Refrigeration Service Engineers Society
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PLOTTING THE A/C CYCLE ON A PRESSURE-ENTHALPY (MOLLIER) DIAGRAM

INTRODUCTION

The basic relationship between pressure and boiling point is at the heart of any discussion of the refrigeration cycle. The boiling point of a liquid is affected by the pressure exerted on its surface. For example, the boiling point of water is 212°F at sea level pressure (generally accepted as 14.7 psi).

Atmospheric pressure is the pressure exerted on the earth's surface by the weight of the air in the atmosphere above the earth's surface. It varies with elevation above sea level and weather conditions. At 14,000 ft (about 2.65 miles) above sea level, atmospheric pressure is reduced to 8.4 psi because less air is above the surface at that elevation (see Figure 1). With the pressure on its surface reduced to this level, water will boil at a temperature of 185°F.

The most important rule in refrigeration is that a liquid's boiling temperature is affected by changes in the pressure exerted on it. An increase in pressure increases the boiling temperature. A decrease in pressure decreases the boiling temperature. This principle applies to the fluorocarbon refrigerants used in the HVAC/R industry. An air conditioning system uses mechanical components to manipulate pressure and establish coil temperatures useful for heat transfer. Desirable refrigerants have low boiling points while being nontoxic, nonflammable, and chemically stable.

Any stable material has a predictable boiling point at a given pressure. Physical characteristic curves, or *pressure-enthalpy diagrams*, can be developed for stable refrigerants. To develop these curves, it is necessary to vary the pressure and measure the amount

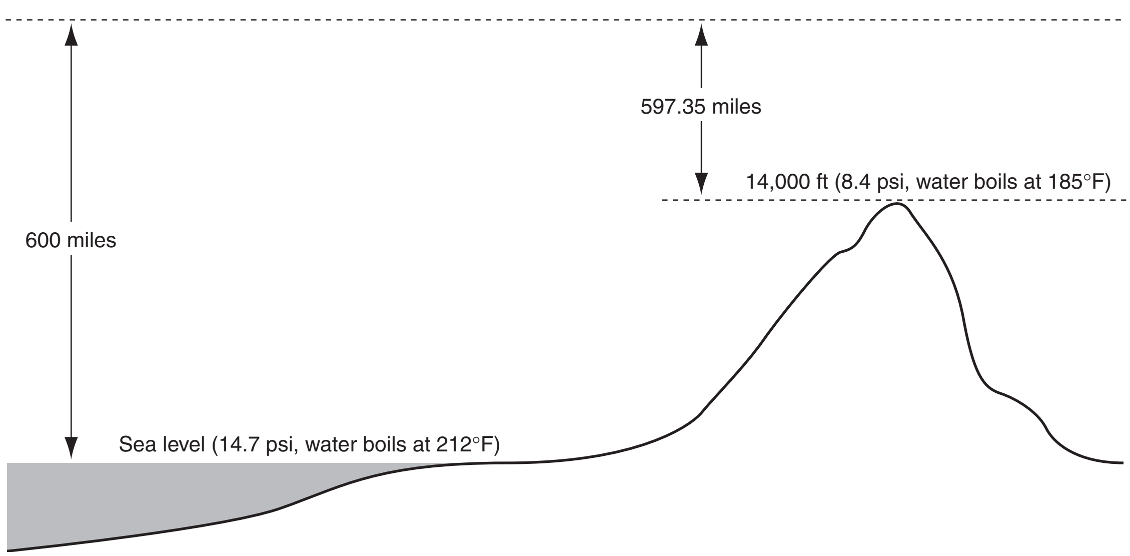

Figure 1. Changes in pressure cause changes in boiling points of liquids *(not drawn to scale)*

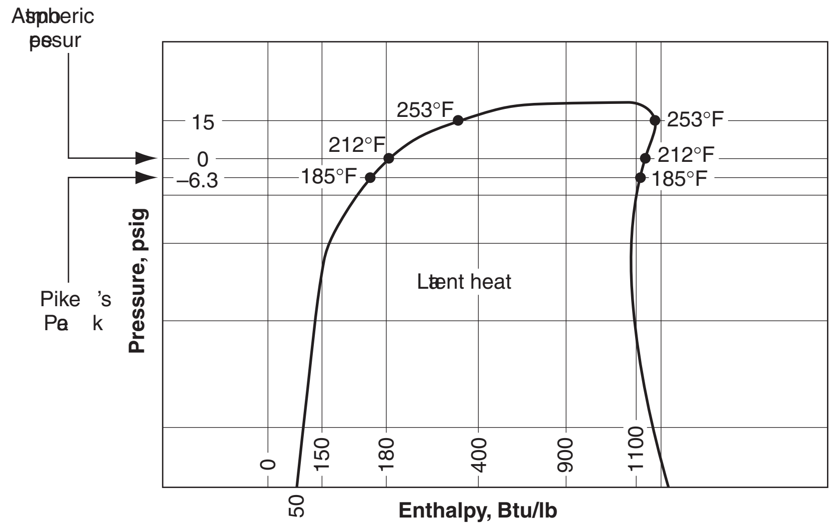

Figure 2. Pressure-enthalpy diagram for water

of heat required to cause the liquid to boil and the vapor to liquefy.

