The Pressure - Enthalpy Chart

By Dave Demma, Senior Application Engineer - Supermarket Refrigeration

How often have you heard the statement “it isn’t cooling?” Well it may seem a bit picky, but it is not entirely accurate to say the refrigeration system “cools”. If the system is operating properly, the refrigerated space should be “cooler” than its surroundings, but it is the result of a heat transfer process. Heat is transferred from the refrigerated space to the refrigerant, and ultimately from the refrigerant to the ambient (at the condenser). A lower temperature in the refrigerated space is the byproduct of this heat transfer process. Perhaps this is a minor shift in thinking, but in viewing the refrigeration system for what it is…a heat transfer process…a more fundamental approach for diagnosis may be obtained.

In an effort to gain a better understanding of the various heat transfer processes occurring in a refrigeration system, the pressure-enthalpy chart can be of great use. Additionally, once understood, “the chart” can be a tremendous benefit in analyzing the relative health of a refrigeration system. Let’s follow the refrigerant on a quick journey through a refrigeration system to see what it experiences, and plot it on “the chart” as we go. Before we start, a few technical definitions are in order:

Refrigeration - The achievement of a temperature below that of the immediate surroundings.

Latent Heat of Fusion - The quantity of heat (Btu/lb) required to change 1 lb. of material from the solid phase into the liquid phase.

Latent Heat of Vaporization - The quantity of heat (Btu/lb) required to change 1 lb. of material from the liquid phase into the vapor phase.

Sensible Heat - Heat that is absorbed/rejected by a material, resulting in a change of temperature.

Latent Heat - Heat that is absorbed/rejected by a material resulting in a change of physical state (occurring at constant temperature).

Saturation Temperature - That temperature at which a liquid starts to boil (or vapor starts to condense). The saturation temperature (boiling temperature) is constant at a given pressure,* and increases as the pressure increases. A liquid cannot be raised above its saturation temperature. Whenever the refrigerant is present in two states (liquid and vapor) the refrigerant mixture will be at the saturation temperature.

Superheat - At a given pressure, the difference between a vapor’s temperature and its saturation temperature.

Subcooling - At a given pressure, the difference between a liquid’s temperature and its saturation temperature.

* Except for zeotrope refrigerants

Ton of Refrigeration - The amount of cooling required to change (freeze) 1 ton of water at 32ºF into ice at 32ºF, in a 24 hour period.

Btu - British Thermal Unit: The amount of heat required to raise 1 lb. of water 1ºF.

1 Ton - 12,000 Btu/hr

Fig. 1 illustrates some of these definitions, using water as the medium experiencing a heat transfer process. This graph plots the water temperature vs. the enthalpy of the water (heat content in Btu/lb). We all know that water boils at 212ºF (atmospheric pressure at sea level). By definition, water at atmospheric pressure, at a temperature lower than 212ºF, is subcooled. So, we start with subcooled water at 42ºF, and begin transferring heat to it. Assuming we are working with 1 lb. of water, for every Btu added, a corresponding temperature increase of 1ºF will be achieved (the definition for one Btu). It we continue to add heat, eventually the water’s temperature will increase to 212ºF (the saturation temperature at atmospheric pressure). At this point, the water begins to change states from a liquid to a vapor (boil). As noted on the graph, the water will experience no further temperature increase…for a given pressure, the saturation (boiling) temperature is the highest temperature a liquid can ever achieve. Increasing the amount of heat transferred to the water simply increases the rate at which the water boils. If the temperature of the vapor were to be measured, we’d find it to be 212ºF (saturated vapor). Once the vapor has separated from the liquid, additional heat transferred to it will result in a temperature increase. By definition, the vapor at 232ºF (20º above the saturation temperature), is superheated.

It is interesting to note that while it takes only 1 Btu to raise 1 lb. of water 1ºF, it takes almost 1000 times that amount (966.6 Btu) for the 1 lb. mass of water to change states from liquid to vapor. A boiling liquid will always absorb more heat than a vapor experiencing a temperature increase (per unit of mass). Understanding this principle explains why the evaporator in a refrigeration system should always be nearly filled with liquid refrigerant. Otherwise, its full potential as a heat transfer device will not be utilized.

The pressure-enthalpy chart, as shown in Fig. 2, displays all the pertinent properties for a given refrigerant (in this example R22). The bubble to the left is the portion of the diagram where the refrigerant is in the saturated condition. The blue line on the left of the bubble represents the 100% saturated liquid line, the thin dashed line on the right represents the 100% saturated vapor line, and anywhere inside the bubble represents the refrigerant as a mixture of saturated liquid and saturated vapor.

To the left of the saturated liquid line is the area where the refrigerant can exist at a temperature lower than the saturated condition; subcooled liquid. To the right of the saturated vapor line is the area where the refrigerant can exist at a temperature higher than the saturated condition; superheated vapor. The critical point is the highest temperature that the refrigerant can experience, and remain in the liquid form. If the temperature exceeds the critical point, regardless of pressure, the refrigerant can only exist in the vapor state.

All of the relevant properties are shown in Fig. 2:

Pressure - The vertical axis of the chart, in psia (see pink line). To obtain gauge pressure, subtract atmospheric pressure.

Enthalpy - The horizontal axis of the chart shows the heat content of the refrigerant in Btu/lb.

Temperature - Constant temperature lines generally run in a vertical direction in the superheated vapor & sub-cooled liquid portion of the chart. In the saturated bubble, the constant temperature line is along the horizontal, illustrating that the saturation temperature is constant at a given pressure (see black line).

Specific Volume - Constant volume lines extend from the red line saturated vapor line out into the superheated vapor-portion of the chart at a slight angle from the horizontal axis. Specific volume is expressed in cu.ft/lb. (see orange line).

Entropy - Entropy is the mathematical relationship between heat and temperature, and relates to the availability of energy. These lines extend at an angle from the saturated vapor line. Their presence on the chart is relevant in that vapor compression (in the ideal cycle) occurs at constant entropy (see dark blue line).

Quality - Lines of constant quality appear vertically, and only within the saturation bubble. The refrigerant within the bubble is a mixture of liquid and vapor at saturation, and the quality is the percentage of the mixture which is in the vapor state (see green line).

The ultimate goal of the refrigeration system is to get the refrigerant into a condition where it can be useful as a medium to transfer heat from the refrigerated space. If the system design requires a -10ºF space temperature, you would expect the refrigerant temperature in the evaporator to be somewhat lower…say -20ºF (a 10º temperature difference). This allows the relatively warmer air (something above the design of -10ºF) to be blown across the evaporator, flowing relatively cooler refrigerant (-20ºF). The result is the transfer of heat from the warmer air to the cooler refrigerant.

