COMPRESSOR DISPLACEMENT

Theoretical compressor size can also be found from a pressure-enthalpy chart or diagram. Multiply the refrigerant circulated in pounds per minute by the volume of gas in cubic feet per pound. This shows theoretical compressor displacement. The volume for the example system was found from the pressure enthalpy diagram in Figure 6 to be 2.16 ft^3/lb. The formula provided in a subsequent paragraph established the refrigerant circulation rate to produce one ton of effective cooling as 3.94 lb/min. The following formula shows how these values are used to find the theoretical compressor displacement:

$$\begin{aligned}\text{Compressor displacement per ton} &= \text{refrigerant circulated} \times \text{gas volume}\\ &= 3.94\ \text{lb/min} \times 2.16\ \text{ft}^3\text{/min}\\ &= 8.52\ \text{ft}^3\text{/min}\end{aligned}$$

To simplify calculating in this example, the volume of saturated vapor was used. This would never occur in practice, however. The superheated gas volume corresponding to a valid situation could have been determined from the diagram in Figure 6. The method will be shown at a more appropriate point in the text.

The last few sample pressure-enthalpy diagrams have given data used to research factors from load, or capacity, to theoretical compressor size. With the same data, you could reverse these steps. Starting with a theoretical compressor size, you could determine how much cooling it would produce under a given set of conditions.

COMPRESSOR HEAT

After being circulated through the evaporator, the refrigerant gas flows to the compressor. There it is compressed. At this point, entropy becomes a factor. When a refrigerant is compressed and entropy remains the same, it is called adiabatic compression. This means that there is no flow of heat away from or into the gas. Gas becomes hot during compression because molecules are pushed together, producing an increase in temperature. Normally, when a refrigerant is compressed, the entropy does not remain the same, but it is often close. The gas is heated from the friction of moving parts and the restricted flow through small valves, etc. Heat also flows out of the gas when the compressor gets hot and heat is lost by radiation. In many cases, these two effects cancel each other out.

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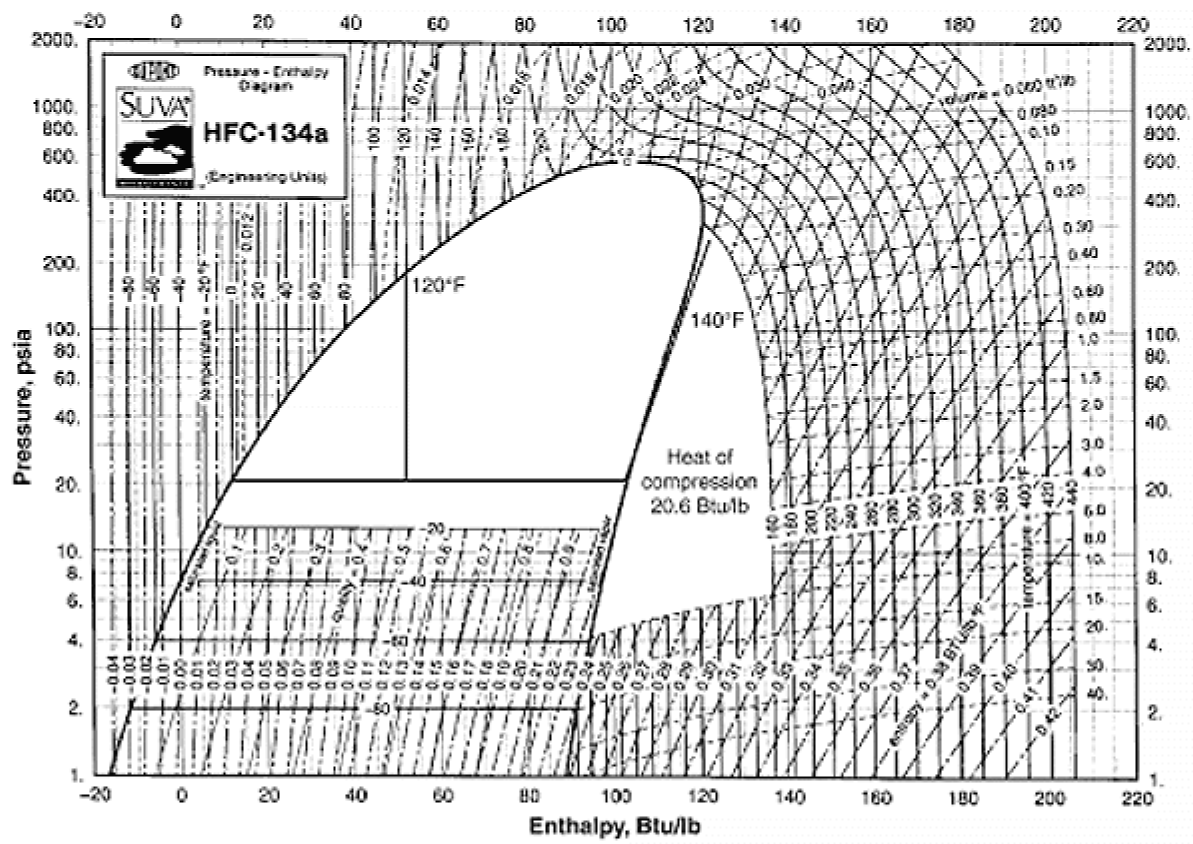

