Understanding The Fundamentals Of Head Pressure Control

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It is said that necessity becomes the mother of invention. As energy prices soared in the 1970's, supermarket owners sought ways to reduce their enormous utility bills. Unlike comfort cooling equipment, which may only operate during the warmer months, the supermarket's compressors run year around. The refrigeration equipment will account for approximately 50% of the total electrical consumption in the typical supermarket **EVERY** month, with the compressors as the largest user. As such, there is the potential for conservation measures to have a huge impact on the monthly utility bill.

One means of reducing a compressor's electrical consumption is to lower the discharge (head) pressure. The head pressure will be at its highest during the heat of summer, when the ambient temperature is close to design conditions. This is also when the compressor motor amperage will be at its highest. Because more work is required to compress a vapor to a higher pressure, it logically follows that more electrical energy is required to accomplish this. There is a proportional relationship between head pressure and compressor motor current draw. When head pressure is either raised or lowered, the motor current draw will increase or decrease in proportion. For example, one manufacturer's 20 HP compressor, using R-404A, and operating at -25ºF SST (13 psig) and 110ºF SCT (272 psig), with 5ºF liquid subcooling, and a 50ºF return gas temperature, will deliver 61,389 Btu/h. At this condition the compressor motor's current draw is 40 amps. This may be one of several compressors (of varying sizes) on the low temperature "rack" in a supermarket. A 30ºF reduction in condensing temperature (110ºF to 80ºF) will reduce the current draw to 37.3 amps. While a 6.75% reduction is significant, it's only part of the story.

The typical supermarket compressor rack will run a reasonably constant suction pressure, regardless of head pressure. This is determined by the saturated suction temperature requirement of the lowest operating system connected to the rack, and maintained by the energy management system. Unless the suction pressure requires readjustment, the ONLY reduction in compression ratio (ratio of absolute discharge pressure/absolute suction pressure) will come from lowering the head pressure. In this case, lowering the condensing temperature to 80ºF (175 psig) will yield a significantly lower compression ratio (10.35:1 vs. 6.87:1). Using the same compressor data, the net result of this lower compression ration is a 31% increase in compressor capacity.

So, in addition to a 6.75% reduction in current draw, the 31% increase in compressor capacity means that **FEWER** compressors will be required to operate to achieve the same pumping capacity. This is where the real savings come from: **INCREASED COMPRESSOR VOLUMETRIC EFFICIENCY.**

Condenser capacities are based, in part, on TD (temperature difference) between the ambient and the refrigerant condensing temperature. As the ambient falls, and the TD increases, the condenser capacity will increase. For example, a condenser rated at 150,000 Btu/h at a 110ºF and 10ºF TD, would have a capacity of 750,000 Btu/h with a 50ºF TD. In laymen's terms, it has now become five times larger than it needs to be. An oversized condenser means lower head pressure, and reduced electrical consumption.

When the actual ambient is below the design ambient, we can take advantage of the now greater condenser capacity, allow the head pressure to fall, and start reaping the benefits—to a point. Too much of any good thing can become problematic, and reducing head pressure is no exception. **If the head pressure is allowed to fall below certain minimums, system performance can be adversely affected in the following areas:**

1. **Underfeeding TEVs (thermostatic expansion valves) and starving evaporators.**
2. **Oil logging.**
3. **Reduced compressor efficiency and higher discharge temperatures.**

1. Underfeeding TEVs and starving evaporators.

There are several factors which determine TEV capacity: refrigerant type, evaporator temperature, ΔP (pressure drop) across the TEV port, and liquid temperature. The nominal valve capacity (the capacity listed on the box) is "that capacity which the valve will deliver at the nominal rating condition." The rating condition for high pressure refrigerants (such as R-404A, R-507, or R-22) is a 40ºF evaporator temperature, 100# ΔP across the valve port, and 100ºF liquid temperature. If the actual conditions are different than the nominal rating

Figure 1

Pressure Drop Correction Factors for R-404A											
Evaporator Temperature °F	**30**	**50**	**75**	**100**	**125**	**150**	**175**	**200**	**225**	**250**	**275**
40°	0.55	0.71	0.87	1.00	1.12	1.22	1.32	1.41	1.50	1.58	1.66
20° & 0°	0.49	0.63	0.77	0.89	1.00	1.10	1.18	1.26	1.34	1.41	1.48
-10° & -20°	0.45	0.58	0.71	0.82	0.91	1.00	1.08	1.15	1.22	1.29	1.35
-40°	0.41	0.53	0.65	0.76	0.85	0.93	1.00	1.07	1.13	1.20	1.35