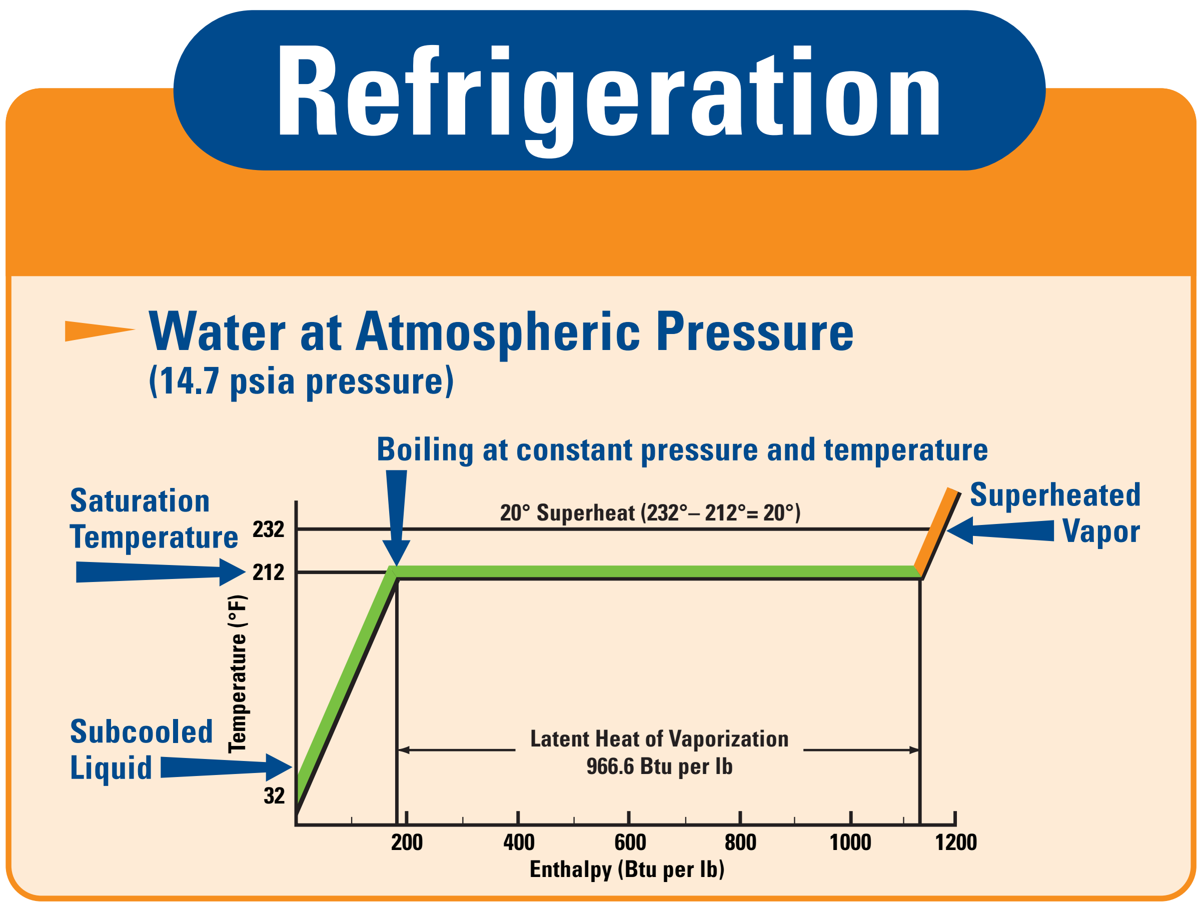

Fig. 1 – Water undergoing a change of state.

The purpose of the compressor is to take a low pressure vapor and compress it into a high pressure vapor. This occurs (in theory) at a constant entropy. In an ideal cycle, the refrigerant vapor would enter the compressor as a saturated vapor…no superheat.

The Ideal Cycle

If the operating temperatures and pressures are known, the refrigeration system can be plotted on the P-H diagram. Let's assume our system is operating at a -20ºF evaporator and 100ºF condensing.

The saturated vapor entering the compressor suction would be at -20ºF, illustrated by point #1 in Fig. 3. The vapor is compressed, following the constant entropy line to the pressure corresponding to 100ºF, or 210.7 psia (point #2). The refrigerant vapor experiences a sensible heat gain during the mechanical compression process, resulting in a superheated vapor. This is illustrated by the location of point #2, to the right of the saturated vapor line. In the ideal system, which does not consider pressure loss in the valves, refrigerant tubing, etc., point #2 represents the outlet of the compressor/inlet of the condenser.

In the ideal cycle, the condenser serves as a two-fold component. Before any condensation occurs, the high pressure vapor must first be brought to a saturated condition (de-superheated). Enough heat must be transferred from the refrigerant to lower its temperature from 180ºF to the saturation temperature of 100ºF (point #2A on the chart). At this point, condensation can begin. As heat continues to be transferred from the refrigerant vapor to the air (or water, if a water cooled condenser is used), the quality of the refrigerant (% of the refrigerant in the vapor state) will continue to decrease, until the refrigerant has been completely condensed. In the ideal system, this occurs at the outlet of the condenser (point #3 on the chart). In the real world, some subcooling would be expected at the condenser outlet. Subcooled liquid provides insurance against liquid flashing as the refrigerant experiences pressure losses in the tubing and components.

The refrigerant is in the liquid state now, and at a high pressure and temperature. It must undergo one more change before it becomes a useful heat transfer medium; a reduction in temperature. This is accomplished by reducing the pressure. You can count on the refrigerant's pressure-temperature relationship to be an infallible law. If the pressure of a saturated liquid is reduced, the law governing its existence requires it to assume the saturation temperature at the new pressure.

So, in order to reduce the temperature, the pressure has to be reduced, and some sort of restriction is required for this to occur. It would be preferable if the restriction could regulate