Look at Figure 2, which shows the pressure-enthalpy diagram for water. As you can see, at a pressure of 15 psig (29.7 psia), water begins to boil at 253°F. At that pressure/temperature condition, the addition of 1,165 Btu of heat energy is required to change each pound of water to steam. At 0 psig (14.7 psia), water begins to boil at 212°F. At this pressure/temperature condition, the addition of only 970 Btu/lb of heat energy is required to turn water to steam. On top of Pike's Peak (Colorado, 14,110 ft), where the atmospheric pressure is –6.3 psig (8.4 psia), water begins to boil at 185°F and only about 880 Btu/lb of latent heat must be added to convert water to steam. As stated earlier, these types of characteristics can be plotted for any stable material that has the ability to change its state without changing its chemical composition. It then becomes a matter of selecting a material that changes at pressures and temperatures appropriate for heating and cooling a home.

R-22 has been widely used in the HVAC/R industry. Today, additional refrigerants are being used because of environmental concerns. (Most major manufacturers are producing equipment that utilizes R-410A in place of R-22.) Like water, all of these refrigerants can be made to change from a liquid to a vapor or from a vapor to a liquid. This is accomplished by changing the pressure to which the refrigerant is subjected, or by changing its heat content using mechanical means. Sound engineering fundamentals must be applied in order to assemble matched components that will cause the pressures and temperatures to change at the desired point.

Temperature-pressure charts like the one shown in Table 1 can be used to determine the boiling point of a refrigerant at any given pressure. For example, if the suction pressure of a system utilizing R-22 is 68.5 psig, the boiling point (evaporator temperature) for the refrigerant will be 40°F. If the suction-line temperature is measured at the same point as the pressure and is found to be 50°F, the refrigerant has been *superheated* by 10°F. If the refrigerant used is R-410A and the evaporating temperature is the same (40°F), the corresponding pressure will be 118 psig. Again, any additional temperature (above 40°F) is superheat.

With the data provided by a temperature-pressure chart and the knowledge of how much heat is required to boil a pound of refrigerant, a pressure-enthalpy diagram can be developed for R-22 similar to the one illustrated for water in Figure 2. Using such a diagram, you can plot the performance of any air conditioning system and identify all the work done by the system.

Cooling capacity normally is determined at 80°F dry-bulb/67°F wet-bulb indoor and 95°F dry-bulb outdoor temperatures. Let's examine the pressure-enthalpy diagrams for typical 12 SEER split systems that use R-22 and R-410A. Enthalpy values for both refrigerants have been taken from tables found in the ASHRAE *Fundamentals Handbook* that list the thermophysical properties of refrigerants. The cooling performance is plotted for each system operating on a day when the ambient outdoor temperature is 90°F and the wet-bulb temperature of the air entering the evaporator is 63°F.

PLOTTING THE COOLING CYCLE

A *Mollier diagram* (another name for a pressure-enthalpy diagram) locates system operating pressures on the vertical (y) axis and enthalpy values on the horizontal (x) axis. Physical properties of the refrigerant are represented by a curve that roughly resembles a thumb or a tongue. The curve for R-22, shown in Figure 3 on the next page, begins at a point just to the left of –20 Btu/lb on the enthalpy scale and returns to the scale at about 90 Btu/lb. The curve for R-410A, shown in Figure 5 on page 9, begins at –30 Btu/lb on the enthalpy scale and returns to the scale at a point just to the right of 100 Btu/lb.

Any plot point located directly on the left side of the curve describes the heat content of refrigerant that is a saturated liquid. Any plot point located directly on the right side of the curve describes the heat content of refrigerant that is a saturated vapor. Plot points falling within the envelope of the curve itself describe refrigerant that is a mixture of saturated liquid and vapor. Within the curve, plot points closer to the left side indicate that the mixture has more liquid than vapor. Plot points closer to the right side indicate that the mixture has more vapor than liquid. Plot points outside the curve to the left of the "saturated liquid" line indicate that the refrigerant is a subcooled liquid. Plot points outside the curve to the right of the "saturated vapor" line indicate that the refrigerant is a superheated vapor.

R-410A is a *blend*. But the glide for this blend is only 0.3°F, so the condensing and evaporating temperatures for R-410A are considered equal at any given pressure. You will be able to use the pressure-enthalpy diagrams for R-22 and R-410A in the same manner.

The "critical point" of a refrigerant is represented at the peak of the curve. This is the last pressure/temperature point at which the refrigerant may be condensed into a liquid. When the refrigerant temperature exceeds this value, no pressure above the critical pressure can condense the vapor into a liquid. In the case of R-22, the critical temperature is 205.24°F (some pressure-enthalpy diagrams show the critical temperature as 204.81°F) and the critical pressure is 722.39 psia. The critical point for R-410A occurs at 161.83°F and 714.5 psia.