Figure 2

Liquid Temperature Correction Factors for R-404A												
20°	**30°**	**40°**	**50°**	**60°**	**70°**	**80°**	**90°**	**100°**	**110°**	**120°**	**130°**	**140°**
1.84	1.74	1.64	1.54	1.43	1.33	1.22	1.11	1.00	0.89	0.77	0.65	0.53

condition, then the actual valve capacity will be different than the nominal capacity. For example, the EGSE-2 has a nominal capacity of 2.04 tons (at the nominal rating condition). When applied on a system operating at -40ºF, the actual capacity for the EGSE-2 is 1.4 tons.

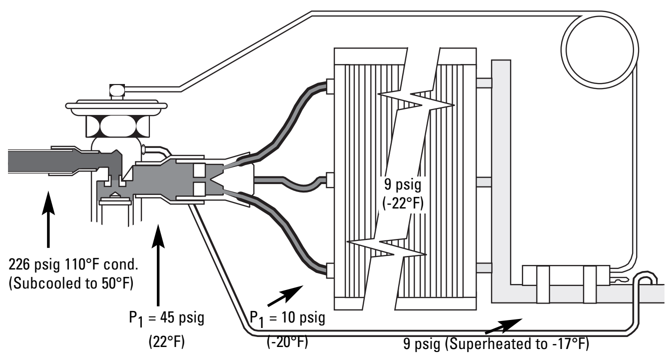
Similarly, altering either the liquid temperature (see Figure 1) or the ΔP across the valve port (see Figure 2) from the rating condition will result in a different valve capacity.

ΔP across the TEV port is not simply the difference between liquid pressure and evaporator pressure. If there is a refrigerant distributor in the circuit, its ΔP must be considered in the equation as well. The typical refrigerant distributor/tube assembly (in a high pressure refrigerant system), when correctly sized, will provide approximately a 35# ΔP. Referring to Figure 3, the TEV ΔP is calculated as follows: 226# (liquid pressure) minus 45# (the sum of 35# distributor/tube assembly ΔP and 10# evaporator inlet pressure), resulting in a 181# ΔP across the TEV port. As the head pressure lowers, the available ΔP across the TEV port is also lowered.

While lowering the head pressure results in a ΔP reduction (which will decrease TEV capacity), this is accompanied by a lower liquid temperature (a result of the lower condensing temperature) which will increase the TEV capacity. The effect of lower ΔP (reduced valve capacity) and lower liquid temperatures (increased valve capacity) will tend to negate each other without any significant change in TEV capacity.

A supermarket rack which utilizes a mechanical liquid subcooler is a different story. A reduction in ΔP **WITHOUT** an offsetting reduction in liquid temperature will result in a lower TEV capacity. The EGSE-2, at a -25ºF evaporator temperature, 219# ΔP (110ºF

Figure 3 – ΔP Across TEV Port



226 psi (liquid pressure) – [45 psi (TEV outlet) – 10 psi (evaporator inlet] = 181 psi

condensing), and 50ºF liquid temperature, will have a capacity of 36,600 BTU. Lowering the condensing temperature to 80ºF in the winter reduces the ΔP to 122#, resulting in a TEV capacity of 27,300 BTU.

While the lower head pressure yields reduced motor current and increased compressor efficiency, if lowered too far, eventually the TEV capacity would not be able to meet the demands of the evaporator load. When this occurs, a portion of the evaporator will cease to effectively transfer heat, as liquid refrigerant would no longer be available to feed it. This will be evidenced by the higher superheat at its outlet. That portion of the evaporator which only sees refrigerant vapor has essentially become an extension of the suction line; it performs no useful work at all. Reducing the TEV capacity, which leads to a starving evaporator, has in effect reduced the evaporator capacity. The end result is increased discharge air temperatures.