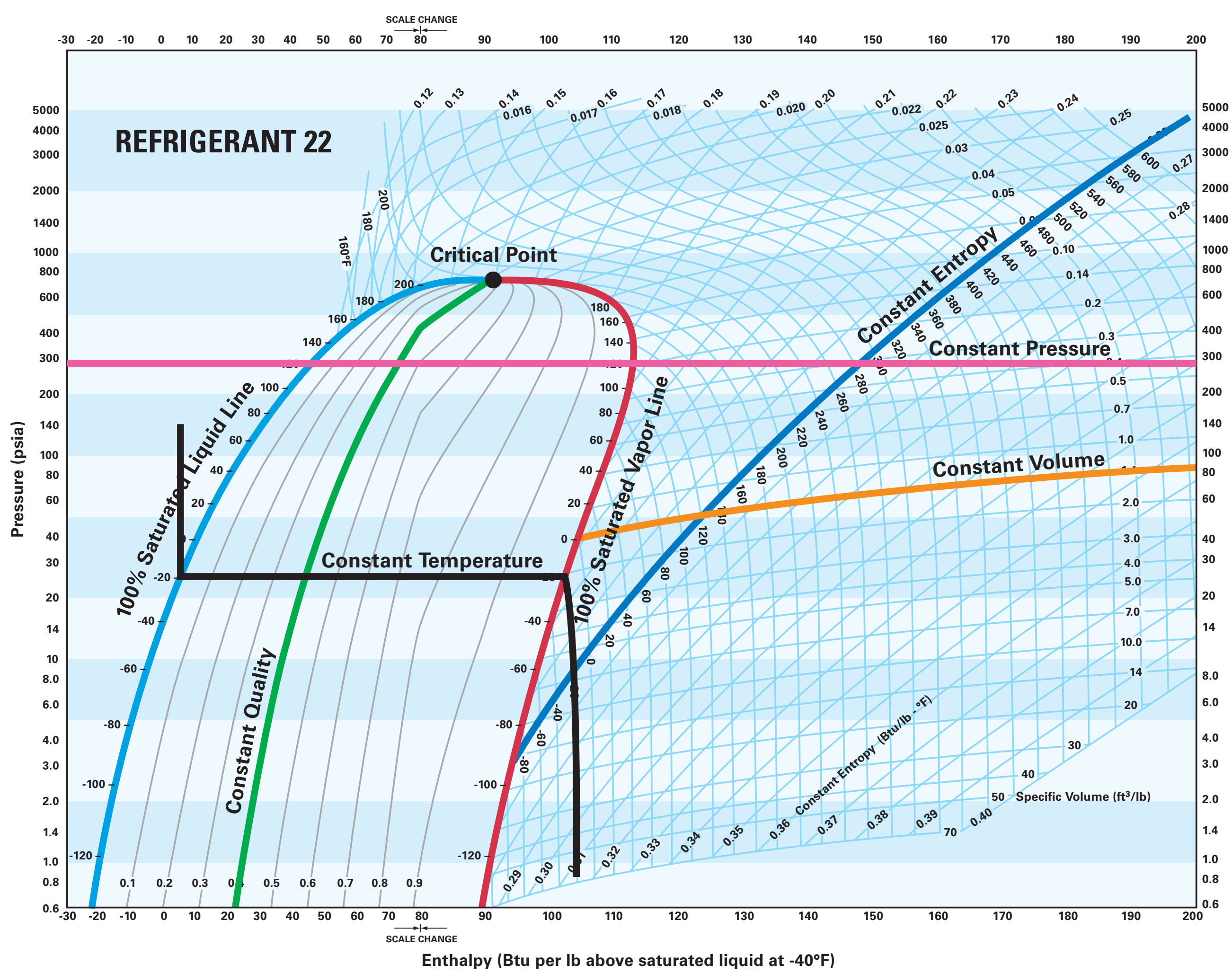

Fig. 2 – R22 Pressure-Enthalpy Diagram

itself as the system load demands change. This is exactly what the thermostatic expansion valve does; it is an adjustable restriction which causes a reduction in liquid refrigerant pressure, yet will modulate in an effort to maintain constant superheat at the evaporator outlet. The TEV is a superheat control, and will not maintain a constant evaporator pressure. It only provides the restriction necessary to reduce the pressure to some level, which will be determined by compressor size, TEV size, load demand, and system conditions. If a constant evaporator temperature is required, it can be achieved very simply by maintaining the pressure corresponding to the saturation temperature required. This is accomplished by adding an evaporator pressure regulating valve to the system.

Our ideal cycle has experienced a pressure drop in the TEV, and for the purpose of discussion we are at a constant 24.9 psia in the evaporator. This is a saturation temperature of -20ºF. You will notice that the refrigerant quality at the TEV inlet was 0%, and has increased to 35% at the evaporator inlet. Subcooling or superheat cannot exist where there is a mixture of liquid and vapor. Therefore, any place in the system where the refrigerant exists in two states (the receiver, parts of the evaporator and condenser, the accumulator at times), it will be at the saturation temperature for its pressure. For example, R22 in a saturated state at 24.9 psia (10.2 psig) will always be at -20ºF.

Because the refrigerant must conform to its law of existence, when the 210.7 psia (100ºF) liquid experiences the reduction in pressure to 24.9 psia, it must drop to the new saturation temperature of -20ºF. Some of the liquid refrigerant is required to boil as a means of removing the heat necessary to achieve this lower temperature. Yet another heat transfer process, which yields a lower liquid temperature. The liquid that is sacrificed in the boiling process explains the increase in refrigerant quality. The greater the difference between the liquid temperature and evaporator temperature, the more liquid will have to be boiled to achieve the new saturation temperature. This results in an even higher refrigerant quality.

The final portion of the refrigerant's journey is as a mixture of saturated liquid and vapor, traveling though the evaporator tubing. Warm air is blown across the evaporator, where its heat content is transferred to the boiling refrigerant. This is a latent heat gain to the refrigerant, causing no temperature increase, while experiencing a change of state. In the ideal cycle, the last molecule of saturated liquid boils off at the evaporator outlet, which is connected to the compressor inlet. Hence, the vapor at the inlet of the compressor is saturated.

The cycle continues this way until the refrigerated space temperature is satisfied, and the equipment cycles off.

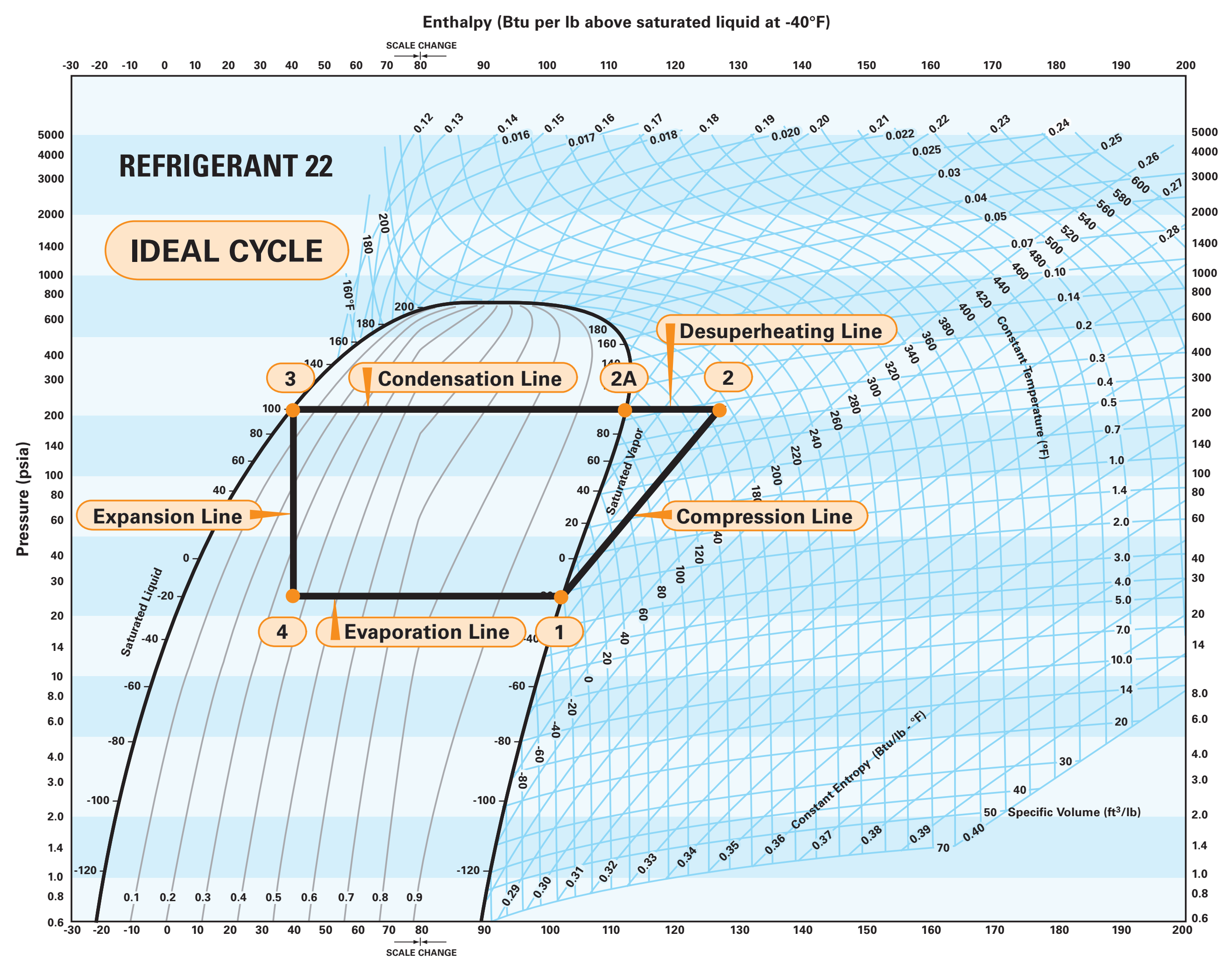

Fig.3 - Plotting the Ideal Cycle on the P-H Diagram

Once the system has been plotted, various data points can be read and used for the system design calculations. Admittedly, this information is not typically something the technician will need for servicing the refrigeration equipment. However, an understanding of how operating conditions affect system design, efficiency, energy consumption, and particularly compressor performance, should be of great worth to the technician.