Notice that a reference point for enthalpy is established on the graph for R-22 and on the graph for R-410A. On both graphs, the enthalpy is considered to be 0 Btu/lb at a temperature of –40°F. Heat energy removed from the refrigerant below this "base point" results in negative enthalpy. Heat energy added to the refrigerant above this point results in positive enthalpy.

Temperature (°F)	Pressure (psig) R-22	Pressure (psig) R-410A	Temperature (°F)	Pressure (psig) R-22	Pressure (psig) R-410A
–40	0.6	10.8	55	92.6	156.0
–35	2.6	14.0	60	101.6	170.0
–30	4.9	17.8	65	111.3	185.0
–25	7.5	21.9	70	121.4	200.0
–20	10.2	26.3	75	132.2	217.0
–15	13.2	31.1	80	143.7	235.0
–10	16.5	36.4	85	155.7	254.0
–5	20.1	42.6	90	168.4	274.0
0	24.0	48.2	95	181.8	295.0
5	28.3	54.9	100	196.0	317.0
10	32.8	62.1	105	210.8	340.0
15	37.8	69.9	110	226.4	364.0
20	43.1	78.2	115	242.8	390.0
25	48.8	87.2	120	260.0	417.0
30	54.9	96.8	125	278.1	445.0
35	61.5	107.0	130	297.0	475.0
40	**68.5**	**118.0**	135	316.7	506.0
45	76.1	129.5	140	337.4	538.0
50	84.1	142.0	145	359.1	573.0