2. Oil logging.

Because the refrigerant and oil do not completely mix, a minimum refrigerant velocity in the suction line (particularly the riser) is required for the oil to properly return to the compressor. There is a delicate balance between an oversized suction line, which would adversely affect proper oil return, and an undersized suction line, which would cause unnecessary pressure drop and the resulting capacity decrease. At a full load condition, the suction riser ΔP, and refrigerant velocity will be at their peak. If sized correctly, the ΔP will be tolerable, and the velocity great enough for proper oil return. When the ambient temperature has decreased to that point where the TEV capacity cannot meet the evaporator load demand (from lower head pressure; available ΔP), the refrigerant mass flow in the evaporator will start decreasing. This in turn reduces the refrigerant velocity in the suction riser. There is some leeway in maintaining proper oil return with slightly reduced velocities. However, as the ambient temperature continues to fall (reducing ΔP, TEV capacity, mass flow and velocity), eventually the point will be reached where the velocity is simply too low for oil to return up the suction riser; it will log in the evaporator. Not only does this use valuable evaporator surface necessary for heat transfer, but if serious enough, may rob the available oil from the compressors for adequate lubrication. This will lead to compressor oil failure controls tripping, and the appearance of low oil levels in the system.

There may be temporary periods where the load is high enough (for example after a defrost cycle) for adequate velocity to allow the logged oil to return. Large amounts of oil sitting in the evaporator, if allowed to return in bulk, could seriously damage a compressor. This is a situation which should be avoided.

3. Reduced compressor efficiency and higher discharge temperatures.

It might seem a little odd for a compressor to operate less efficiently, and at higher discharge temperatures, from a reduced load condition. However, this is the result when the reduced load comes in the form of an underfeeding TEV, and the resulting high superheats. High superheat at the evaporator outlet means the suction vapor temperature entering the compressor will be warmer too. The specific volume (in cubic feet/lb) of the refrigerant vapor will increase as the temperature increases. While the compressor will continue to pump the same "volume" of refrigerant, the mass flow (in lbs/min) will decrease, reducing its effective pumping capacity. In addition, increasing the suction vapor temperature will result in higher discharge temperatures. In fact, for every 1ºF increase in suction gas temperature, there is an approximate 1ºF increase in discharge temperature. So while the underfeeding TEV will reduce the Btu load on the compressor, it will also cause less efficient operation at higher discharge temperatures.

We've now made the following case: reducing head pressure lowers the operating expense of the compressor, however reducing it too much can very adversely affect the health and longevity of the system. We can also conclude that the determining factor in deciding what the minimum allowable head pressure should be is the minimum TEV ΔP required for its capacity to meet the demands of the evaporator load. Once that has been determined, it becomes a simple matter of adjusting the head pressure controls on the system to maintain that minimum. There are several different methods of maintaining head pressure, and it is important to understand the principles of how each one operates so they can be set correctly. In addition, on systems which utilize more than one method of head pressure control, setting them to operate **TOGETHER** is imperative. **The common methods of head pressure control are:**

1. **Fan cycling.**
2. **Condenser flooding.**
3. **Condenser splitting.**

1. Fan cycling.

The capacity of the condenser mentioned earlier, rated at 150,000 Btu/h (at 110ºF and a 10ºF TD), is based on all of the fans operating. If this is an eight fan condenser, we can reduce the condenser capacity as needed by cycling off the fans. Typically, fan motors 1 & 2 would be controlled by a pressure control, fan motors 3 & 4 by another pressure control, etc. This would allow for four stages of fan cycling. A typical pressure control setup is shown in Figure 4. Excluding a cold windy day, where the wind is able to blow through the tube bundle on the condenser, the head pressure would never fall below 160 psig with this control strategy.

Figure 4 – Typical Fan Cycle Control Set Points

Condenser Set Point		
Fan #	Cut-In	Cut-Out
1 & 2	180 psig	160 psig
3 & 4	190 psig	170 psig
5 & 6	200 psig	180 psig
7 & 8	210 psig	190 psig