Data Points and System Design Calculations:

Refrigeration Effect (RE): This is the total heat transfer, in Btu/lb, from the refrigerated space to the refrigerant. H1 minus H4 (H1 is the enthalpy of the refrigerant at point #1 in Fig. 3, and so forth).

Note: For the purpose of this discussion, H1 will be considered the point where the evaporation line intersects with the saturated vapor line. In the real world, the location of H1 would be to the right of the saturated vapor line, reflecting the superheated vapor at the evaporator outlet. With an expansion valve maintaining a typical amount of superheat (in the 4° - 6° range for low temperature applications), the heat transferred to the vapor is minimal (less than 1 Btu/lb). The heat transferred to the vapor between the suction piping outside of the refrigerated space and the compressor cylinder inlet is never considered as part of the refrigeration effect, as this heat was not transferred from the refrigerated space.

Heat of Compression (HOC): This is the amount of heat added to the refrigerant from the compression process (represented by H2 minus H1 on the chart).

Heat of Rejection (HOR): This is the amount of heat that has to be rejected at the condenser…the heat transferred to the refrigerant from the refrigerated space (RE), and the heat transferred to the refrigerant during compression (HOC). It is this value, plus some safety factor, from which the condenser selection is made (represented by H2 minus H3 on the chart).

Refrigerant Circulation Rate (RCR): The amount of refrigerant in lbs/min which must circulate in the system to meet the demands of the load.

$$\frac{\textbf{(200 Btu/min – ton)}}{\textbf{RE (Btu/lb)}}$$

Compressor Horsepower Required: The horsepower/ton requirement to meet the load demand. Contrary to popular belief, 1 horsepower and 1 ton are not necessarily synonymous.

$$\frac{\textbf{RCR (lb/min-ton) X HOC (Btu/lb) X 778 ft-lb/Btu}}{\textbf{33,000 ft-lb/min-hp}}$$

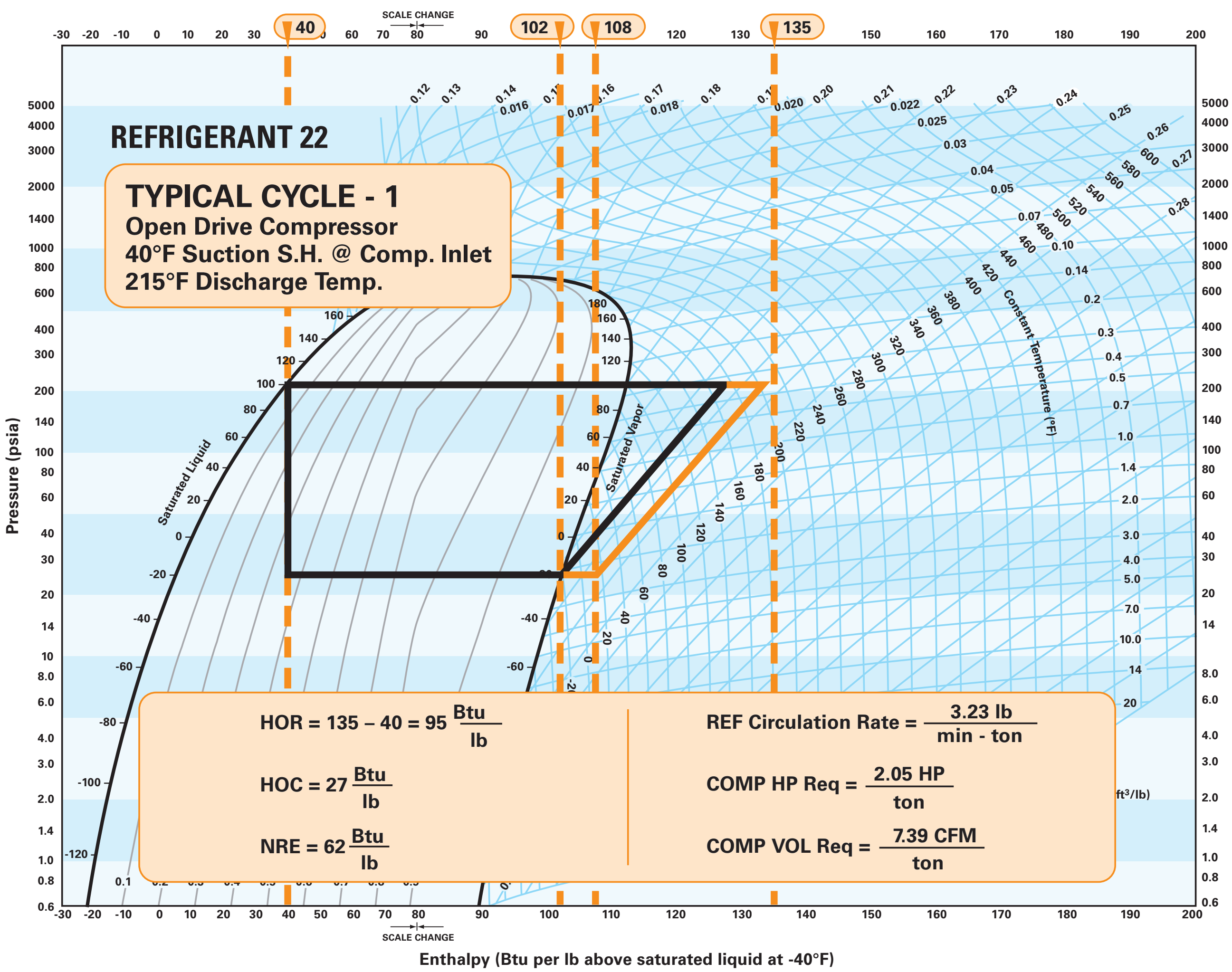

Fig. 4 – Typical Cycle #1

Compressor Volume Required: The compressor cylinder volume requirement needed to pump the RCR, in cu ft/ton. The vapor specific volume is read on the lines of constant volume.

RCR (lb/min-ton) X vapor specific volume (cu ft/lb)

Let's look at a few typical system scenarios, plot them on the P-H diagram, and then compare the performance measurements.

Typical Cycle #1 (Fig. 4): It is neither realistic nor safe to have a saturated vapor at the compressor inlet. Because liquid cannot be present with superheated vapor, **some amount of superheat at the compressor inlet becomes the margin necessary to insure the safety of the compressor**. Here we see the suction vapor superheated to +20ºF (40º superheat). This is the result of an expansion valve set to maintain some amount of superheat, plus the temperature increase the refrigerant vapor experiences in the suction line. The suction line connected to the compressor will have some accumulation of frost on it. This is the result of the 20°F pipe temperature and mositure in the air, NOT the result of floodback. Floodback is not possible when the vapor's temperature is 40° above the saturation temperature (40° superheat). Do not confuse frost buildup with floodback.

The suction superheat will insure that the compressor is protected from liquid flooding. The cost of this protection comes in the form of a larger compressor volume requirement. This is due to the warmer, thinner suction vapor; at +20ºF the specific volume of the vapor (measured in cu.ft./lb) is greater than it was at -20ºF. The compressor's cylinders are a measured volume...they never change. The density of the refrigerant may change, and this will affect the pounds per minute of refrigerant that the compressor will pump, however the volume pumped remains constant. So, because the suction vapor is less dense, we now require more compressor cylinder volume to pump the same mass flow.

While all the various data points and system design calculations are listed, they will appear in a comparative chart later.