FIGURE 7. Compressor heat.

For simplification, assume again that saturated vapor enters the compressor, and that compression is adiabatic. Only with this assumption can the line of constant entropy in Figure 7 be used as the basis for calculations. The point to be reached on this line will depend on the condenser in use.

The opening and closing of the compressor discharge valve depends on which side has the higher pressure against it. On one side of the valve is the gas being compressed by the compressor. On the other side is the pressure from the condenser. Whatever the temperature of the condensing liquid, it will always have a corresponding vapor pressure. The pressure-enthalpy diagram in Figure 7 shows that the pressure corresponding to liquid HFC-134a at 120°F is about 186 pounds per square inch absolute (psia.) The pressure against which the compressor discharge valves must open is controlled by the vapor pressure of the liquid in the condenser. This pressure will be the same in the piping between the condenser and the compressor as it is within the condenser, except for a slight correction for pressure drop in the piping. When the pressure in the compressor reaches condenser pressure, the compressor work is, in effect, finished.

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AZEOTROPE REFRIGERANTS

Some refrigerants, such as R-502, are mixtures of other refrigerants. They are called azeotropes. Once mixed, the refrigerant acts like one refrigerant with one boiling point for each pressure.

NEAR-AZEOTROPES

Some of the new refrigerants that are being introduced to replace the CFCs are called near-azeotrope mixtures. MP39, for example, is a mixture that has a range of boiling points at a given pressure. The point at which the MP39 begins to boil is called the bubble point. The point at which the last drop of refrigerant has boiled away is called the dew point. The difference in temperature between the bubble point and the dew point is called the glide. In the condensing part of the cycle, the process is reversed. The point at which the refrigerant starts to condense is the dew point. The bubble point is reached as the last drop of refrigerant is condensed.

USING MOLLIER DIAGRAMS FOR NEAR-AZEOTROPES

Due to the temperature range of the glide, there is no single saturation temperature for a given pressure when using a near-azeotrope. When viewed on the Mollier diagram, the saturation temperature appears as a slightly slanted line between the saturated vapor and saturated liquid lines (see Figure 1H). The absolute pressures are horizontal, as they are in diagrams of other refrigerants.

PLOTTING MP39

Plotting the refrigeration cycle on a MP39 diagram is the same as in a single component or azeotrope refrigerant. The condenser is plotted along the constant pressure line. This does not line up with the constant temperature line. The diagram will show the drop in temperature across the condenser (the glide). The evaporator is plotted the same way. The temperature glide of the evaporator can also be determined. The rest of the cycle is plotted the same.

USING THE DIAGRAM TO ESTIMATE ENERGY REQUIREMENTS

Previous portions of this chapter explained how to use pressure-enthalpy diagrams to analyze the compression cycle. Entropy as a factor in system performance was also discussed. The pressure-enthalpy diagram also can be used to estimate the energy required for a compressor to perform properly.

It requires energy to compress a gas. The energy added to the gas to compress it is the difference between the enthalpy of the gas as it enters and leaves the compressor. Refer back to Figure 4. Note that “compressor in” and “compressor out” points are shown. Measure the enthalpy on the bottom scale between the two points. This shows the energy in Btu per pound added to the gas to compress it. Figure 7 shows the result of this calculation. It is 20.6 Btu/lb.

Now look at Figure 4 again. The theoretical discharge temperature is the point at which the compression line meets the condensing pressure line. Figure 7 shows that it is 140°F. This is 20° higher than condenser temperature. In practice, discharge temperature may be 25 to 40° higher than the theoretical figure. The difference between actual and theoretical temperatures relates to several factors. They include compressor design and operating conditions, and the "balancing out" of heat loss and heat gain to keep entropy constant.

