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Compressor Overheating

Still Today's Most Serious Field Problem?

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More than 20 years ago an application bulletin entitled "Compressor Overheating: Today's Most Serious Field Problem" was published by a major compressor manufacturer. At first look, it might seem that compressor overheating would not be a major factor in the many refrigeration compressors that fail on any given day. Closer inspection, however, would reveal that many compressors suffering mechanical/electrical failures have their roots in overheating.

Why does overheating have such a devastating affect on a compressor? The answer is twofold:

1. **Loss of lubrication film:** Refrigeration oils have been highly refined in an effort to elevate the temperature at which chemical decomposition will occur. As such, they are vulnerable to losing the lubrication film necessary to prevent metal to metal contact between bearings and journals, or piston rings and cylinders, prior to the temperature at which decomposition begins. With mineral oil this will occur approximately between 310ºF and 330ºF. When these temperatures are achieved, the probability of extreme piston and ring wear is imminent.

2. **Chemical decomposition:** This happens at elevated temperatures, and is accelerated in the presence of other contaminants such as air or water. 18 is an important number to remember, for the rate of chemical reaction doubles with every 18ºF temperature increase.* For example, a chemical reaction that takes 10 years to complete at 100ºF, will only take 5 years to complete at 118ºF. At 136ºF it would be complete in 2-1/2 years, and so forth. The process by which the refrigerant and/or oil chemically breaks down can occur in a matter of seconds if there have been enough 18° temperature increases.

Mineral oil will start to decompose at approximately 350ºF (400ºF for POE oil). As temperatures increase above this threshold, the oil starts to polymerize. In plain English this means that the molecules that constitute the oil's makeup will start to combine into larger and larger molecules. First the oil is transformed into a dark thick oil, then a sludge, and finally a solid powder.

The presence of oil breakdown in the refrigeration system has many negative side effects. Sludge and solid particulate can plug up the oil inlet screen in the compressor sump or the lubrication passages in the crankshaft. Each of these will have the same devastating consequence: loss of lubrication, and ultimately failed bearings.

Oil breakdown deposits can also attach themselves to the internal surfaces of the refrigeration system. The inner walls of the piping, compressor, and control valves are all subject to this problem. While this will cause TEVs to stick/plug, and restrict the lubrication passages, oil breakdown on the interior walls of the piping may remain undisturbed for years (Figure 1).

Converting a system such as this from R-22/mineral oil to R-404A/POE oil will awaken the oil breakdown like a sleeping giant. With POE's solvent-like abilities, its presence in the system will literally clean these deposits from the interior surfaces and bring them back into circulation. One can expect contaminant problems after a refrigerant retrofit where the mineral oil has been replaced by POE oil: the new oil that was added to the compressor will likely be dark the next morning, and the odds are that a few TEVs will plug. Several filter-drier/oil filter/oil changes may be necessary before the problem is resolved. While unpleasant, it is simply a natural consequence of excessive discharge temperatures causing oil decomposition, and the solvent-like properties of POE bringing it back into circulation.

Figure 1 — *Oil Breakdown Deposits on Inner Walls of System Piping*

Now, let's take a look at a real world scenario in how destructive discharge temperatures can be. Most journeymen technicians have heard a compressor operating with what I'll term a "jackhammer inside the cylinder" syndrome. It is the result of a connecting rod wrist pin bore with excessive wear (Figure 2). The extra play allows the wrist pin to slap against

* *The Arrhenius Equation*

both ends of the bore. In addition, the piston may hit the bottom of the valve plate at the end of each compression cycle.

***Figure 2** — Worn Wrist Pin Bore*

Now here is the shocker…this mechanical failure was the result of excessive discharge temperatures. When this system was started, the installing contractor was pressed for time and neglected to set the thermostatic expansion valves (TEV). The resulting high superheat at the evaporator outlets translated into a higher suction superheat at the compressor inlet. For every 1ºF increase in the suction temperature an approximate 1ºF increase in the discharge temperature will be realized.

Now jump ahead a year or so. This system is suffering from lack of maintenance in the form of a dirty condenser (Figure 3). The consequence is a higher condensing temperature. According to EPA research, a heat transfer coil with a meager .042" film of dirt on its surface will result in up to a 21% loss of heat transfer capacity.*

***Figure 3** — Dirty Condenser*

The increased suction temperature (from underfeeding TEVs) plus the higher condensing temperature (dirty condenser) add up to a system operating with excessive discharge temperatures, and ripe for oil decomposition. After some period of time, what used to be clean looking straw colored oil in the crankcase will become thick and black. Some of this oil breakdown will be deposited on the valve plates (Figure 4), preventing the reed valves from sealing properly.