Typical Cycle #1A (Fig. 5): Using an open drive compressor, with a 20ºF vapor temperature at the compressor inlet, we see the benefit of subcooling the liquid to 50ºF. Note the change in refrigerant quality at the TEV outlet. Instead of 65% liquid, we now have 80% liquid. Because the difference between liquid temperature and evaporator temperature has been reduced, there is less refrigerant flashing during the expansion process. As a result, the TEV and distributor nozzle (if used) can possibly be downsized.

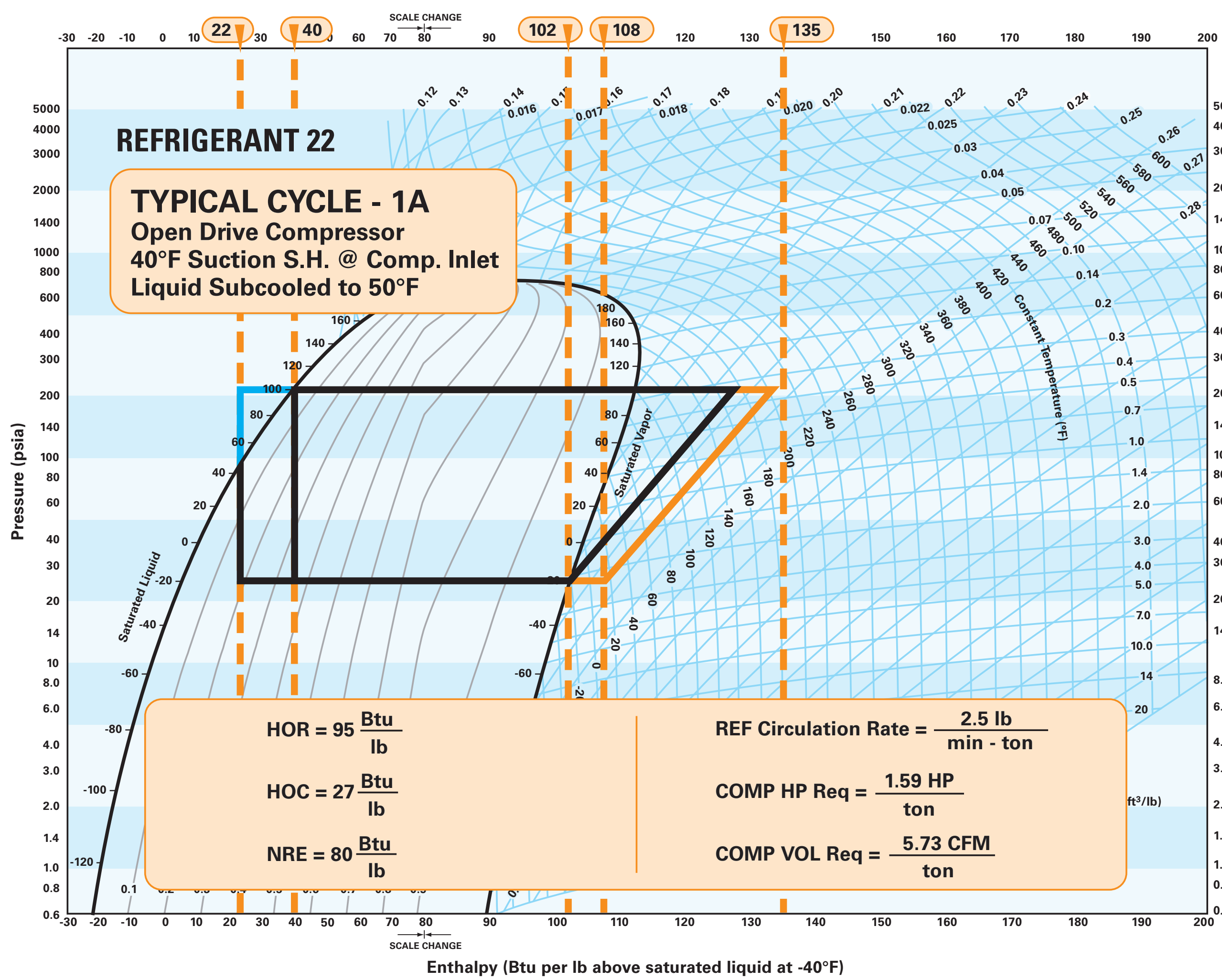

Fig. 5 – Typical Cycle #1A

The use of a liquid to suction heat exchanger will yield subcooled liquid, but at the expense of higher suction vapor temperatures. While this method of subcooling will insure vapor free liquid refrigerant at the TEV inlet, it has little effect on increasing system efficiency. The benefit realized resulting from subcooling will be offset by the higher suction vapor temperatures, and the volume requirement penalty they impose. In this supermarket example the liquid for the low temperature rack is subcooled using the medium temperature rack. There is no heat gain to the low temperature rack as a byproduct of the subcooled liquid. The refrigerant is subcooled on a rack that is operating between 2 to 2-1/2 horsepower per ton, and the benefit is being experienced on a rack that is operating near 5 horsepower per ton.

The comparative chart will show the benefits: reduced refrigerant circulation rate, reduced horsepower requirement, and reduced compressor volume requirement. If the subcooling can be accomplished very inexpensively, such as ambient subcooling with a receiver bypass, the energy savings can be enormous. For more details on this see Sporlan Form 90-134.

Typical Cycle #2 (Fig. 6): The advantage of using a hermetic compressor is the elimination of either belts or drive motor couplings which require precise alignment, and crankshaft seals. The disadvantage is that there is now an electric motor in the refrigerant circuit (at least on suction vapor cooled hermetic compressors). In addition to the guaranteed system contamination problem when a hermetic motor burns, you have the heat from the motor being transferred to the refrigerant vapor. An approximate 80ºF temperature increase can be expected between the vapor entering the compressor service valve, and the vapor entering the compressor cylinders. This brings the suction vapor temperature up to 100ºF, with a corresponding increase in discharge temperature…now approaching 300ºF. This is the upper limit at which most compressor manufacturers agree shouldn't be exceeded. The refrigerant vapor at 100ºF has a specific volume approximately 20% higher than at 20ºF. This translates into a compressor volume requirement which will be approximately 20% greater. The higher suction vapor temperature also results in a higher HOC, which raises the horsepower requirement.

Typical Cycle #3 (Fig. 7): Now, let's take a look at the real world. It is now the dead of summer – the most extreme condition for the equipment. When the system design and equipment selection was made, it was this summer weather that was used as the worst case condition of operation. Now, your company was so busy in the winter that the yearly preventative maintenance was not done. Or perhaps it's one of those customers who