While this is a simple method of control, it does have a few disadvantages. For example: on a cool day, when the head pressure reaches 170 psig, pressure control #2 will open the control circuit for the contactors powering fan motors 3 & 4. Once fan motors 3 & 4 cycle off, the head pressure will gradually start to rise, as the reduction in air flow has reduced the condenser capacity. The corresponding refrigerant saturation temperature will also rise. In the receiver, where liquid and vapor are present, the refrigerant will always be at a saturated condition (R-404A is 78ºF at 170 psig). For simplicity, we will assume the pressure in the receiver is equal to the head pressure. In an operating system there would be some pressure loss in the piping and flow controls, resulting in a receiver pressure that would be less than the head pressure. When the switch in pressure control #2 closes at 190 psig, the temperature of the refrigerant in the receiver will be 85ºF (saturation temperature at 190 psig). With the second bank of fan motors operating, the head pressure will fall rapidly, as will the corresponding saturation temperature. The temperature reduction of saturated refrigerant in the receiver is accomplished by liquid flashing into vapor. As a portion of the liquid flashes, it absorbs enough heat from the remaining liquid to lower its temperature to the new saturation condition. Depending on the refrigerant level in the receiver, and how quickly the pressure falls, the ability to provide vapor free refrigerant to the liquid header may be temporarily compromised. If there is little or no subcooling in the liquid line, flashing may occur all the way to the TEV inlet. This temporary disruption of vapor free refrigerant to the TEV results in erratic operation and poor superheat control.

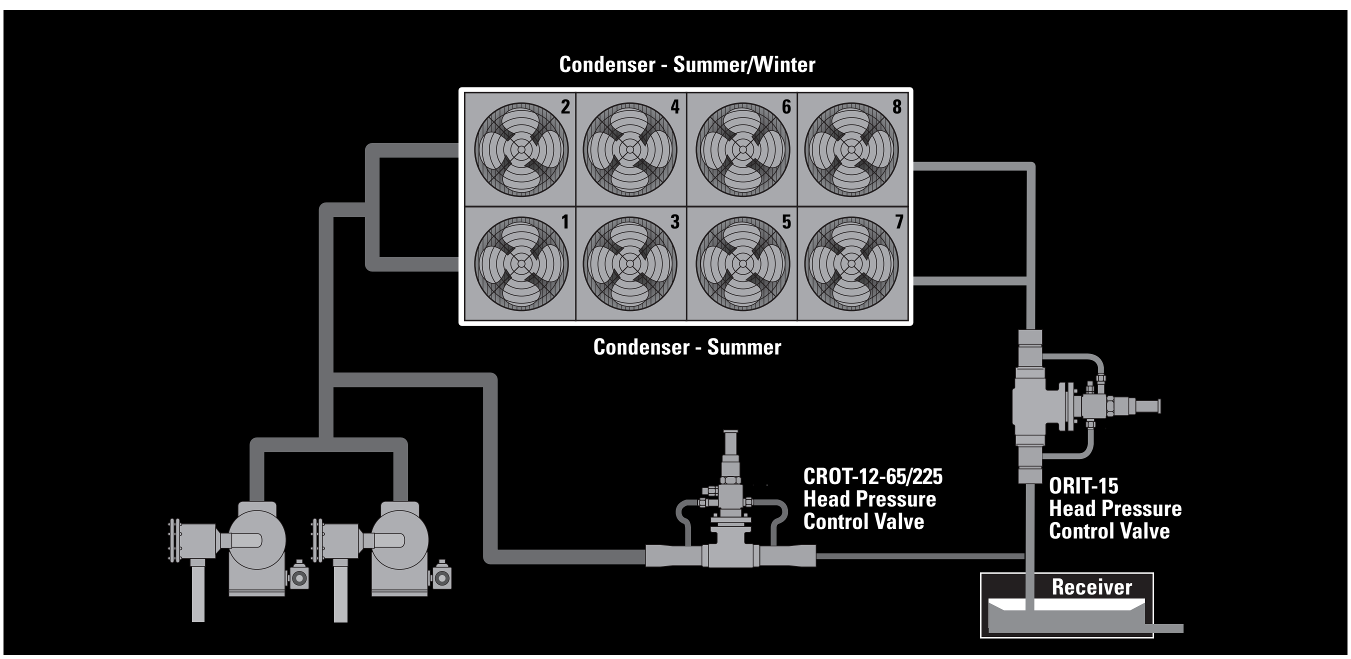
In addition, as the pressure controls cycle the fan motors, and the pressure fluctuates between cut-ins and cut-outs, the available ΔP across the TEV port will vary. In a perfect world the condition of the refrigerant at the TEV inlet would be constant year around. Allowing the head pressure to fluctuate up and down every few minutes will result in TEV capacities that proportionally fluctuate.