***Figure 4** — Valve Plate with Oil Breakdown Deposits*

Clearly, the oil breakdown deposits are the result of high discharge temperatures. Let's see what the full implications are. It starts with a discussion on how the piston wrist pins and their bearings are lubricated.

There is a small oil reservoir (hole) on the top of the wrist pin end of the connecting rod (Figure 5). Its function is simply to collect oil from the oil/refrigerant mist that is present in the crankcase. This oil is then supplied to the wrist pin bore by means of an oil groove, centered in the wrist pin bearing. In the case of this aluminum connecting rod, the precisely machined wrist pin bore (with oil groove) serves as the wrist pin bearing too.

As shown in Figure 6, when the piston is traveling down (suction), the point of contact creating the piston travel is between the top of the wrist pin and wrist pin bore. This allows the clearance to transfer to the bottom of the wrist pin/wrist pin bore. Oil from the wrist pin bore groove will flow into the clearance between these two moving metal parts, creating a lubricating film. This is how the bottom half of the wrist pin and bearing receives lubrication.

When the piston is traveling up (compression), the point of contact creating piston travel shifts to the bottom of the wrist pin and wrist pin bore. This now allows the clearance to shift to the top of the wrist pin and wrist pin bore. Oil that has accumulated in the reservoir can now flow into this clearance, lubricating the top portion of the wrist pin and bearing.

It is imperative that the point of contact between the

** Contracting Business — September 2003* wrist pin and wrist pin bore continue to shift from the top to bottom in this manner, as it allows the clearance to shift as well. Without the clearance shifting, the ability to lubricate the top and bottom of the wrist pin and bearing would be compromised.

***Figure 5** — Connecting Rod Wrist Pin Bore*

This is where the valve plate in Figure 4 comes into play. After the compressed vapor exits the cylinder, the discharge reeds should seat tightly, preventing any leakage back into the cylinder. Unfortunately, the oil breakdown deposits prevent this, allowing high pressure vapor to enter the cylinder during the downstroke (suction). Not only does this rob the compressor of its pumping capacity, it keeps high pressure on the top of the piston at all times. This is ultimately what causes the wrist pin bore to wear.

Continuous high pressure on top of the piston prevents the wrist pin clearance from shifting, keeping the point of contact at the lower portion of the wrist pin/wrist pin bore (Figure 6). Because of this, the lower portion of the wrist pin/wrist pin bore is unable to receive any oil, and the resulting metal to metal contact causes an abnormal wear pattern to the wrist pin bore.

This is most certainly a mechanical failure. A review of the steps that led to this failure will reveal that its roots are in excessive discharge temperatures.

1. High suction vapor temperature, resulting from underfeeding TEVs. The higher suction temperature increases the discharge temperature.

2. A dirty condenser raises the condensing temperature (and pressure), resulting in even higher discharge temperatures. Safe temperatures have now been exceeded, causing the oil to decompose.

3. The oil breakdown is deposited on the discharge reeds, preventing them from seating. Discharge vapor can now leak back into the cylinders during the suction stroke.

4. The constant high pressure exerted on the piston dome prevents the wrist pin/wrist pin bore clearance from shifting. The bottom portion of the wrist pin/wrist pin bore becomes the constant point of contact during upstroke and downstroke, preventing complete wrist pin/wrist pin bore lubrication.

5. After some period of time, the metal to metal contact causes abnormal wear to the softer aluminum wrist pin bore, resulting in the elongated bore.

Maintaining normal discharge temperatures would have extended the life of this compressor and reduced its electrical consumption.

Before prevention measures can be implemented, one must know what caused the excessive discharge temperatures in the first place. While many sources can be attributed, four main root causes lead to excessive discharge temperatures:

1. **High Suction Superheat**: One of the built-in inefficiencies in the vapor-compression cycle is the