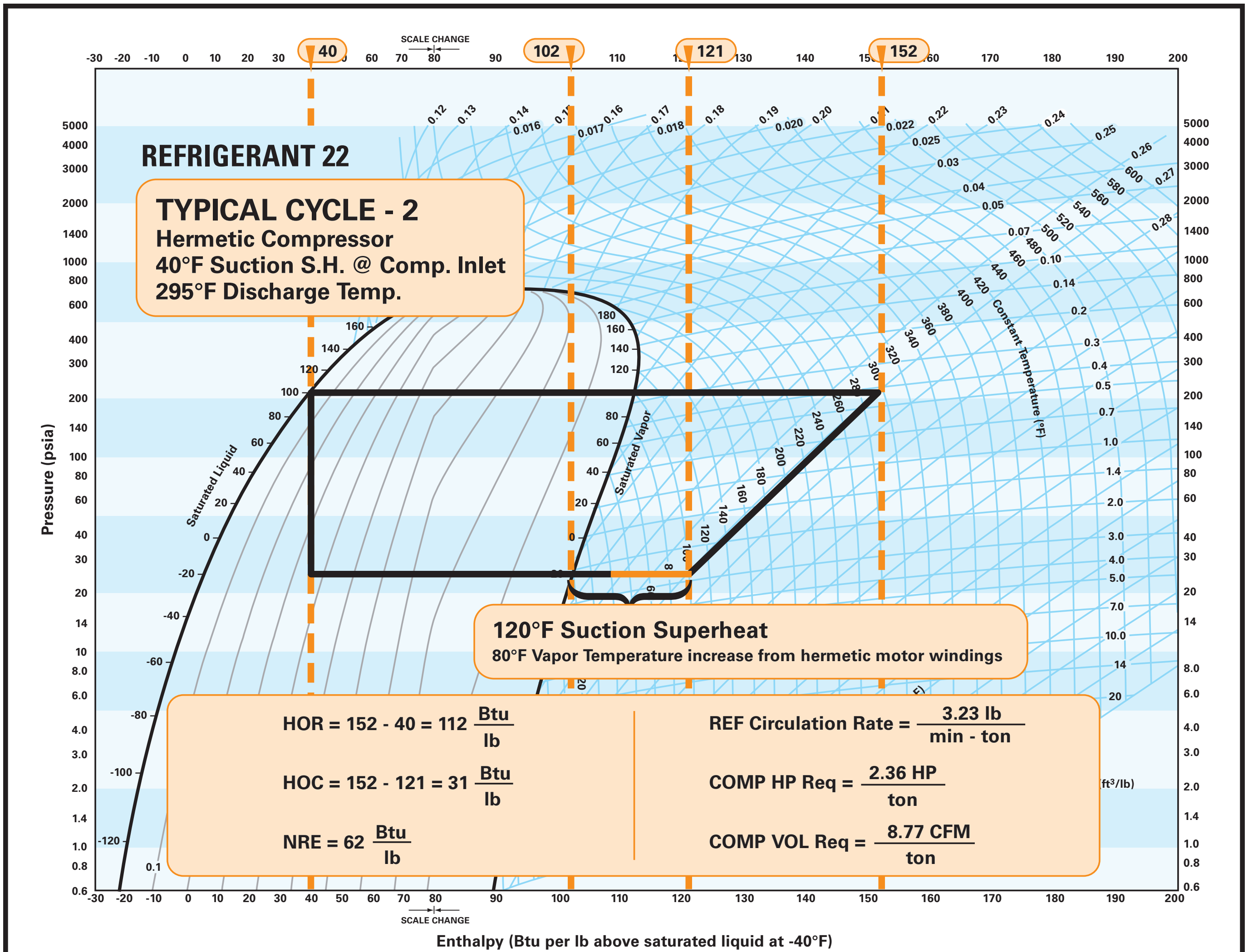

Fig. 6 – Typical Cycle #2

never wants to spend money on prevention…the one you tell "you can either pay me a little now, or you can pay me a lot more, later". In either case, the condenser is dirty, and the bottom line is that the condensing temperature has increased; from 100ºF to 120ºF.

With the higher discharge pressure, the compression ratio has increased from 8.5:1 to 11:1. This higher pressure that the compressor is pumping against has a twofold negative impact. First, the motor amperage will be higher. Secondly, the volumetric efficiency of the compressor will be reduced.

There is a pocket of vapor between the bottom of the valve plate, and the top of the piston, called the clearance volume. It is there to insure that the piston does not run into the valve plate during the operation of the compressor. The vapor in this pocket requires re-expansion to a lower pressure before any new suction vapor can enter the compressor's cylinders.

Let's take a look at the compression cycle. It begins when suction vapor enters a cylinder as its piston is traveling down. As the piston starts to travel back up, reducing the volume of the cylinder, the vapor pressure increases. When the piston reaches the top of its stroke, the entire volume of compressed vapor will have exited the cylinder through the discharge valves EXCEPT for the vapor trapped in the clearance volume. It too is at the discharge pressure. Before any suction vapor can re-enter the cylinders, the clearance volume vapor must experience a reduction in pressure to a level slightly below that of the suction pressure. Otherwise, there would be no flow into the cylinder. It is piston travel, which increases the cylinder volume, that reduces this clearance volume pressure. This portion of the piston travel, which is entirely dedicated to lowering the clearance volume pressure, performs no useful work at all. In fact, it is advantageous to keep this to a minimum. The higher the clearance volume pressure is above the suction pressure, the more of this wasted piston travel will be required. Simply put, this is the definition of a high compression ratio (substituting discharge pressure for clearance volume pressure, as they are one and the same).

Note the higher HOC. Referring to the system design calculations reveals that more horsepower will be required. In addition, the quality of the saturated liquid/vapor mixture at the TEV outlet has further deteriorated, resulting in a lower RE. Referring again to the system design calculations will reveal that the lower the RE is, the higher the RCR has to be. This, in turn, will require more cylinder volume to meet the demand of an increased RCR. In a typical supermarket there are backup compressors that only operate under high load conditions, so