2. Condenser flooding.

The ability to maintain **CONSTANT** head pressure (liquid pressure) during varying periods of low ambient operation would be ideal. One method of achieving this is to use condenser flooding valves. In larger systems, two valves are required (see Figure 5). The first, commonly referred to as the condenser holdback valve, is installed at the outlet of the condenser. Its function is to maintain a constant pressure in the condenser.

The ORIT valve is normally closed, and **O**pens on a **R**ise of **I**nlet pressure. If the ORIT valve was set to maintain 180 psig, it would simply remain closed until the condenser pressure increased to that level. While the ORIT valve is closed, the compressor continues to pump refrigerant into the condenser. As heat is removed from the superheated discharge vapor it will start to condense into a liquid, and the liquid refrigerant will start backing up from the inlet of the closed ORIT valve, "flooding" a portion of the condenser. That portion of the condenser which is full of liquid refrigerant (flooded) no longer serves as a condenser. Flooding a portion of the condenser reduces its effective heat transfer surface, and therefore its capacity. When the appropriate amount of condenser flooding has

Figure 5 – Head Pressure Flooding Valves



occurred, the reduced condenser capacity will cause the pressure to increase to 180 psig. At this point, the ORIT will begin to open and allow refrigerant to flow into the receiver.

Pressure regulating valves can control either upstream pressure or downstream pressure, but not both. When the ORIT throttles, maintaining constant condensing pressure by flooding the condenser, it does so at the expense of its outlet pressure (receiver pressure). The ORIT valve may influence receiver pressure, but it cannot maintain it at a constant level. Without an additional regulating valve, the pressure in the receiver will be erratic during periods of low ambient operation due to the ORIT valve throttling.

A second valve is necessary to maintain constant receiver pressure. It is commonly referred to as the "receiver pressurizing valve" (see Figure 5). The CROT valve is normally open, and **C**loses on a **R**ise in **O**utlet pressure, and is typically set to maintain a pressure approximately 20 psig less than the ORIT valve set point. As the ORIT valve throttles, maintaining constant condenser pressure **AND** interrupting the flow of refrigerant to the receiver, it is the CROT valve which maintains a constant pressure in the receiver.

The benefit of condenser flooding is the ability to provide very consistent liquid pressure in the receiver during periods of low ambient operation. Consistent liquid pressure will result in very stable TEV operation during the winter months.

There are two drawbacks to this method of head pressure control:

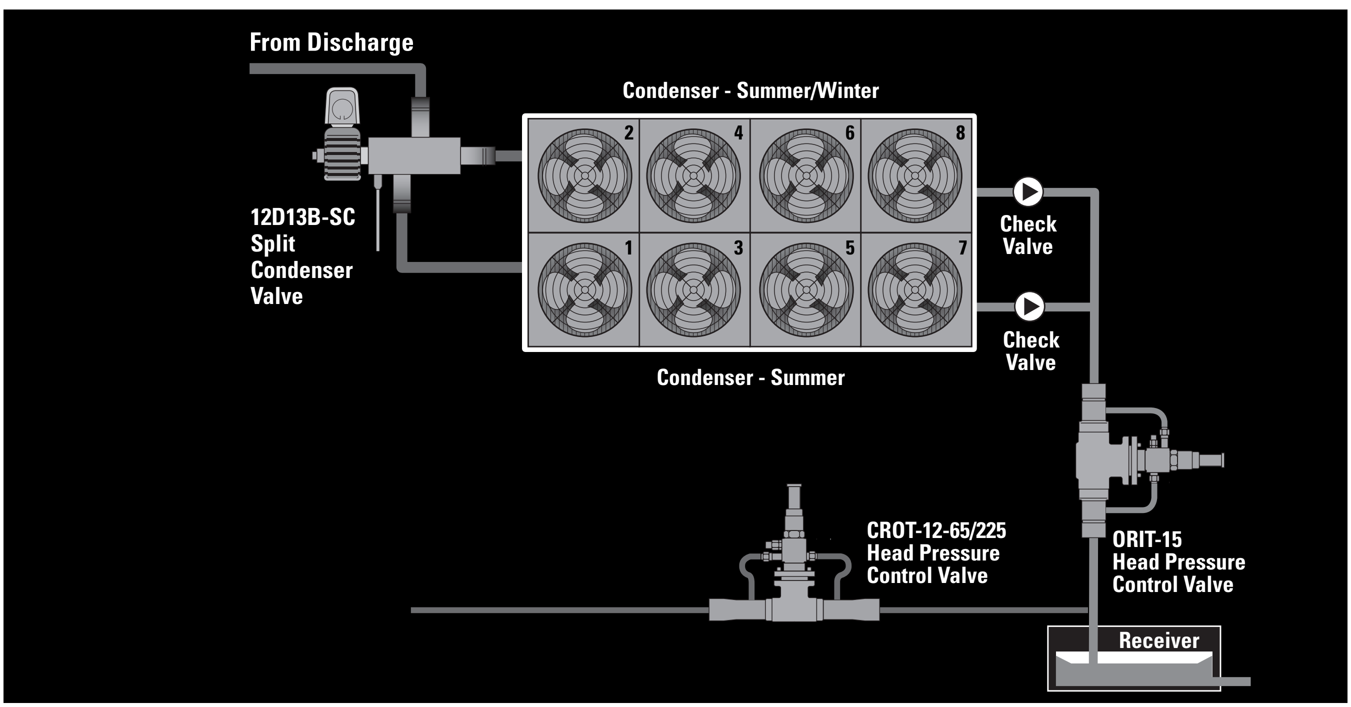
1. Extra refrigerant is required to accomplish condenser flooding. The approximate percentage of required condenser flooding can be calculated in Sporlan's Bulletin 90-31. During extreme low ambient conditions, it may be necessary to flood upwards of 85% of the condenser. Depending on the size of the condenser, this may require several hundred pounds of extra refrigerant. In today's marketplace, this can become quite expensive.