Table 1. Temperature-pressure chart

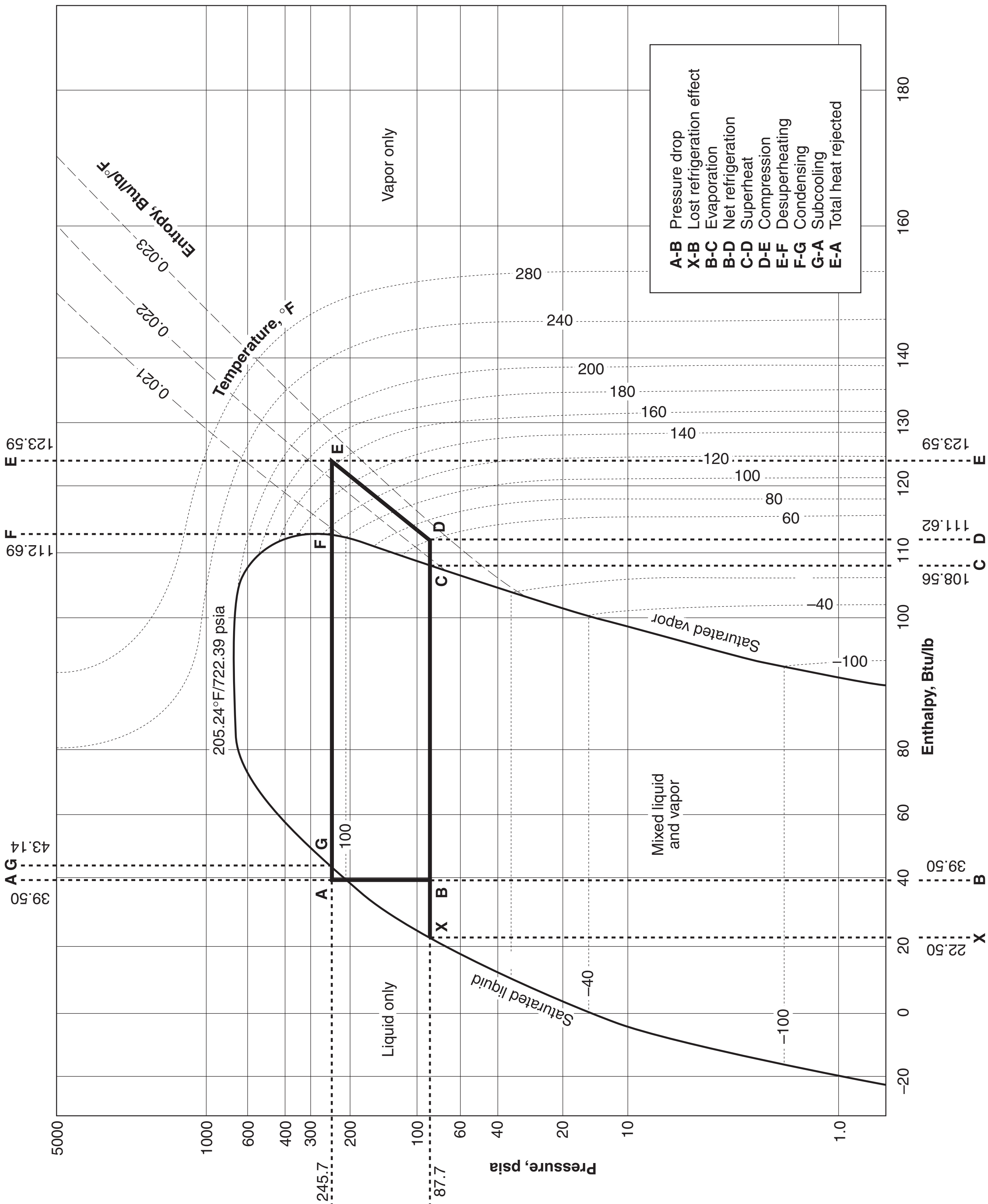

Figure 3. Pressure-enthalpy diagram for R-22

The four-sided polygon drawn in heavy black lines represents the refrigeration cycle. The right side of this polygon is sloped and indicates compression within the compressor. The upper horizontal line is representative of the condensing pressure/temperature and the lower horizontal line represents the evaporating pressure/temperature. These lines do not slope, because evaporation and condensation occur at a constant pressure and temperature. The vertical line forming the left boundary of the polygon indicates the pressure drop that occurs as the refrigerant is metered into the evaporator.

Because the purpose of an air conditioning system is to control temperature and provide comfort within a structure, the metering device that supplies refrigerant to the evaporator is the logical place to begin an explanation of the operating cycle.

Let's begin our study of the cooling cycle with the unit that uses R-22. The upper horizontal line of the cooling cycle plotted in Figure 3 indicates that the compressor discharge pressure in this example is approximately 245.7 psia. The saturation temperature of R-22 at 245.7 psia is 111.4°F. Assume that a measurement of the liquid line shows the actual liquid temperature at this point to be 100°F. Subtracting the measured temperature from the saturation temperature indicates that the liquid is subcooled 11.4°F (111.4 – 100). This is reflected by the location of plot point "A" outside and to the left of the curve (directly above the 100°F point on the curve). Subcooling is necessary to ensure that a solid column of liquid refrigerant reaches the metering device. If the liquid seal at the metering device is lost, flash gas develops. Gas or saturated vapor is unstable and restricts flow to the expansion device.

Note that vertical line A-B descends at a right angle to the horizontal axis and passes through the 100°F point on the curve, because the measured temperature of the liquid leaving the condenser is 100°F. Vertical line A-B indicates that a pressure drop occurs as the refrigerant passes through the metering device into the evaporator coil. In this example, the pressure drops from 245.7 psia to 87.7 psia. Subjected to a lower pressure, the refrigerant begins to boil off as it passes through the metering device. The evaporating temperature at this new, lower pressure is only 43°F, so the higher temperature of the indoor air is more than enough to cause boiling. As air moves over the evaporator, heat is removed and the air returns to the conditioned space at a lower temperature. Heat from the space is moved to the coil by the blower system to provide heat that boils the refrigerant.

Horizontal line segment B-C represents the vaporization of the refrigerant in the evaporator. At point C, the refrigerant is completely vaporized. Vaporization takes place at a constant temperature and pressure—therefore, point C is located on the diagram by following the line of constant pressure or constant temperature from point B to the point of intersection with the saturated vapor curve. The refrigerant at point C exists as a saturated vapor at the evaporator's vaporizing temperature and pressure.

Most of the cooling work done in the evaporator is transferred as *latent* heat and is represented by line segment B-C. Direct expansion evaporators are operated in a slightly starved condition to ensure that all liquid refrigerant boils off in the evaporator and that only dry vapor enters the compressor. This means that the vapor will be *superheated* above its boiling point before it is returned to the compressor. Therefore, a small amount of cooling work will be transferred as *sensible* heat when the vapor is superheated. This work is represented by line segment C-D. The horizontal line is extended to the right of the saturated vapor curve to intersect the 60°F vapor temperature curve, because the measured temperature of the suction line in this example is 60°F.

The enthalpy of the refrigerant increases as the refrigerant flows through the evaporator and absorbs heat from the space during process B-D. The quantity of heat absorbed by the refrigerant (known as the *net refrigeration effect*) is the difference between its enthalpy at points B and D. In this example, the enthalpy of the refrigerant at point B is 39.50 Btu/lb. At point C, it is equal to 108.56 Btu/lb. This means that 69.06 Btu/lb (108.56 – 39.50) of cooling work is transferred as *latent* heat. The enthalpy of the refrigerant at point D is 111.62 Btu/lb. The difference in enthalpy values between points D and C indicates that 3.06 Btu/lb (111.62 – 108.56) of cooling work is transferred as *sensible* heat. Remember, this work is called "sensible" heat because the refrigerant vapor changes temperature when the heat is transferred.

The *net refrigeration effect* is the difference in enthalpy between points B and D, which is equal to 72.12 Btu/lb (111.62 – 39.50). The term "net" is used because the *total* cooling capacity of R-22 liquid refrigerant at a pressure equal to 87.7 psia is not utilized. A portion of the refrigerant's cooling capacity is "sacrificed" as it passes through the metering device and begins to flash.