Despite the differences between the actual operating parameters of a system and the theoretical values plotted, plotting a system can be useful. Plotting a cycle can show what will happen to capacity, horsepower, and heat of rejection when evaporator temperature is raised or lowered. It will show the effect on a system of changes in superheat, subcooling, and condensing temperature. It is possible to calculate the effect of changes of one or more of the system conditions on the system as a whole. By plotting the refrigeration cycle on a pressure-enthalpy diagram, you can see the complete system and the interrelationships of its components.

COMPRESSOR POWER

Heat added to gas during compression is calculated in Btu per pound. To convert this value to power, a time factor is required. Multiply compressor heat in Btu per pound by the refrigerant circulated in pounds per minute. This gives the compressor power in Btu per minute. The following example shows this calculation. The data are from Figures 6 and 7.

Compressor power per ton (heat and time)	
	= compressor heat x refrigerant circulated
	= 20.6 Btu/lb x 3.94 lb/min
	= 81.6 Btu/min per ton

The most common way to state compressor power on a pressure-enthalpy diagram is horsepower per ton of refrigeration. It is found by dividing the Btu per minute per ton by 42.43 Btu per minute. For the example above, the compressor needs 1.91 horsepower to produce one ton of refrigeration. Watts are used as the measure of electric power. Electric power can be found by multiplying the horsepower by 746 watts per horsepower. This calculation assumes that the motor is 100% efficient. In reality, the electric power needed will be greater, because the motor will always be less than 100% efficient.

Currently, there is a trend to report equipment capacity in Btu per hour at a specified condition. With Refrigerant HFC-134a, 1.91 horsepower of compression produces one ton of refrigeration in the example shown. This is true only when the evaporator, condenser, superheating, and subcooling are exactly as stated in the example. Changes in any of these temperatures or pressures will change the horsepower required to produce one ton of refrigeration. To produce the same amount of cooling at a lower evaporator temperature takes more horsepower if the condensing temperature remains the same.

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Occasionally foot-pounds per second (ft-lb/s) may be used as a power term. It takes 1,050 ft-lb/s to produce the ton of refrigeration in this example. Whichever term is used, it is the theoretical power necessary to produce one ton of refrigeration under the above conditions.

CONDENSER DATA FROM PRESSURE-ENTHALPY DIAGRAMS

As the refrigerant passes through the condenser, the gas is cooled to the condensing temperature by removing the superheat. As more and more heat is removed, the refrigerant will start to condense. However, it remains at the condensing temperature. The pressure stays at the vapor pressure of the liquid, which is 120°F.

For the example system, the pressure-enthalpy diagram in Figure 8 shows that the amount is 5.7 Btu/lb of refrigerant. The vapor starts condensing to a liquid when this amount of heat is removed. As more heat is removed, more liquid is formed. Eventually, all of the one pound of vapor becomes liquid refrigerant. As shown in Figure 8, 65.9 Btu are removed to change one pound of vapor to one pound of liquid. When fully condensed, the liquid is ready to go back to the evaporator and through the cycle again.