2. Receivers should be sized such that they are at 80% of their capacity while containing the entire system charge. If extra charge is needed in the system for condenser flooding, a larger receiver will be required. The extra refrigerant added to flood the condenser during periods of low ambient will be in the receiver during the warmer months. In systems where the additional refrigerant charge hasn't been considered in the receiver sizing, the technician will have to remove refrigerant every spring to prevent high discharge pressures at design ambient, only to add it back in the fall when it will be required for flooding.

3. Condenser splitting.

Reducing the amount of extra charge needed for condenser flooding can be accomplished by splitting the condenser into two identical circuits; one for summer/winter operation, and the other for summer operation only. The summer condenser will be cycled off as needed during periods of low ambient. This requires the addition of a condenser splitting valve; a three way valve which will be installed in the discharge line (see Figure 6).

Figure 6 – Split Condenser Valve



When the 12D13B-SC is de-energized, the main valve piston is positioned to enable refrigerant flow from the inlet port to flow equally to the two outlet ports, feeding both condenser halves. When required, energizing the solenoid coil will shift the main piston, closing off the flow of refrigerant to the port on the bottom of the valve. This removes the summer half of the condenser from the circuit, and now the minimum head pressure can be maintained by flooding the summer/winter half of the condenser.

To prevent the summer condenser from logging refrigerant during low ambient periods, a check valve is installed at its outlet. This will eliminate the possibility of refrigerant backflow into the idle summer condenser. While a check valve isn't necessary at the outlet of the summer/winter condenser for backflow prevention, it is added so that the ΔP though each condenser is equal. This is necessary to insure equal refrigerant flow through the two condensers.