In this example, the liquid refrigerant that reaches the metering device is at temperature of 100°F—a temperature that is much too warm to cool the evaporator coil. The first flash gas that appears as the refrigerant is metered is used to cool the remaining 100°F liquid down to 43°F. Point X in Figure 3 indicates that the enthalpy of the saturated liquid at 87.7 psia is equal to 22.50 Btu/lb. Line segment X-C indicates the total latent heat of vaporization of 1 lb of R-22. Line segment X-B represents the loss of refrigeration effect. The loss, in this case, is 17 Btu/lb (39.50 – 22.50).

A typical R-22 system circulates approximately 3 lb of refrigerant per ton per minute, depending on the operating conditions and the system used. According to the manufacturer's product performance data, the total cooling capacity for the system used in this example, when operating at the conditions specified earlier, is approximately 32,200 Btu per hour (Btuh), or 2.68 tons:

$$\frac{32{,}200 \text{ Btuh}}{12{,}000 \text{ Btuh/ton}} = 2.68 \text{ tons}$$

The heat gain in the evaporator is 72.12 Btu per pound (Btu/lb). Using these values, you can calculate the refrigerant flow rate per hour as follows:

$$\frac{32{,}200 \text{ Btuh}}{72.12 \text{ Btu/lb}} = 446.48 \text{ lb/hr}$$

This corresponds to a refrigerant flow rate of approximately 2.78 lb per ton per minute for the system in our example:

$$\frac{446.48 \text{ lb/hr}}{2.68 \text{ ton/hr}} = 166.6 \text{ lb/ton/hr}$$

$$\frac{166.6 \text{ lb/ton/hr}}{60 \text{ min/hr}} = 2.78 \text{ lb/ton/min}$$

Continuing with the diagram, now follow the diagonal line upward to the right from point D to point E. The diagonal line runs roughly parallel to the lines of constant entropy and intersects the horizontal, head pressure line at a vapor temperature of approximately 160°F (point E). This intersection point indicates that the hot vapor temperature will be approximately 160°F. This is a theoretical value. The actual hot vapor temperature will be slightly higher due to compressor inefficiencies.

Again, note the gain in enthalpy from 111.62 Btu/lb at point D to 123.59 Btu/lb at point E. This heat gain is due to the heat of compression. If you subtract 111.62 Btu/lb from 123.59 Btu/lb, you will find that each pound of refrigerant gained 11.97 Btu of heat during the compression process, for a total gain of 5,344.4 Btuh:

$$11.97 \text{ Btu/lb} \times 446.48 \text{ lb/hr} = 5{,}344.4 \text{ Btuh}$$

Line E-A represents the condenser performance. Note that the enthalpy decreases from 123.59 Btu/lb at point E to 39.50 Btu/lb at point A by passing 90°F air over the coil. Thus, the amount of heat removed equals 84.09 Btu/lb (123.59 – 39.50), resulting in 37,544.5 Btuh of total work done:

$$84.09 \text{ Btu/lb} \times 446.48 \text{ lb/hr} = 37{,}544.5 \text{ Btuh}$$

The work done from point E to point F represents *desuperheating* of the hot vapor. Work done in this process is transferred as sensible heat, because the vapor is cooled to the condensing temperature. Typically, only 5% of the coil volume is needed to desuperheat the hot vapor. If you subtract the enthalpy of the refrigerant at point F from that at point E, you will find that the heat removed is 10.9 Btu/lb (123.59 – 112.69). The work done to desuperheat the hot vapor in this example is 4,866.6 Btuh:

$$10.9 \text{ Btu/lb} \times 446.48 \text{ lb/hr} = 4{,}866.6 \text{ Btuh}$$

Line segment F-G represents the work done to condense the refrigerant back into a liquid. Work done in this process is transferred as latent heat, because the refrigerant changes state at a constant temperature and pressure. Typically, approximately 85% of the coil volume is used to condense the refrigerant. Note that the enthalpy is changed from 112.69 Btu/lb at point F to 43.14 Btu/lb at point G, for a change equal to 69.55 Btu/lb. The work done to condense the refrigerant in this example is 31,052.7 Btuh:

$$69.55 \text{ Btu/lb} \times 446.48 \text{ lb/hr} = 31{,}052.7 \text{ Btuh}$$

The work done from point G to point A is transferred as sensible heat, because the liquid refrigerant is cooled from the condensing temperature of 111.4°F to 100°F. Typically, 10% of the coil is used to subcool the liquid refrigerant. To calculate the heat transfer required to subcool the liquid, first subtract the enthalpy at point A from the enthalpy at point G, for a change equal to 3.64 Btu/lb (43.14 – 39.50). The total heat transfer needed to subcool the liquid refrigerant in this example is 1,625.2 Btuh:

$$3.64 \text{ Btu/lb} \times 446.48 \text{ lb/hr} = 1{,}625.2 \text{ Btuh}$$

As with the evaporator coil, the majority of the work done in the condensing coil is transferred as latent heat.