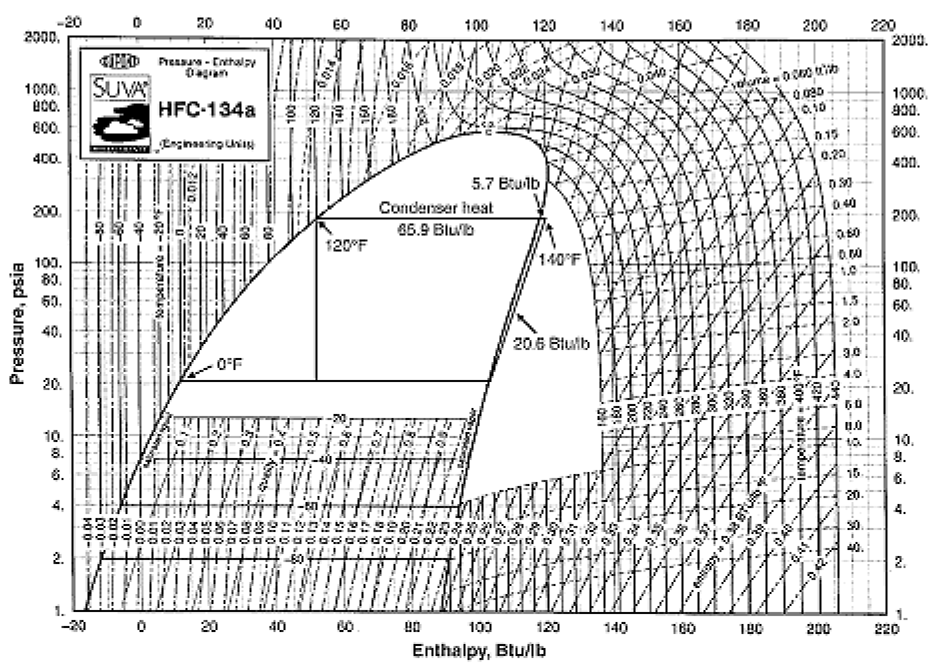

FIGURE 8. Condensing vapor to liquid.

Heat transfer from a gas is not as effective as from a condensing liquid. Thus, the amount of superheat in the gas en route to the condenser is important to system capacity. Removing heat from a gas could require several times the condenser surface used to take the same heat from a liquid. In practice, this means that the more superheat in gas from the compressor, the larger the condenser needed.

EFFECTS OF LOWER CONDENSING TEMPERATURE

As noted earlier, an important use of pressure-enthalpy diagrams is to compare different operating conditions. For example, say that a condenser is functioning at 100°F instead of 120°F. Perhaps it was just cleaned, the ambient air is cooler, or it might be a replacement condenser with a slightly larger surface. Figure 9 shows that the reduction in condensing temperature will increase capacity and reduce horsepower requirements, or both, assuming that other conditions remain the same.

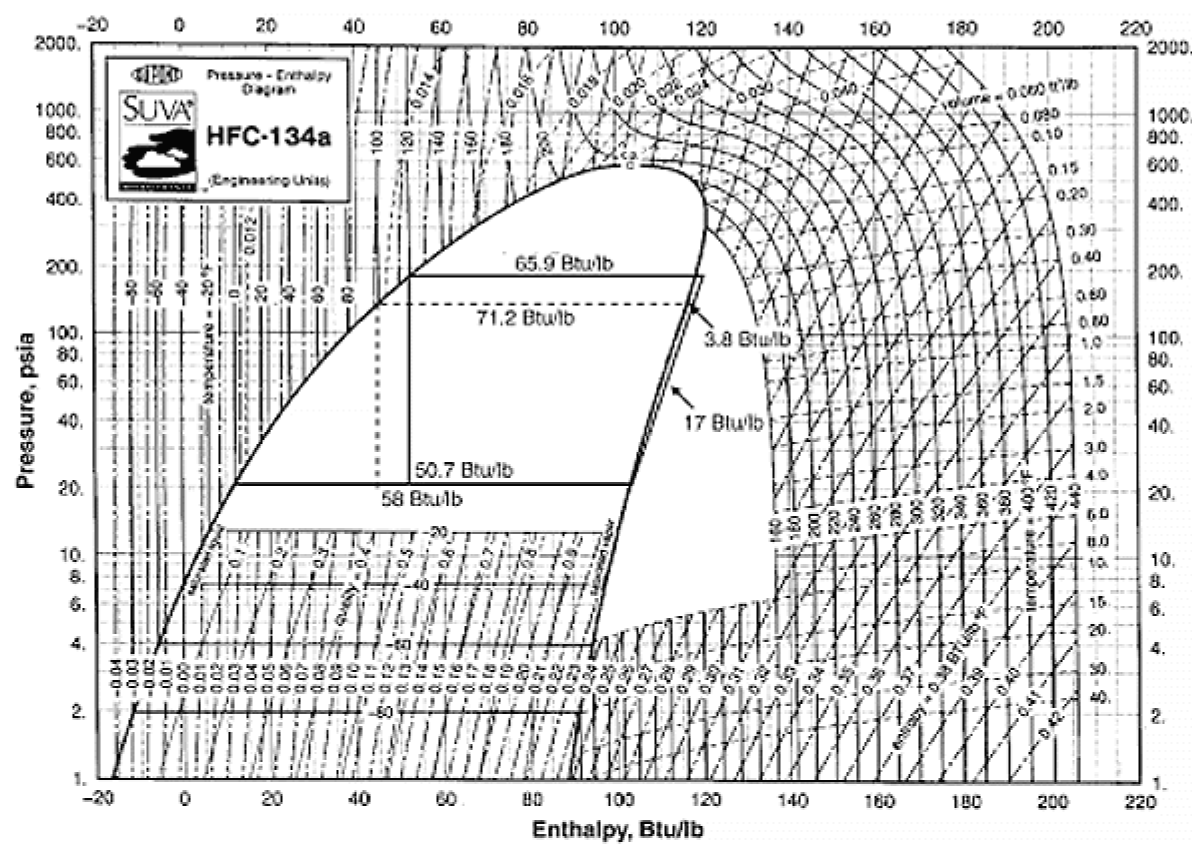

FIGURE 9. Lower condensing temperature.