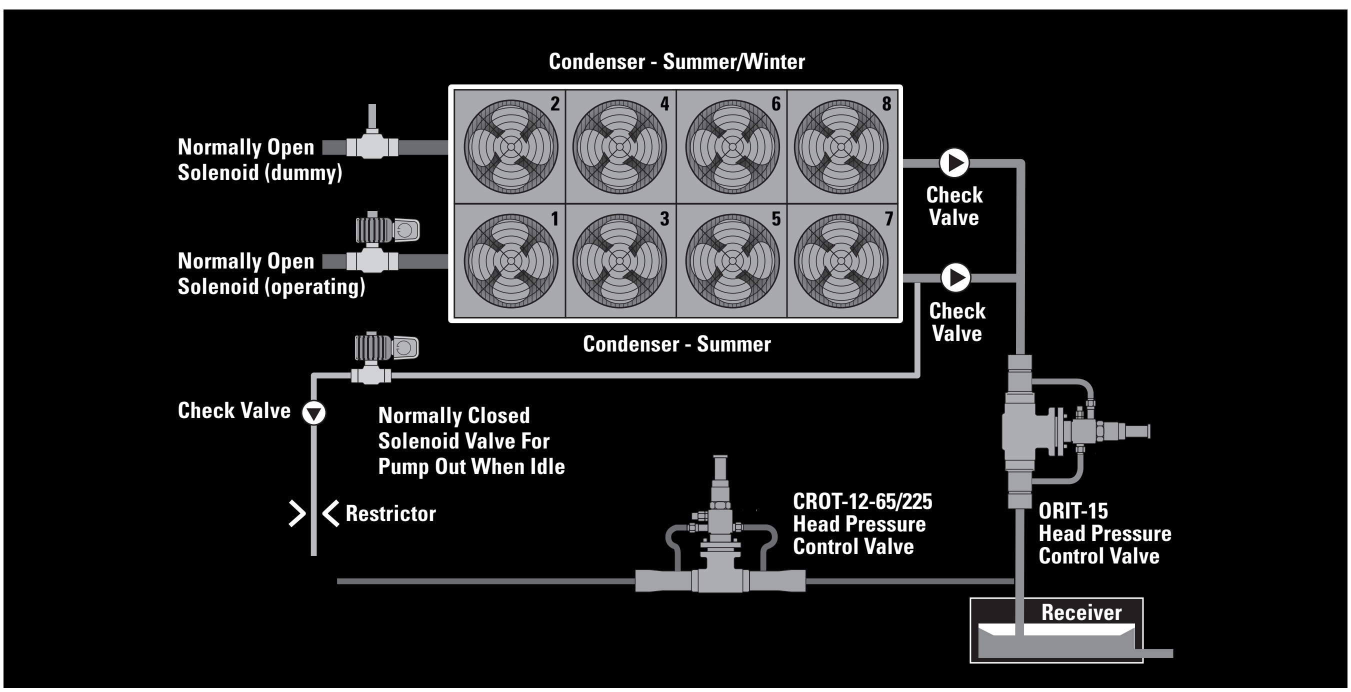
During periods of split condenser operation, it is recommended that the refrigerant in the idle condenser be transferred back into the system. This can be accomplished by using the "B" split condenser valve model, which has a bleed hole in its upper piston. The refrigerant will flow through the piston, into the valve's pilot assembly, and back to the suction header. If the "B" version is not used, a dedicated pump out solenoid valve is required, which vents the idle condenser to the suction header through a restriction such as a cap tube.

The pump out line must also have a check valve installed to prevent backflow.

As an alternative to a 3-way split condenser valve, two normally open solenoid valves can be used (see Figure 7). In this application there would be a normally open solenoid at the inlet of the summer condenser, which would cycle closed during low ambient conditions. An identical solenoid is installed at the inlet of the summer/winter condenser. It does not require a solenoid coil; this valve is installed to maintain equal pressure drop through the two condensers. As in the three way split condenser valve application, a check valve will be necessary for the outlet of each condenser. With this method, a dedicated normally closed pump out solenoid valve will be required to vent the refrigerant from the idle summer condenser into the suction header.

In summary, allowing the head pressure in supermarket refrigeration systems to operate at reduced levels during periods of low ambient results in lower compressor motor amperage, increased compressor efficiency, and lower monthly utility bills. Head pressure has a direct affect on available ΔP across the TEV port, which in addition to evaporator and liquid refrigerant temperatures, will determine the TEV capacity. The minimum ΔP required to deliver the necessary TEV capacity to meet the load demand of the evaporator **IS** the limiting factor on how low the head pressure can be allowed to float. Once this is calculated, based on TEV capacity

Figure 7 – Split Condenser With 2 Normally Open Solenoid Valves



data, the minimum head pressure can be determined, and used to establish the set points of the head pressure control devices.

While several methods of head pressure control are available, one that allows the head pressure (liquid pressure) to remain constant is desirable. This is best accomplished by condenser flooding valves, which maintain consistent head pressure by flooding a portion of the condenser with liquid refrigerant.

While condenser flooding valves provide the most consistent head pressure, this method requires adding extra refrigerant to the system. Using a 3-way condenser splitting valve (or two normally open solenoid valves) will offer the ability to reduce the condenser capacity by 50% during low ambient conditions. After cycling off the summer condenser, the remaining summer/winter condenser can utilize the flooding method with a minimum of additional refrigerant to maintain constant head pressure.

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