***Figure 6** — Normal Lubrication of Wrist Pin/Wrist Pin Bore*

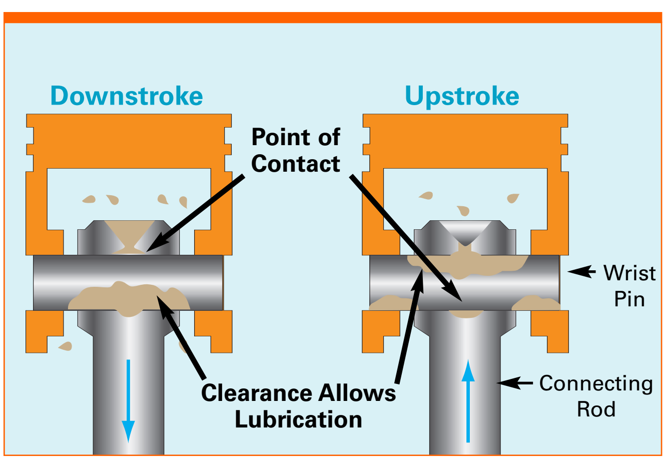
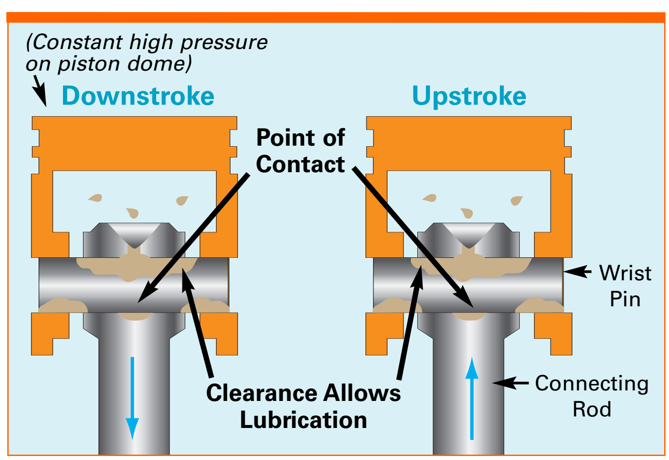


***Figure 7** — Loss of Lubrication to Lower Portion of Wrist Pin/Wrist Pin Bore*



heat added to the refrigerant between the evaporator outlet and the compressor discharge. A large amount of this can be attributed to the compression process. While the amount of heat added will vary depending on the refrigerant and system conditions, this “heat of compression” cannot be eliminated by any procedure. It is inherent in the compression process.

The compression process somewhat follows the constant entropy line on the P-H diagram (Figure 8). The constant entropy lines extend at an angle from the saturated vapor line, and become less vertical with each succeeding (or increasing value) line. So, the greater the value of the constant entropy line, the farther it will travel horizontally between two pressure points. It is this horizontal travel that represents the increase in the refrigerant’s heat content in Btu/lb. The entropy of a vapor will increase as its temperature increases. Therefore, it is the vapor’s temperature (which, is partially influenced by the amount of suction superheat) that will determine the constant entropy line from which the compression process follows. The laws of physics in this scenario never change: the higher the suction vapor temperature entering the compressor (resulting in a higher entropy), the higher the heat of compression.

For example, the system in Figure 8 represents an R-22 low temperature system operating at -20ºF saturated suction temperature (SST) and 100ºF saturated condensing temperature (SCT), with an open drive compressor. The refrigerant vapor temperature at the evaporator outlet is -15°F (5° superheat) The orange line reflects 40º of superheat at the compressor inlet, and the heat of compression (HOC) is 27 Btu/lb. The discharge temperature is 215ºF.

The system in Figure 9 represents a system with the same operating conditions, using a suction cooled hermetic compressor. The cool suction vapor must now travel across the warm compressor motor windings, experiencing an approximate 80ºF temperature increase in the process. The vapor temperature entering the cylinders is 100ºF (120º of superheat). The resulting discharge temperature is 295ºF.

Note that the suction temperature of the system in Figure 9 was 80ºF higher than that of the system in Figure 8. This translated into an 80ºF increase in the discharge temperature. For every