The condensing coil must reject the heat gained in the evaporator plus the heat gained in the compressor. Keep in mind, too, that the condenser must reject this heat when weather conditions are extreme. This is why the condenser must be larger than the evaporator. Far more surface area is required to reject the heat when weather conditions are extreme.

In our example, the condenser rejected 37,545 Btuh of heat energy while the cooling work done at the evaporator was approximately 32,200 Btuh. As a general guide, you may estimate that the condenser rejects 15 to 25% more heat than is absorbed in the evaporator in the cooling mode, depending on the system design and weather conditions. Thus, for each 12,000 Btuh (1 ton) of cooling done in the evaporator, the condenser must reject as much as 15,000 Btuh.

This cycle is repeated continuously, as long as the room thermostat is calling for cooling. Figure 4 illustrates those points in the system where the pressure and temperature readings described above were taken. Remember that absolute pressure values (psia) are derived by adding 14.7 psi to the gauge readings.

Concerns about the effect of fluorocarbon refrigerants on the ozone layer have prompted legislation that affects the HVAC/R industry. When released to the atmosphere, refrigerants that contain chlorine, such as R-22, deplete the ozone in the stratosphere. For this reason, R-22 is being replaced with chlorine-free refrigerants. The most popular replacement choice among manufacturers of residential equipment is R-410A. The refrigeration capacity of R-410A is 45% greater than that of R-22. Consequently, the system charge for R-410A will be less than that of a comparable R-22 system. The refrigerant charge of the R-22 system with a 15-ft line set used in this example is 7 lb 4 oz, while the refrigerant charge of the R-410A system with a 15-ft line set is 6 lb 8 oz. The operating pressures of systems that use R-410A are much higher than systems that use R-22. Let's

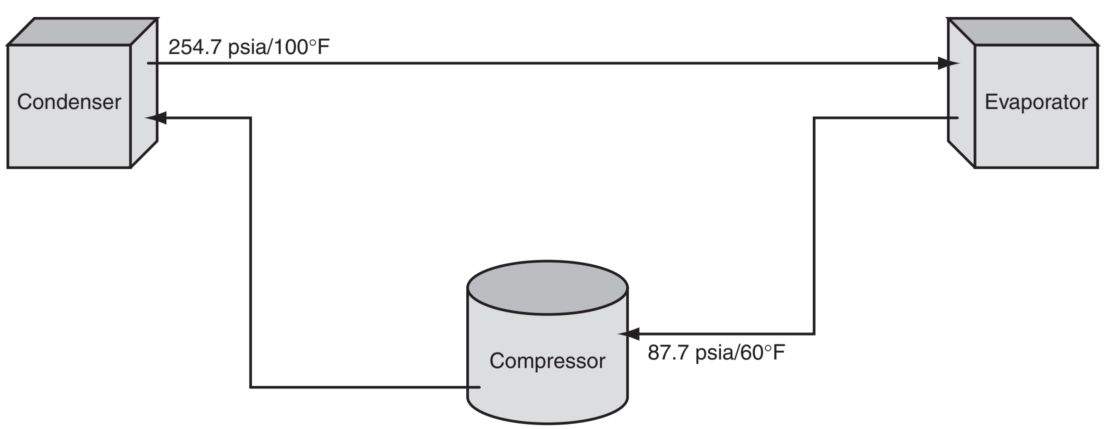

Figure 4. Measurement locations *(R-22 readings)*

take a look at the pressure-enthalpy diagram for a cooling cycle using R-410A (see Figure 5).

The refrigeration cycle is represented by a four-sided polygon, in the same manner as it was for R-22. Horizontal lines are drawn to represent the head pressure and the back pressure of an operating A/C unit. The head pressure line forms the upper boundary of the polygon, and the back pressure line forms the lower boundary.

As with the earlier R-22 system, this unit is operating when the ambient temperature is 90°F and the wet-bulb temperature of the air entering the evaporator is 63°F. The head pressure for this system is 375.7 psia, which has a corresponding condensing temperature equal to 109.2°F. The back pressure is 141.2 psia, and the evaporating temperature at that pressure is 43.5°F.

As before, begin at that point where the liquid refrigerant enters the metering device (point A in Figure 5). The head pressure line is extended to the left of the saturated liquid curve to a point directly above 100°F on the curve. This is done because the measured temperature of the liquid leaving the condensing coil is 100°F in this example. A vertical line is drawn downward from point A to connect the head pressure and back pressure lines. This vertical line connects to the back pressure line at point B, forming the left boundary of the polygon. Line segment A-B represents the drop in pressure from 375.7 psia in the liquid line to 141.2 psia in the evaporator. The enthalpy of the refrigerant at this point in the cycle is 50.83 Btu/lb.