Figure 9 shows that latent heat remains the same, but net refrigeration effect is increased, because the liquid entering the evaporator requires less cooling. Less compressor heat and power are required since the gas is not compressed at as high a pressure. At 120°F condensing temperature, 20.6 Btu/lb are added to the gas to compress it. At 100°F, only 17 Btu/lb are added. As a result, discharge temperature will be lower and the amount of superheat in the gas will be slightly less. The amount of heat removed to condense the liquid will be greater at the lower temperature, resulting in a slight net increase in the heat removed in the condenser.

LIQUID SUBCOOLING

A pressure-enthalpy diagram helps explain what happens when a refrigerant is subcooled. Look at Figure 10. Liquid pressure is still governed by vapor pressure, so the pressure of liquid in the condenser remains at vapor pressure level. The heat content of subcooled liquids can be assumed to be the same as the heat content of the liquid if it were saturated at the lower temperature. For example, the enthalpy of HFC-134a liquid at 120°F and 186 psia is 52.4 Btu/lb. If the liquid is subcooled (with no change in pressure) to 100°F, the enthalpy is assumed to be the same as the saturated liquid at 100°F. In this example, the enthalpy at 100°F and 186 psia is 45.1 Btu/lb. For moderate pressures and subcooling, this method will be reasonably accurate.