***Figure 8** — Pressure-Enthalpy Diagram Open Drive Compressor*

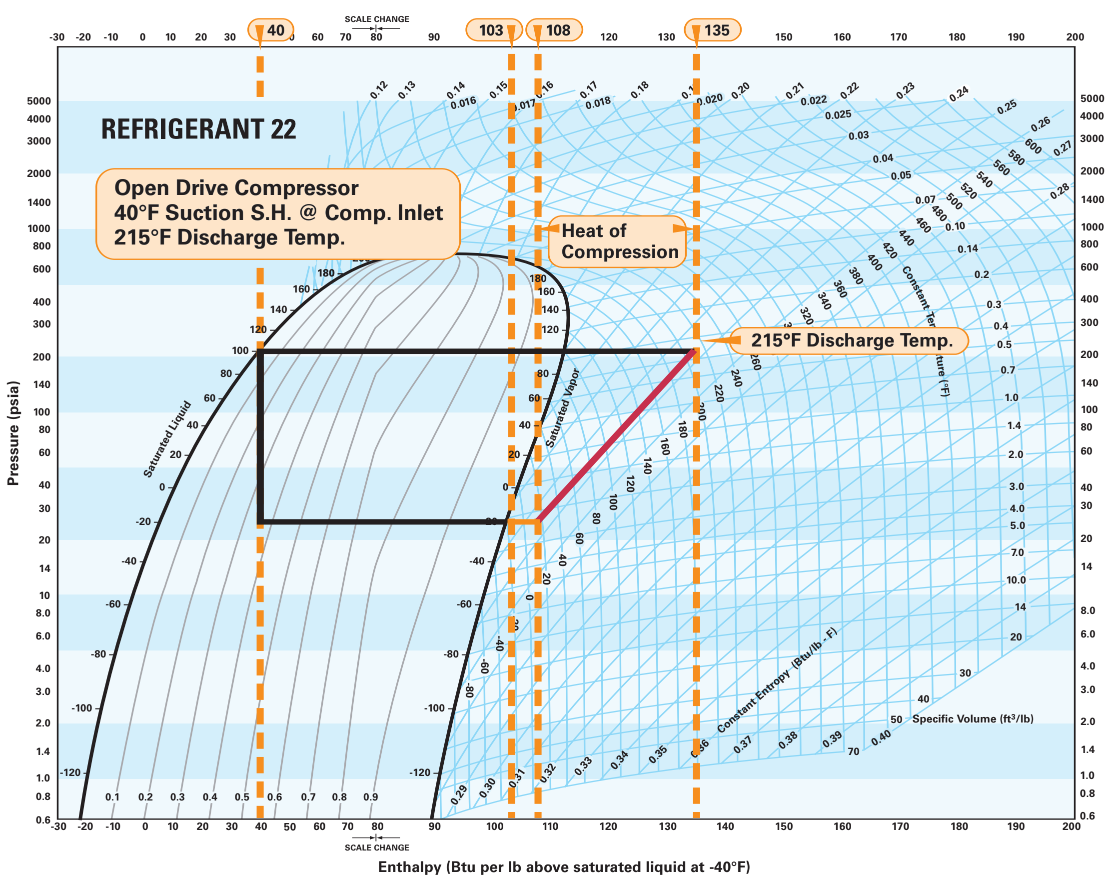
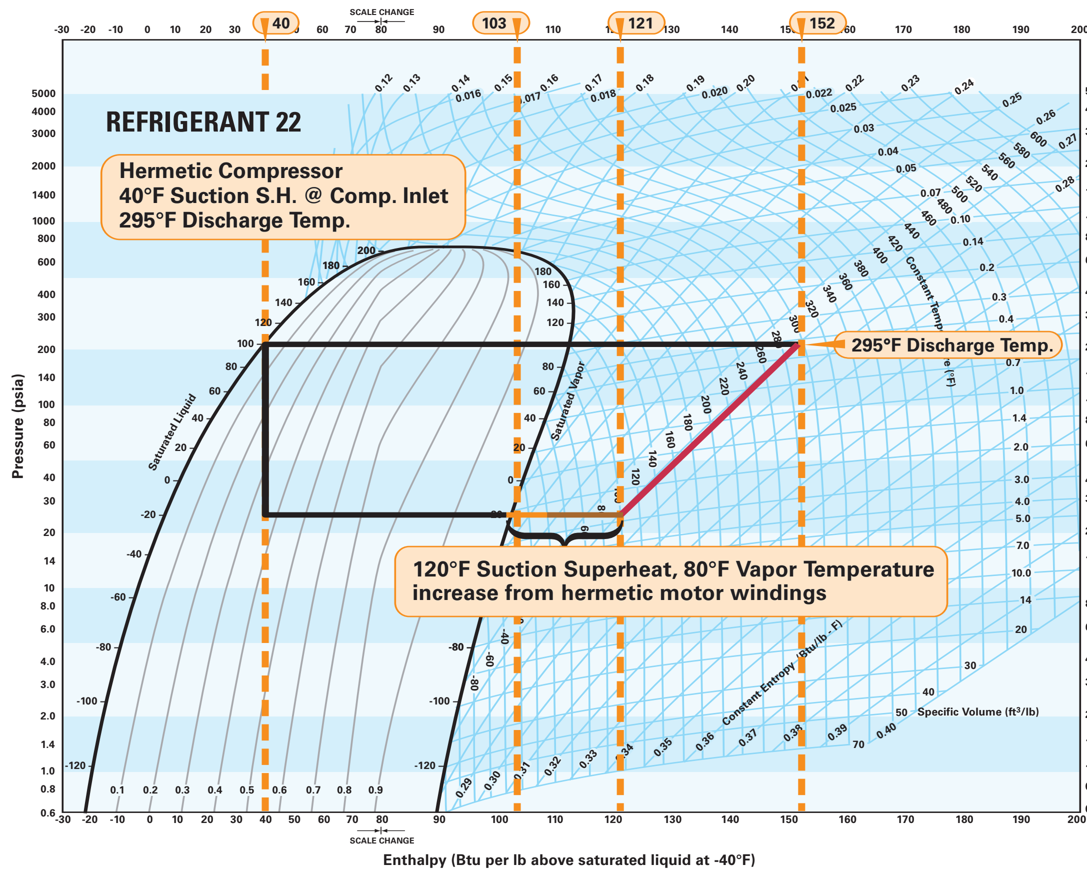