Heat supplied by the room air circulated over the coil is absorbed by the refrigerant as it boils off in the evaporator. The majority of the work done while cooling the air is transferred as latent heat, represented by line segment B-C. The enthalpy value at point C is 120.41 Btu/lb. Therefore, the latent heat transferred is 69.58 Btu/lb (120.41 – 50.83).

To protect the compressor from liquid slugging, the evaporator is operated in a slightly starved condition by allowing the refrigerant vapor to provide a small cooling effect in the evaporator. Superheating the suction vapor in the evaporator ensures that the vapor is dry and suitable for return to the compressor. The cooling work represented by line segment C-D is transferred as sensible heat, because it causes a change in the vapor temperature. The enthalpy at point D is 125.20 Btu/lb, which means that the sensible heat transferred while superheating the suction vapor is 4.79 Btu/lb (125.20 – 120.41).

The total refrigeration effect is represented by line segment B-D. It totals 74.37 Btu/lb in this example (125.20 – 50.83).

According to the manufacturer's product performance data, the total cooling capacity for this R-410A system operating at the conditions cited earlier is approximately 34,200 Btuh, or 2.85 tons. You can estimate the refrigerant flow rate for this unit as follows:

$$\frac{34{,}200\ \text{Btuh}}{74.37\ \text{Btu/lb}} = 459.86\ \text{lb/hr}$$

The refrigerant flow rate per minute, therefore, is:

$$\frac{459.86\ \text{lb/hr}}{60\ \text{min/hr}} = 7.66\ \text{lb/min}$$

Refrigerant flow rate per ton per minute is estimated as follows:

$$\frac{7.66\ \text{lb/min}}{2.85\ \text{tons}} = 2.69\ \text{lb/min/ton}$$

Line segment X-B represents the lost refrigeration effect as the liquid is metered into the evaporator. The enthalpy of the refrigerant at point X is 28.79 Btu/lb and the enthalpy at point B is 50.83 Btu/lb. Therefore, the amount of refrigeration effect lost is 22.04 Btu/lb (50.83 – 28.79).

Compression is represented by line segment D-E, which runs roughly parallel to the lines of constant entropy. Enthalpy at point D is 125.20 Btu/lb, while at point E it is 140.75 Btu/lb. Thus, the heat of compression in this example is 15.55 Btu/lb (140.75 – 125.20). The vapor temperature is raised from 60°F to approximately 158°F during compression. Remember, the calculated 158°F hot vapor temperature is a theoretical value. It actually will be slightly higher due to compressor inefficiencies. The heat of compression equals 7,150.8 Btuh, calculated as follows:

$$459.86\ \text{lb/hr} \times 15.55\ \text{Btu/lb} = 7{,}150.8\ \text{Btuh}$$

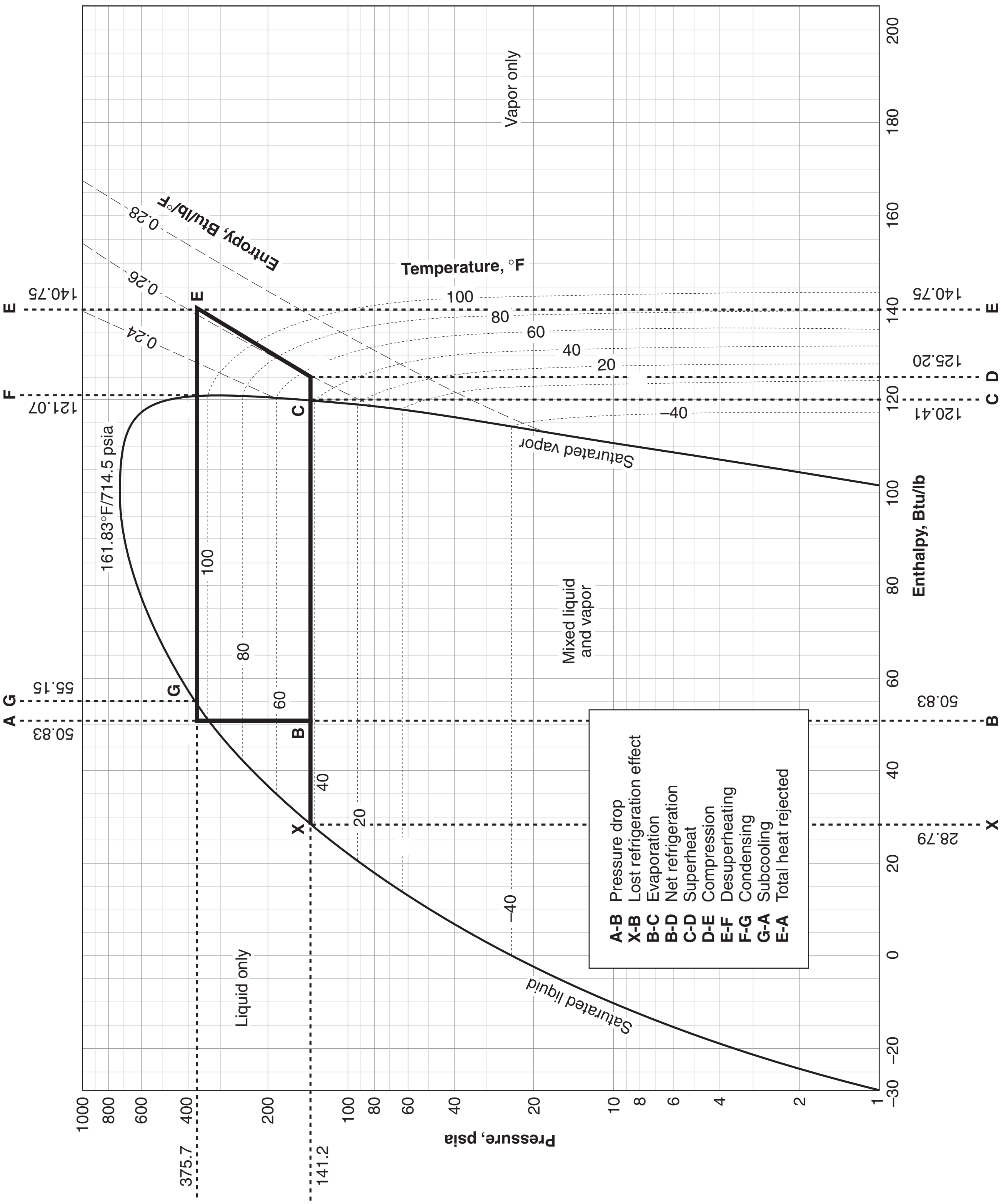

Figure 5. Pressure-enthalpy diagram for R-410A

The condenser first desuperheats the hot vapor by transferring heat energy from the vapor to the air circulating over the coil. The work done while cooling the hot vapor to the saturation temperature of 109.2°F is transferred as sensible heat. It is equal to 19.68 Btu/lb. This can be calculated by subtracting the enthalpy at point F (121.07 Btu/lb) from the enthalpy at point E (140.75 Btu/lb). The amount of heat removed to desuperheat the hot vapor is 9,050 Btuh, calculated as follows:

$$459.86 \text{ lb/hr} \times 19.68 \text{ Btu/lb} = 9{,}050 \text{ Btuh}$$

Next, the condensing coil condenses the refrigerant by transferring even more heat energy into the circulating air. This process is represented by line segment F-G on the diagram. Condensation takes place at a constant pressure and temperature—therefore, the work done is transferred as latent heat. In this example, the enthalpy is 121.07 Btu/lb at point F and 55.15 Btu/lb at point G. This means that the change in enthalpy is 65.92 Btu/lb (121.07 – 55.15). The heat removed to condense the refrigerant is equal to 30,313.97 Btuh, calculated as follows:

$$459.86 \text{ lb/hr} \times 65.92 \text{ Btu/lb} = 30{,}313.97 \text{ Btuh}$$

Finally, the condenser subcools the liquid refrigerant to ensure that a solid column of liquid reaches the metering device. The subcooling process is depicted by line segment G-A. The work done in this process is transferred as sensible heat, because the liquid refrigerant changes temperature as a result of the work done on it. The change in enthalpy is 4.32 Btu/lb during the subcooling process (55.15 – 50.83). The heat removed to subcool the liquid is 1,986.6 Btuh, calculated as follows:

$$459.86 \text{ lb/hr} \times 4.32 \text{ Btu/lb} = 1{,}986.6 \text{ Btuh}$$

The total work done in the condenser is represented by line segment E-A, and is equal to approximately 89.92 Btu/lb (140.75 – 50.83). This figure must total the sum of the work done to desuperheat, condense, and subcool the refrigerant, which is 41,351 Btuh (9,050 + 30,314 + 1,987). The work represented by line E-A also equals approximately 41,351 Btuh, and is calculated as follows:

$$459.86 \text{ lb/hr} \times 89.92 \text{ Btu/lb} = 41{,}350.6 \text{ Btuh}$$

The Law of the Conservation of Energy teaches us that the work done in the condenser must equal the work done in the evaporator plus the work done during compression. (This is also true for the R-22 system discussed earlier.) The following comparison shows the relationship:

line segment B-D (net refrigeration effect)
= 74.37 Btu/lb (125.20 – 50.83)

plus

line segment D-E (work of compression)
= 15.55 Btu/lb (140.75 – 125.20)

equals

89.92 Btu/lb (total work)

which is also equal to

line segment E-A (work done in condenser)
= 89.92 Btu/lb (140.75 – 50.83)

In this example, the heat rejected by the condenser is approximately 21% more than the heat absorbed by the evaporator:

$$\frac{15.55 \text{ Btu/lb}}{74.37 \text{ Btu/lb}} \times 100 = 20.9\%$$

Notice that the performance of the R-22 A/C system is similar to that of the R-410A system. Each delivered the rated system capacity, even though the two systems operated at very different pressures.

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