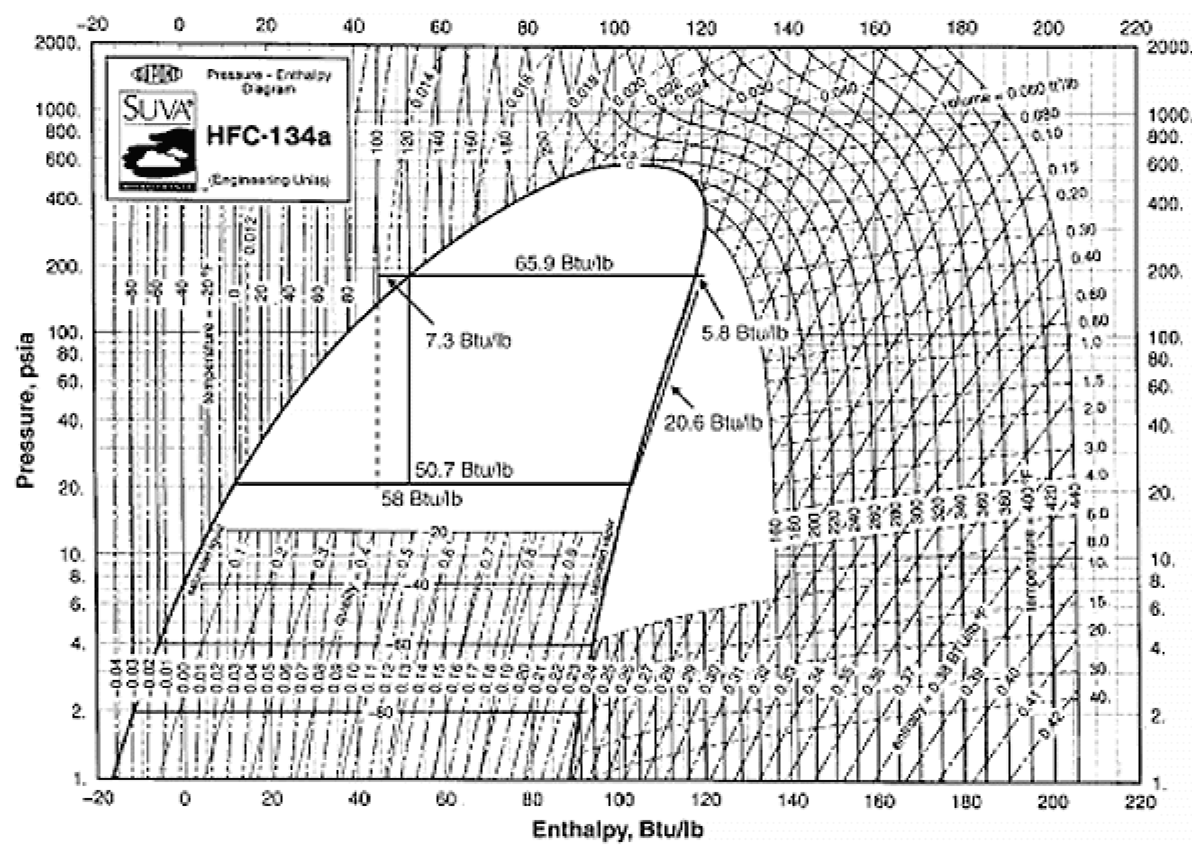

FIGURE 10. Subcooling.

The example in Figure 10 assumes that the liquid is subcooled 20°, from 120°F to 100°F. Figure 10 shows that this amount of subcooling removes 7.3 Btu/lb. The net refrigeration effect will be the same as when the entire condenser is operated at 100°F. The compressor must still do the same amount of work whether the liquid is subcooled or not. Obviously, subcooling of the liquid produces more capacity. The simple calculations using the pressure-enthalpy diagram will determine the amount of the increase. Subcooling effects will be compared in subsequent paragraphs where appropriate.

SUCTION VAPOR SUPERHEATING

A pressure-enthalpy diagram can show the effect of different operating conditions. Look at Figure 11. It shows what happens if the suction gas is superheated. In the current example, the refrigerant gas is heated to 65°F. Since the saturated temperature of the gas leaving the evaporator is 0°F, the superheat is 65°F. Its pressure does not change, because it is controlled by the presence of the liquid in the evaporator.