Figure 9 *— Pressure-Enthalpy Diagram Suction Cooled Hermetic Compressor*



1ºF increase to the suction temperature, you can expect an approximate 1ºF increase in the discharge temperature. Simply stated, any system condition that causes an increase in suction temperature will result in a nearly equal increase to the discharge temperature.

Some of the more common system conditions that are cause for increased suction temperatures are:

A. High TEV superheat setting. This should always be checked at startup and set to the equipment manufacturer's specification. The system TEV is not designed to regulate refrigerant vapor temperature at the compressor inlet. Allowing the system TEV to operate at a flood-back condition to ensure proper vapor temperature at the compressor inlet is dangerous (the potential for compressor damage) and inefficient (refrigerating the suction line adds operating expense, yet has no effect on product temperature). When set properly, the system TEV will maintain the lowest compressor suction temperature without sacrificing compressor safety or system efficiency.

B. Ineffective or missing insulation (Figure 10). There may be a temptation to use the lower cost 3/8" wall thickness insulation. This should be avoided. On long piping runs, or where the suction piping is routed through an attic, 1/2" or 1" wall thickness is recommended.

C. Liquid to suction heat exchangers. This is the classic "rob Peter to pay Paul" situation. The cool suction vapor subcools the liquid, which will ensure vapor free liquid feeding the TEV. It does so at the expense of higher suction vapor temperatures. Not only does this lead to increased discharge temperatures, the higher suction vapor temperatures reduce compressor efficiency. This cancels any efficiency gained from the subcooled liquid.

2. Insufficient Condenser Capacity: On rare occasions this might be the result of an undersized condenser. However, for every undersized condenser in service, probably thousands more have not been maintained. As the condenser fins become restricted with dirt, the airflow necessary to deliver the condenser's rated capacity is compromised.

Figure 11 shows the system mentioned previously, but with a dirty condenser. The condensing temperature has increased from 100ºF to 120ºF, and