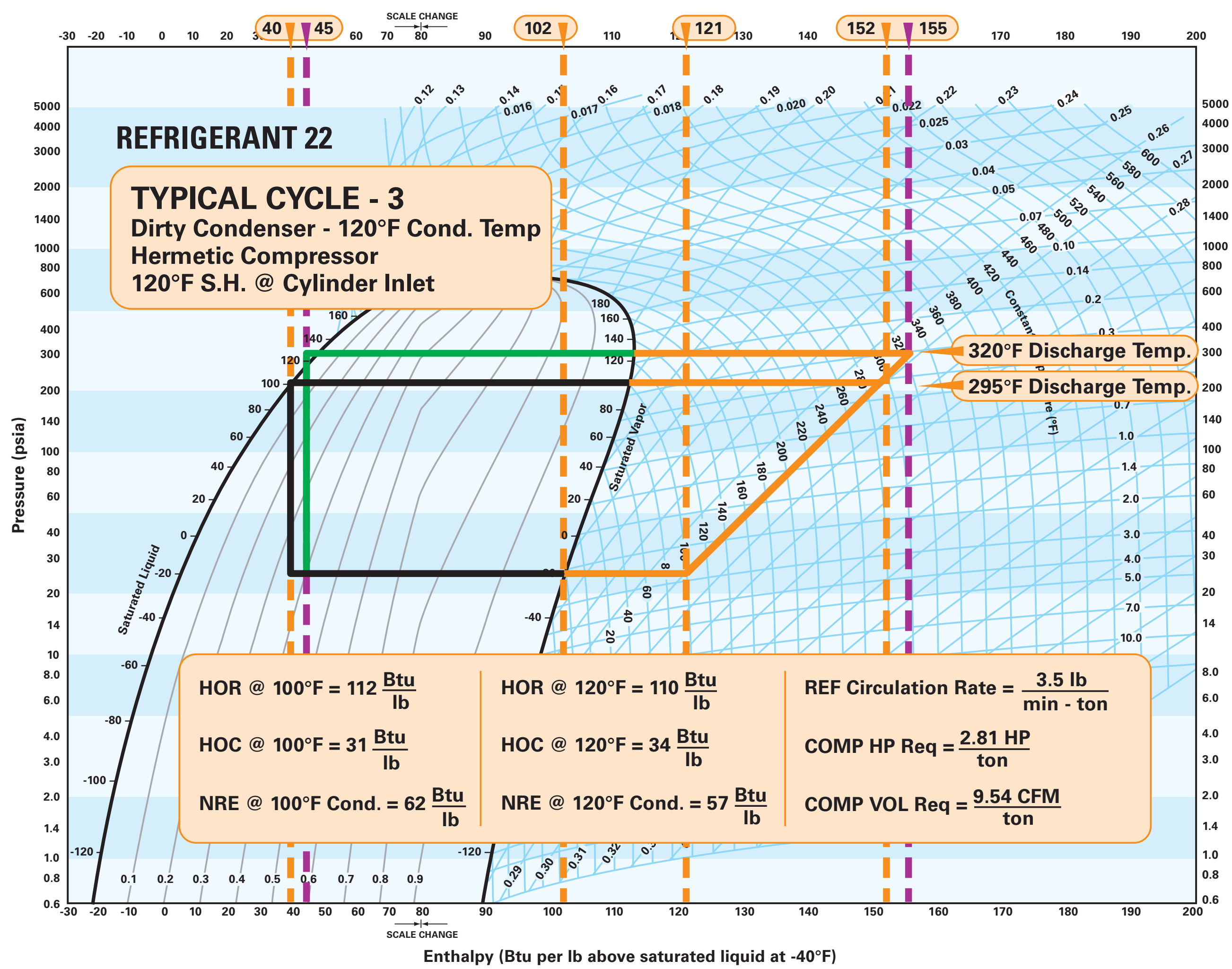

Fig. 7 – Typical Cycle #3

extra capacity is not a problem. For the store owner, paying for the additional electricity to operate the extra compressor(s) might be a major problem.

Finally, the discharge temperature has increased to 320ºF. At this elevated temperature the mineral oil's lubrication qualities will be diminished. Additionally, at 320ºF mineral oil will most certainly start to decompose. Under certain circumstances, the refrigerant (R22) may start to decompose as well. All of this spells a short destructive life for the compressor.

Typical Cycle #4 (Fig. 8): Now for the double whammy. Not only is the condenser filthy, but the TEV's were never adjusted. They are operating at abnormally high superheats, which in effect have reduced the size of the evaporators. As a result, the discharge air temperature in the glass door frozen food display cases is too high, causing the frozen juice to melt.

Because it is the dead of summer, and the service call log is overflowing, a quick solution to this problem would be great. So, in order to compensate for the high discharge air temperatures in the fixture lineup mentioned above, the service technician decides to lower the EPR setting. It doesn't occur to him that a -20ºF saturated suction temperature should be low enough to achieve a -10ºF discharge air temperature.

So, this makes sense…if a lower discharge air temperature is required, what easier way is there to accomplish it? By reducing the refrigerant pressure in the evaporator, the saturation temperature will be reduced. This allows for a higher temperature difference between the air entering the evaporator and the heat transfer medium (the refrigerant), resulting in a higher rate of heat transfer…or in simple terms a lower discharge air temperature. There is only one problem. The EPR is already wide open. Hey…how about lowering the set point for the common suction pressure on the rack's energy management control system. There's plenty of extra horsepower in the form of idle compressors, so capacity isn't an issue…just bring another compressor on.

With a simple adjustment, the fixture lineup's discharge air temperature can be brought in line. The pressure set point is lowered to 20 psia (-29ºF). Simple…yes…but there's a doozy of a problem with this approach.

Lowering the suction pressure will increase the compression ratio: in this example from 11:1 to 13.6:1…an approximate 20% increase over the compression ratio in Typical Cycle #3. This further reduces the compressor volumetric efficiency, and increases the horsepower requirement.

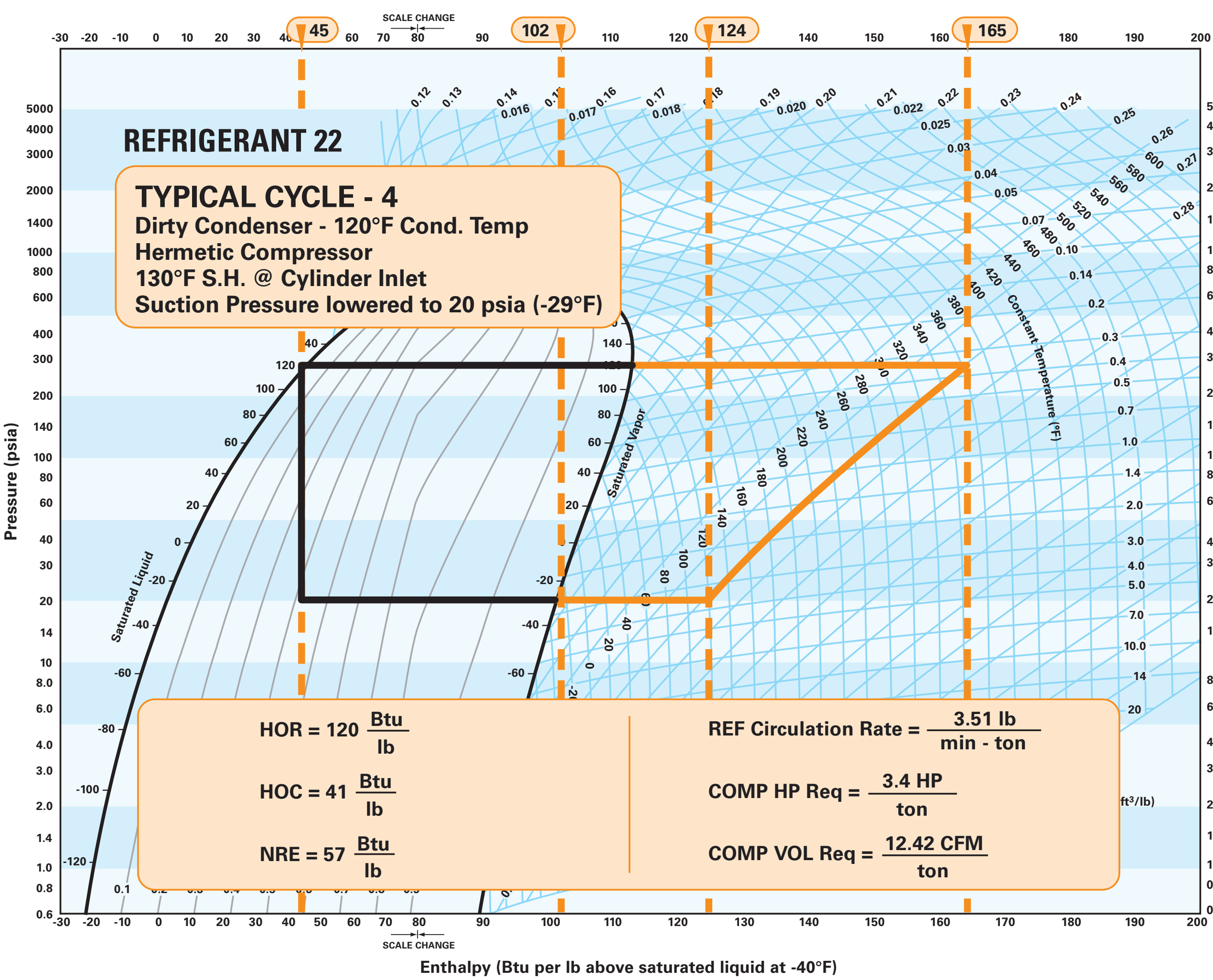

Fig. 8 – Typical Cycle #4

The lower the suction pressure is, the higher the refrigerant's specific volume will be, which translates into a greater compressor volume requirement. Simply reducing the suction pressure has resulted in a 30% increase in volume requirement over Typical Cycle #3.

The final blow is a higher discharge temperature, which is now approaching 370ºF…way beyond any margin of safety. It's guaranteed that the mineral oil will vaporize off the cylinder walls. This leaves the compressor's metal to metal moving parts vulnerable to accelerated wear, and certain failure. A look in the crankcase will reveal a black sludge where oil used to be. The crankshaft lubrication passages are now in danger of plugging up with the oil breakdown. It's quite possible that some of this oil breakdown will end up in TEV ports, which will leave them unable to control proper superheat. Unless preventative measures are taken, this compressor is headed for the scrap heap. In addition, the operation has become extremely inefficient, resulting in even higher energy costs.