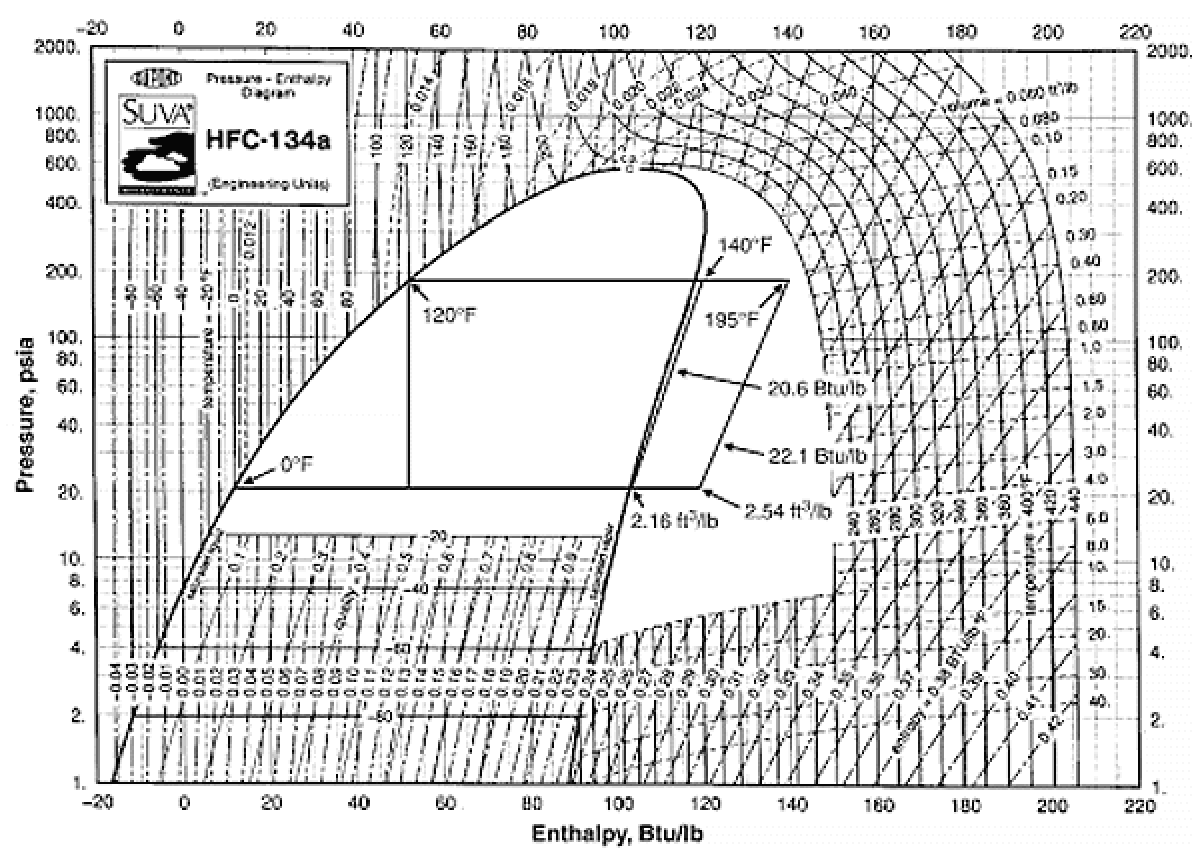

FIGURE 11. Superheating.

Some reaction has to take place when a gas is heated. The volume increases from 2.16 ft^3/lb to about 2.54 ft^3/lb. The gas entering the compressor now has a different entropy. As a result, this new value will follow a different line on the pressure-enthalpy diagram. Figure 11 shows that the discharge temperature is considerably higher and directly relates to the superheat value. A diagram that indicates excessive discharge temperature shows the need to reduce superheat, perhaps by insulating the suction line. Figure 11 shows how the change in discharge temperature can be found when the temperature of suction gas is known.

Heat of compression is also increased at higher suction gas temperatures. The amount of increase can be calculated from the diagram. Whether increased heat of compression results in more power depends on what occurs in the condenser and evaporator, and on whether the weight of circulated refrigerant increases or decreases.

PROPERTIES COMPARISON—ONE TON REFRIGERATION CAPACITY

Table 1 summarizes the effects of changes in operating conditions analyzed thus far. We assume that capacity stays the same in the continuing example—one ton, or 200 Btu/min. The conditions selected are an evaporator at 0°F and a condenser at 120°F. The data in Table 1 compare capacity levels in a simple system under the following conditions:

- with liquid subcooled at 100°F
- with suction gas superheated at 65°F
- when both conditions exist at the same time
- with condenser at 100°F.

The first line in Table 1 shows that compression ratio depends only on evaporating and condensing temperatures. The second line shows that net refrigeration effect depends only on the latent heat and heat content of the liquid as it enters an evaporator. When the liquid is subcooled, net refrigeration effect per pound increases. When the entire condenser is cooled, net refrigeration effect increases in the same proportion. Lines 3 through 7 show the following:

- When capacity stays the same, the rate at which refrigerant circulates varies inversely with net refrigeration. Fewer pounds of refrigerant are needed when Btu per pound are higher.
- The specific volume of the gas depends only on the existing superheat level for a particular evaporating temperature.
- Compression heat is not affected by subcooling, but is increased by superheating. The compressor heat is decreased when condensing temperature is lowered.

Table 1 Capacity levels under different operating conditions.

SYSTEM CAPACITY PER TON					
	0°/120°				0°/100°
		Liquid subcooled to 100°F	Suction superheated to 65°F	Both	

Compression ratio	8.79	8.79	8.79	8.79	6.57
Net refrigeration, Btu/lb	50.70	58.00	50.70	58.00	58.00
Refrigerant circulated, lb/min	3.94	3.45	3.94	3.45	3.45
Gas volume, ft³/lb	2.16	2.16	2.54	2.54	2.16
Compressor heat, Btu/lb	20.60	20.60	22.10	22.10	17.00
Compressor power, hp	1.91	1.67	2.05	1.79	1.38
Compressor displacement, ft³/min	8.54	7.45	10.00	8.80	7.45

To find compressor power, multiply compression heat (Btu per pound) by the rate at which the refrigerant circulates through the system (pounds per minute.) Subcooling the liquid has no effect on heat of compression, but does reduce required horsepower. This is because fewer pounds of refrigerant circulated produce the same refrigeration effect. In this example it is one ton, or 200 Btu per minute. Superheating the suction gas increases the horsepower requirement. In the example, comparison horsepower is reduced slightly when there is both subcooling and superheating. The amounts of subcooling and superheating were chosen to show the effect. They do not necessarily correspond to a real situation. When condensing temperature is lower—for example, 100°F— horsepower requirement is also much lower, as Table 1 shows.

The seventh line shows the compressor displacement in cubic feet per minute needed for various capacity values. Displacement can be changed by changing either motor speed or compressor size. Neither of these methods is common in the field. It is important to research other possible remedies first.