***Figure 11** — Typical Cycle With Dirty Condenser*

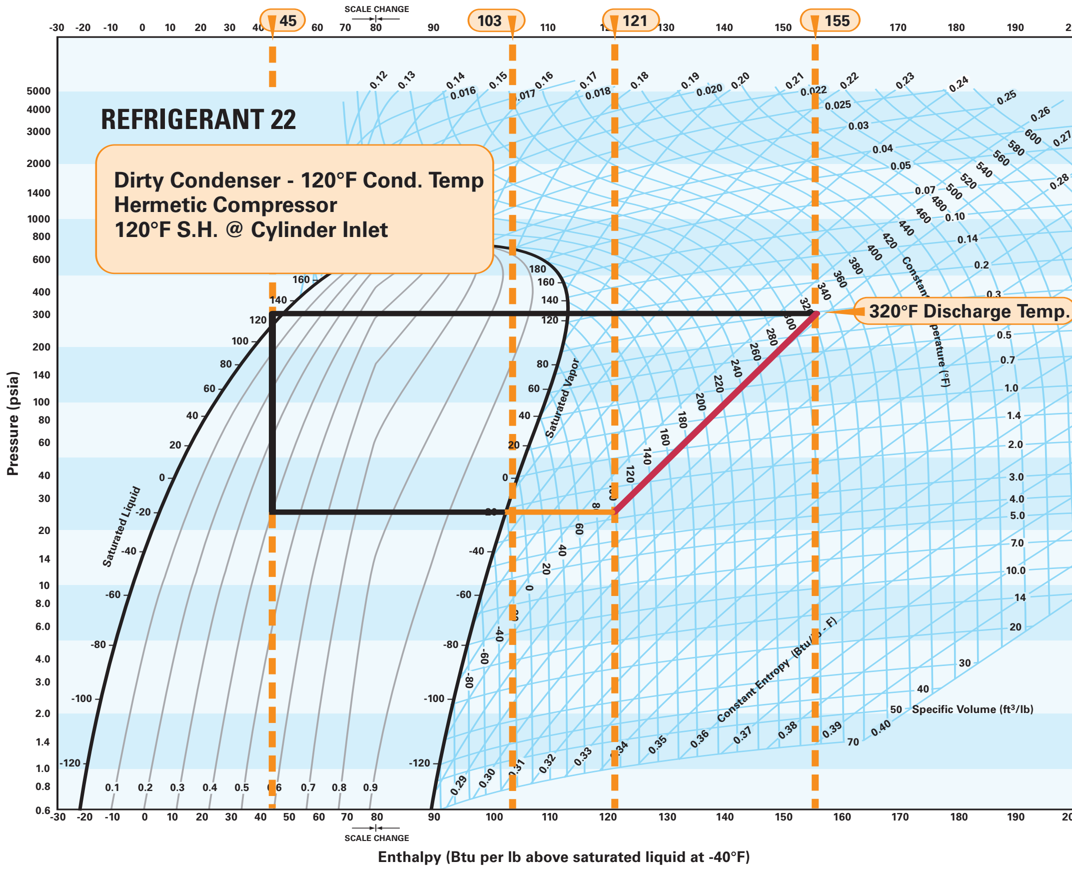


***Figure 12** — Typical Cycle With Dirty Condenser and Lowered Suction Pressure*

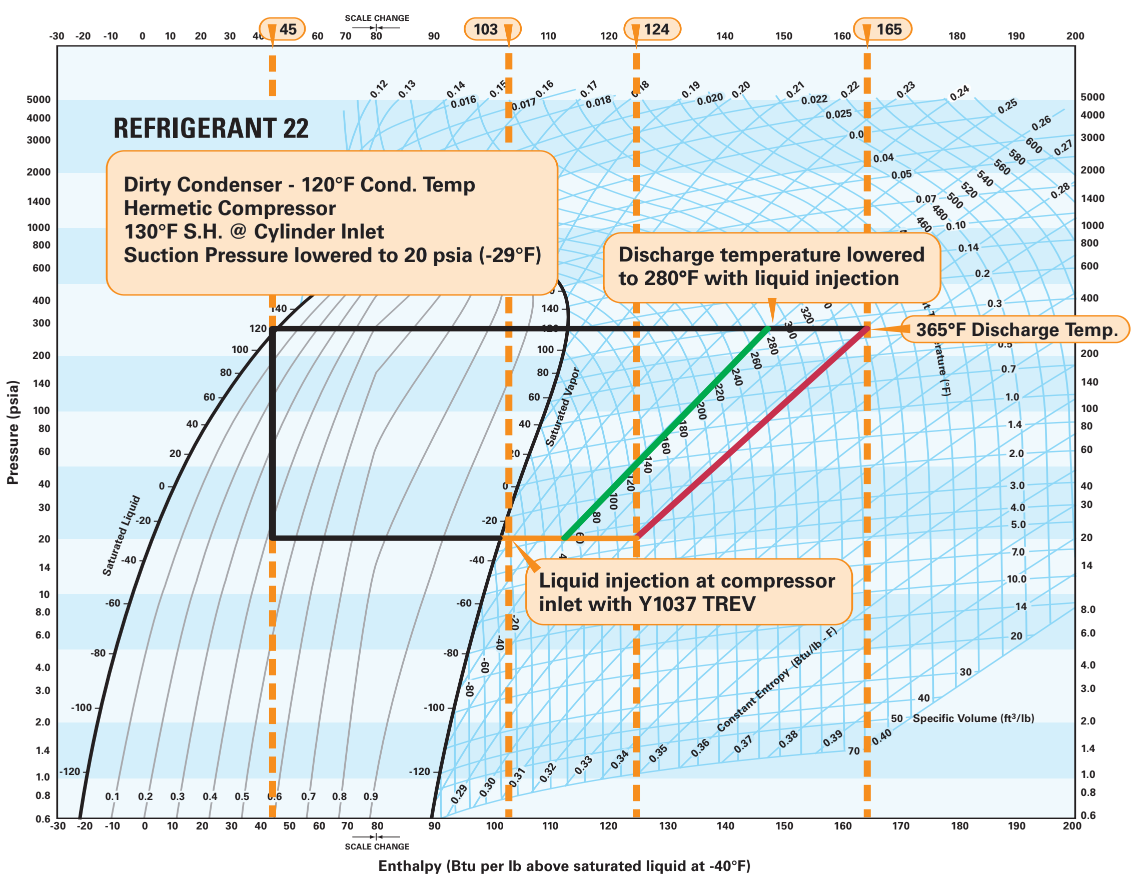
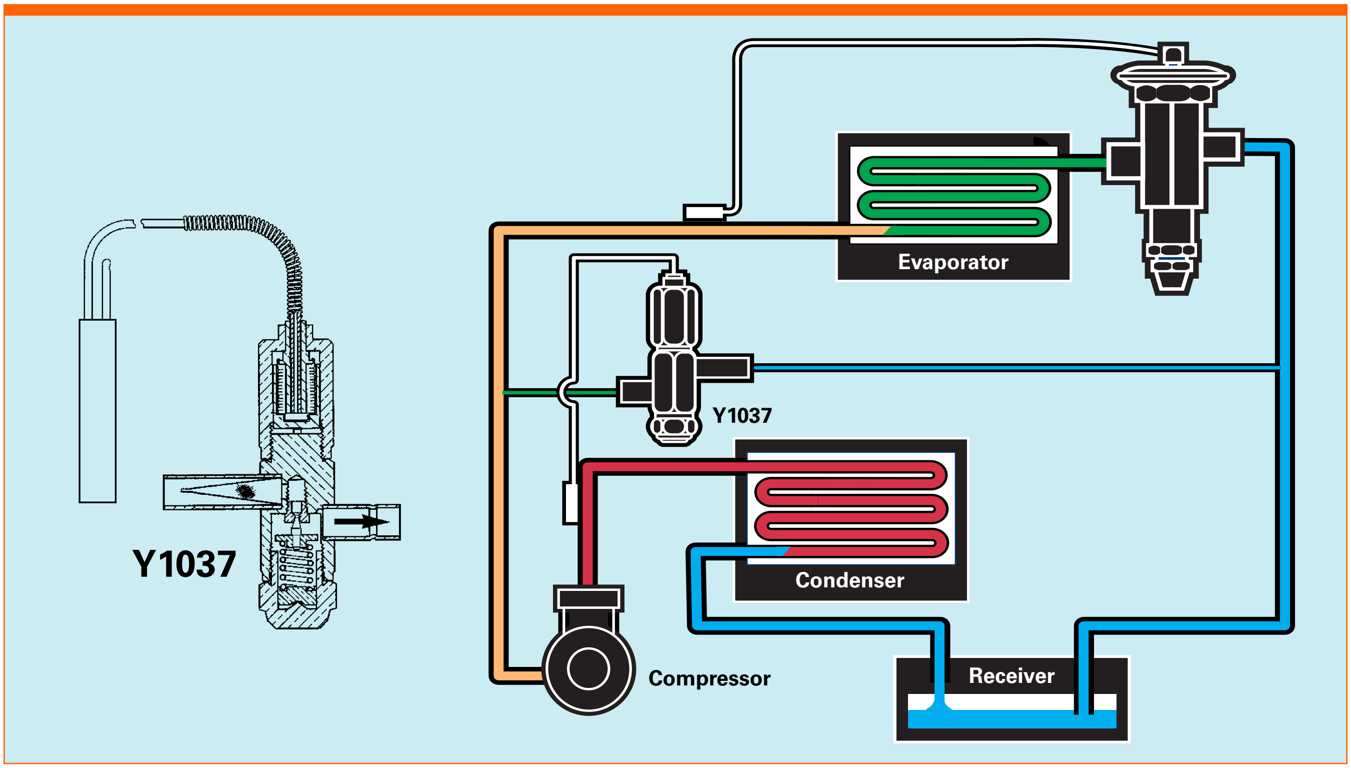


***Figure 13** — Sporlan Y1037 Temperature Responsive Expansion Valve*



the resulting discharge temperature has increased to 320ºF.

Figure 3 shows an example of this condition; it's a compressor waiting to fail. Sadly, this is easily…and completely…preventable.

***Figure 10** — Missing Suction Line Insulation*

3. **Lowering the Suction Pressure:** As mentioned above, the compression process roughly follows the constant entropy line on the Pressure-Enthalpy diagram.