Let's take a look at the chart comparing the system design calculations of the various cycles discussed. This can be seen in Fig. 9. The important values to look at are:

Compression Ratio: The ratio of absolute discharge pressure/absolute suction pressure. The compressor motor amperage will increase as the compression ratio increases. Also, the higher compression ratio results in reduced compressor volumetric efficiency. The net result is that it will take more energy to run a less efficient compressor. So, additional compressors will have to operate to compensate for this.

Discharge Temperature: A good measure of the relative health or sickness of the system. Most compressor manufacturers highly suggest the discharge temperature be kept below 300ºF. It is not realistic to put a temperature probe inside the cylinder to monitor temperature. Experience has shown that the discharge temperature 6" from the discharge service valve will be between 50º - 75º less than the actual discharge temperature. In addition to cleaning condensers, discharge temperatures can be reduced by proper expansion valve setting, compressor body cooling fan motors, good suction line insulation, liquid injection and…keeping the suction pressure as high as possible. For more details regarding discharge temperatures and liquid injection, refer to Sporlan Form 10-197.

Refrigerant Circulation Rate: The larger this number is, the more compressors will be required in operation to achieve capacity. It would be ideal to search out ways to reduce the amount of refrigerant to be pumped. Anything that affects the refrigerant quality at the TEV outlet will influence the RCR. It's not how much refrigerant we are feeding to the TEV, but how much of it remains after the expansion process, for the liquid refrigerant remaining is our only medium for transferring heat from the refrigerated space. Higher condensing temperatures have a negative affect on this. Notice the great reduction in the RCR where liquid subcooling is employed. Again, if the subcooling can be accomplished inexpensively, the potential for savings is great. It was the increase in discharge pressure (temperature) in Typical Cycle #3 which increased the RCR.

Compressor Horsepower: We can all relate to horsepower. The higher the horsepower requirement, the higher the electric bill will be at the end of the month. The subcooled liquid reduced the horsepower requirement. Because the refrigerant quality was reduced at the outlet of the TEV (more liquid present), it was used more efficiently. Therefore less of it was required to circulate, which caused a reduction in the horsepower required.

HOC and RCR are the two values which will determine the horsepower requirement. We see a 20% increase in the horsepower requirement in Typical Cycle #3 (as compared to Typical Cycle #2). This comes from the discharge pressure being raised (from 210 psia to 274 psia, an increase in the HOC). Such an easy thing to keep the condenser clean; yet how huge the impact is, if not done.

In Typical Cycle #4 the suction pressure is lowered from 24 psia to 19 psia. This 5 psi reduction in suction pressure results in a 20% increase in the system horsepower requirement as well. The lower suction pressure yields a smaller refrigeration effect, which results in a higher RCR. It also leads to a higher HOC. Since lowering the suction pressure will increase the HOC and RCR, it is important to keep the suction pressure as high as possible.

Compressor Volume: The compressor horsepower and the cylinder volume requirement are independent of one another. We see that from Typical Cycle #1 where there is no suction superheat, to Typical Cycle #2 where there is 40º of superheat, there is an approximate 20% increase in the cylinder volume requirement. The horsepower remains constant, however. It is the warmer, thinner suction vapor which drives the volume requirement up. One can see the tremendous penalty for using a hermetic compressor. Perhaps the reduced maintenance offsets the higher power bills. While high suction superheat increases the vapor specific volume (cu. ft./lb), it is the lower suction pressure in Typical Cycle #4 which has the greatest impact on increasing the specific volume.

There is a threefold negative affect from lower suction pressure:

> First, it will cause an increase in compression ratio, and the resulting decrease in volumetric efficiency. Lowering the suction pressure by a few pounds will result in a much greater increase to the compression ratio than by raising the discharge pressure the same amount.
>
> Secondly, the refrigerant's specific volume increases with a decrease in pressure. As the specific volume increases (a decrease in density), more cylinder volume is required to pump the same mass flow.
>
> Thirdly, the compression process follows the constant entropy lines. Therefore, the lower the suction pressue is, while following a constant entropy line during compression to a given condensing pressure, the higher the resulting discharge temperature will be. This has destructive consequences on the chemicals in the system, of which in part the compressor relies on for a long life.

The simplest way to insure against abnormally low suction pressure is to set the expansion valve for the proper superheat. When the TEV is set properly, the evaporator is efficiently used as a heat transfer device due to the boiling liquid present through most of its tubing length. High superheat settings reduce the amount of liquid available for heat transfer, in effect reducing the evaporator capacity. This can be negated by lowering the suction pressure, which lowers the saturation temperature, and yields a higher temperature difference between the entering air and the refrigerant. Evaporator capacities increase as the TD increases, so this allows for proper product temperature. It is done at the expense of lower suction pressure, and the negative impact it has on system health.

While several "realistic" system scenarios have been plotted and dissected, as stated earlier there are still some aspects of each cycle that are represented in an ideal fashion. There are pressure losses in the tubing, valves, accessories, etc. which are not shown. The compression process occurs at a constant entropy ONLY in the ideal cycle. In the real world, entropy will increase during the compression process, resulting in even higher discharge temperatures and HOC values. These factors should not detract from the basic focus of this article: to give one a solid foundation of the refrigeration cycle, based on what happens to the refrigerant as it travels throughout the system, and how the operating conditions can influence the relative health and efficiency of the system.

REFRIGERANT 22

Comparative Data with Varying Conditions

	Evaporator Temp (°F)	Suction Temp (°F)	Suction Superheat (°F)	Condenser Temp (°F)	Discharge Temp (°F)	Compressor Ratio	Discharge Superheat (°F)	Heat of Compression Btu/lb	Heat of Rejection Btu/lb	Refrigeration Effect Btu/lb	COP	REF Circulation Rate lb/min-ton	Comp HP Reqd hp/ton	Vapor Spec Vol cu ft/lb	Comp Vol Req cfm/ton
Ideal Cycle	-20	-20	0	100	180	8.5	80	27	89	62	2.30	3.23	2.05	2.07	6.68
Typical Cycle - 1 Open Drive Compressor (40°F Suction S.H.)	-20	20	40	100	215	8.5	115	27	95	62	2.30	3.23	2.05	2.29	7.39
Typical Cycle - 1A Open Drive Compressor (40°F Suction S.H.) Liquid subcooled to 50°F	-20	20	40	100	215	8.5	115	27	95	80	2.96	2.50	1.59	2.29	5.73
Typical Cycle - 2 Hermetic Compressor (add 80°F S.H.) 120°F S.H. vapor entering cylinders	-20	100	120	100	295	8.5	195	31	112	62	2.00	3.23	2.36	2.72	8.77
Typical Cycle - 3 Hermetic Compressor (add 80°F S.H.) 120°F S.H. vapor entering cylinders Dirty Condenser (120°F Cond. Temp)	-20	100	120	120	320	11.1	200	34	110	57	1.68	3.51	2.81	2.72	9.54
Typical Cycle - 4 Hermetic Compressor (add 80°F S.H.) 120°F S.H. vapor entering cylinders Dirty Condenser - 120°F Cond. Temp Suction pressure lowered to compensate for starving TEV's	-30	100	130	120	365	13.9	245	41	120	57	1.39	3.51	3.39	3.54	12.42

Fig. 9 – Comparison of System Design Calculations

Sporlan Division
Parker Hannifin Corporation
206 Lange Drive
Washington, MO 63090
636-239-1111 • FAX 636-239-9130
www.sporlan.com