VARYING CAPACITY VALUES WITH SAME DISPLACEMENT

The data in Table 2 assume that the compressor in the example Always operates at the same speed. It also assumes a displacement of 8.5 ft³/min. Compression ratio depends on evaporating and condensing temperatures. Therefore, the compression ratio should be and is the same as that shown in Table 1. Net refrigeration effect in Btu per pound is also the same as shown before. The rate at which refrigerant circulates now depends on what goes through the compressor. Subcooling the liquid has no effect on the rate of circulation. Note what happens when the gas is superheated. The rate of circulation is reduced due to the increased volume. Further comparison shows that gas volumes are the same in both tables.

Table 2 Varying capacity values.

SAME COMPRESSOR DISPLACEMENT (8.5 ft³/min)					
	0°/120°				0°/100°
		Liquid subcooled to 100°	Suction superheated to 65°F	Both	
Compression ratio	8.79	8.79	8.79	8.79	6.57
Net refrigeration, Btu/lb	50.70	58.00	50.70	58.00	58.00
Refrigerant circulated, lb/min	3.94	3.94	3.35	3.35	3.94
Gas volume, ft 3 /lb	2.16	2.16	2.54	2.54	2.16
Compressor power, hp	1.91	1.91	1.91	1.91	1.57

Load, Btu/min	200.00	228.00	170.00	194.00	228.00
Load, tons	1.00	1.10	0.80	0.90	1.10

In this example, compression power is reasonably consistent with subcooling or superheating, or both, in effect. As we saw before, the heat of compression on a pound basis is slightly higher when the gas is superheated. Gas volume, however, is also increased. As a result, fewer pounds of refrigerant are required to circulate. As a rule, higher compressor heat and reduced circulation cancel each other out. Thus, compressor power stays about the same.

The effect of changes in Table 2 is greater in the area of capacity, or load. A simple cycle was chosen for the example. This allows calculations on the basis of 200 Btu/min, or one ton of refrigeration effect. Subcooling the liquid is always productive. In the subject situation, it gives an increase of about 14% in capacity. Superheating the gas causes a decrease in capacity. In this instance, the capacity is reduced almost 13%. The net result of both effects is a slight decrease in capacity.

Understand that the example is for illustration purposes only. The accepted method of subcooling liquid refrigerant is a liquid-vapor heat exchanger that takes heat from the liquid and adds it to the suction gas. Additional suction line superheat may come from absorbing heat from the air as it goes to the compressor. Use of a liquid-vapor heat exchanger in a HFC-134a system may give a net capacity increase. This is different from the decrease shown in Table 2. The increase in capacity usually remains the same, whether the condensing temperature is reduced to 100°F or the liquid is subcooled to 100°F. The power required, however, is much less when condensing temperature is reduced.

SUMMARY

The calculation of refrigerant properties using pressure-enthalpy diagrams gives only theoretical answers–but they can be of real help in many application problems. Use of the diagram, for example, can give a picture of the entire refrigeration cycle and the connection of one part to another.

Calculations based on the diagram are a valuable means of comparison. They are a valuable tool to assist in evaluating the effect of changes in operating conditions. For any piece of equipment, there are factors that tend to affect theoretical calculations. They include such things as power loss in the motor, friction losses, volumetric efficiency of the compressor, pressure drops in lines, and ratio of heat losses and gains. These factors remain at about the same level when operating conditions are changed. Therefore, corrections to theoretical answers from the diagram will for the most part cancel out when comparing operating variables.

The pressure/temperature relationship in the condenser and evaporator can be found on the diagram. From these values it is easy to calculate compression ratio.

PRESSURE-ENTHALPY CHARTS AND THEIR USE

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Using the diagram to determine the effect on capacity of different discharge temperatures will help determine the feasibility of insulating the suction line to reduce superheat and discharge temperature. Use the diagram to find out how much capacity can be increased by lowering the condensing temperature. Temperature can be lowered by cleaning, by using colder water or a faster flow rate, or by using bigger blowers on air-cooled equipment. The effect on capacity or compressor size or displacement for different evaporating temperatures can be found. The effect of liquid subcooling and vapor superheating can be determined.

Pressure-enthalpy diagrams are tools that you can use to better understand refrigeration cycles. They help the technician determine if system operation is normal. The effect of changes in operation can be estimated.

Know how to read pressure-enthalpy diagrams and do the simple calculations involved. Their use in field service and installation pays rich dividends in all phases of refrigeration work.