   With a constant discharge pressure, when the compression process starts on a constant entropy line of a higher value, the resulting discharge temperature is higher. Starting the compression process on a higher constant entropy line can occur in two ways: higher suction temperature OR lower suction pressure. The system in Figure 12 is operating at a lower suction pressure; it has been lowered from 10 psig to 5.5 psig, while maintaining a condensing temperature of 120ºF. The result is a higher discharge temperature; 365ºF. It is important to operate the system with the highest possible suction pressure.

4. **Refrigerant Type:** Many criteria are used in the selection of a refrigerant for a given application: cost, availability, performance, and environmental issues. When R-502 joined the endangered species list, and suitable replacements for refrigeration applications were needed, it seemed logical to enlist R-22 as a replacement. It had been used for refrigeration applications before R-502 was developed in the 1960s.

   You have to wonder, "If R-22 was a good refrigerant for refrigeration applications, why was R-502 developed?" The answer is that R-22 has some limitations as a "refrigeration" refrigerant, particularly in low temperature applications. It is subject to higher compression ratios, which put increased stress on bearing surfaces and render the compression process less efficient. This can be overcome using a two-stage compression process; either a compound compressor or two separate compressors that compress the vapor in "two stages."

   More importantly, in low temperature applications R-22 has the potential for devastatingly high discharge temperatures. If the system is running per design...meaning that the condenser is sized correctly AND is clean, the TEV superheat is set correctly, the suction line is well insulated, and the suction pressure is operating at the highest possible level, it will still operate with high discharge temperatures. This is simply due to the physical properties and characteristics of R-22.

Because of this limitation, if R-22 is chosen as a low temperature refrigerant, something must be done to neutralize its "natural" high discharge temperature problem. A compressor body cooling fan motor is a start, as it can reduce the discharge temperature perhaps 15-25 degrees.

The real "something" will be to attack the problem at the compressor inlet. Recall that for every 1ºF change in the suction vapor temperature, there will be an approximate corresponding 1ºF change in the discharge temperature. This can either be an increase OR a reduction. Knowing this, it becomes rather easy to control the discharge temperature by controlling the suction temperature. Hence, the birth of the liquid injection valve.

Figure 13 shows the application of a liquid injection valve; a Sporlan Y-1037 Temperature Responsive Expansion Valve. Its function is quite simple: to monitor the compressor discharge temperature via a sensing bulb. As the discharge temperature exceeds the setting of the Y-1037, it injects a mixture of saturated liquid and vapor into the suction line. This will decrease the suction vapor temperature, which in turn reduces the discharge temperature. Knowing that the sensing bulb location is 50-75 degrees less than the actual discharge temperature, a technician can easily select a Y-1037 setting based upon the desired maximum discharge temperature. If the desired point for the valve to start injecting is a 280ºF discharge temperature, then the Y-1037 setting would be 205ºF (280 – 75). The system in Figure 12 illustrates the benefit of liquid injection.

The Y-1037 TREV responds to temperature only. There is no equalizer connection, and pressure does not influence its operation. The sensing bulb should be securely attached 6 inches from the compressor service valve for accurate control. Insulating the bulb is recommended to eliminate ambient temperatures from influencing the bulb temperature. The Y-1037 inlet should be connected to a vapor free source of liquid refrigerant; the outlet should be piped to the suction line 12"-18" from the compressor suction service valve. Because of the Y-1037s positive shut off capability, an additional solenoid valve is not typically required.

Liquid injection can be used to reduce excessive discharge temperatures on a single stage compression process, or to control the interstage temperature on a two stage compression process.

Conclusion: As stated more than two decades ago, compressor overheating IS today's most serious field problem. Many system problems and compressor failures can be directly traced to high discharge temperatures.

Several causes exist for this condition and most can be remedied by proper diagnosis and action. Condensers should be cleaned as needed to keep them operating at their rated capacity. The suction vapor temperature should be kept within acceptable limits by setting the system TEV correctly and insulating the suction line properly. Compressors should not be allowed to operate at abnormally low suction pressures, as this will lead to higher discharge temperatures.

With some refrigerants and applications, additional methods are required to alleviate high discharge temperatures. This can be accomplished to a degree with a compressor body cooling fan motor. For complete control in preventing this problem, a Temperature Responsive Expansion Valve, which monitors the discharge temperature, can be used. Injecting saturated liquid/vapor into the suction line will reduce the temperature of the superheated suction vapor, and in turn reduce the excessive discharge temperatures.

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