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Heat Reclaim

Benefits, Methods, & Troubleshooting

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While the vapor-compression cycle has four stages (compression, condensation, expansion, and evaporation), there is one ultimate goal to its purpose: converting the refrigerant into a state/condition that makes it useful as a heat transfer medium. For example: if the design discharge air temperature for a particular application is 25ºF, and the evaporator is applied at a 10º temperature difference, then the saturation temperature of the refrigerant in the evaporator should be 15ºF. The goal of the vapor-compression cycle for this application would be to deliver the necessary mass flow of saturated liquid refrigerant at 15ºF to the evaporator inlet; the amount would be sufficient to transfer the design heat load from the refrigerated space to the refrigerant.

So, if the goal is to convert the refrigerant to the state/condition necessary to be a useful heat transfer medium (the evaporation stage), then the other three stages can be looked upon simply as "a means to an end". They are all essential and each can be altered to improve efficiency, however they perform no useful function other than "preparing" the refrigerant in its journey of transformation into something useful. Or do they?

Consider the compression stage. Every experienced technician knows how hot the discharge line can be; this heat is the waste byproduct from the vapor compression process, and is normally rejected at the condenser. Heat is a form of energy; rather than "throw it away" at the condenser, couldn't it be used somewhere…somehow?? Yes, in fact, it can! If the proper controls and accessories are added to the system, this heat can be reclaimed and put to use.

The modern supermarket consists of several hundred feet of refrigerated display fixtures in the sales area, along with several medium and low temperature walk-in boxes for storage. It is not uncommon to employ large multiplexed compressor racks to provide the necessary Btu capacity to refrigerate the various fixtures in the store; these may be in the range of 30 to 40 tons each. Each one of these large capacity racks rejects a substantial amount of heat at the condenser as part of the vapor compression cycle. Heat reclaim takes a portion of the condenser heat of rejection and uses it to heat water and/or the supply air in the building's HVAC system. Instead of refrigerant flow from the compressor discharge to the condenser inlet, the 3-way heat reclaim valve (Figure 1) diverts the flow of refrigerant from the compressor discharge to the heat reclaim coil inlet.

3-Way Valve Operation

The 8D7B in Figure 1 is an example of a pilot operated 3-way heat reclaim valve; its main piston is controlled by a three-way pilot valve. The common port on the 3-way pilot is connected to the main piston chamber. The pilot's normally open upper port and normally closed lower port are respectively connected to the suction header, and the valve's inlet fitting (discharge pressure). The 3-way valve is in the normal condensing mode when the 3-way pilot valve's solenoid coil is de-energized. This allows the main piston chamber to vent to the suction header, and the resulting pressure differential (discharge pressure acting on the bottom of the lower main piston, and suction pressure acting on top of the upper main piston) causes the main piston to shift upwards. The 3-way valve's upper outlet port closes (heat reclaim port), and allows full refrigerant flow to the bottom outlet port (normal condenser port).

Energizing the 3-way pilot solenoid coil will shift the 3-way valve into the heat reclaim mode. This simultaneously closes the pilot's suction port, and opens the pilot's discharge port. The piston chamber is pressurized with discharge refrigerant vapor, and the resulting pressure differential (discharge pressure and the opening spring acting on top of the larger diameter upper main piston, and discharge pressure acting on the bottom of the smaller diameter lower main piston) causes the main piston to shift downwards. The 3-way valve's lower outlet port closes (normal condenser port), and allows full refrigerant flow to the upper outlet port (heat reclaim port).

If heat reclaim is used for heating the building's supply air, the 3-way valve would be controlled by the building HVAC controls. When the building temperature falls below its set-point, the HVAC controls would energize the 3-way valve solenoid coil, shifting the main piston and allowing the flow of discharge vapor into the heat reclaim coil. Heat would then be transferred from the discharge vapor to the ventilating supply air passing through the heat reclaim coil's fin/tube bundle. In milder winter climates this may provide enough heating capacity to meet the comfort needs of the building, meaning no additional heating equipment need be purchased, installed and maintained. If not, a second stage of gas or electric heat can be employed.

3-Way Heat Reclaim Valve
Type B with condenser pump out

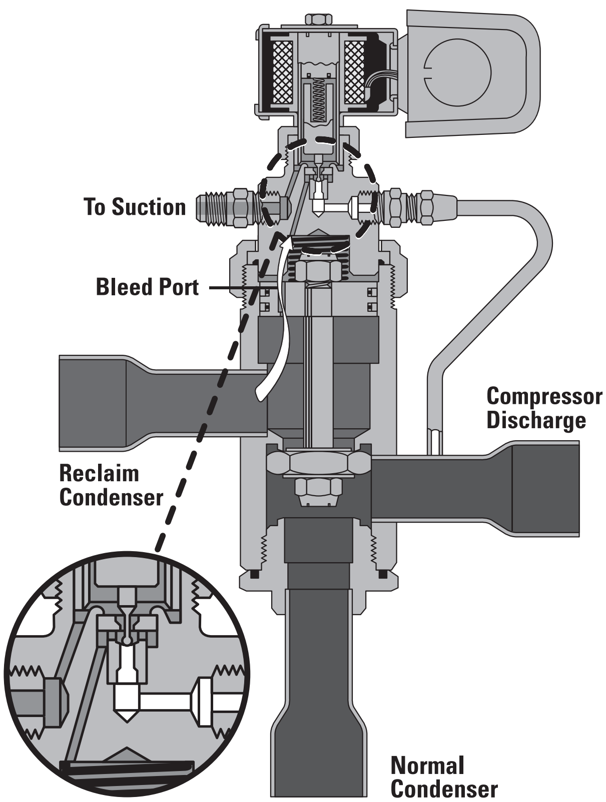

Figure 1:
3-Way Heat Reclaim Valve

Reclaim Coil Pump Out

When the building temperature is satisfied, the HVAC controls will de-energize the 3-way valve solenoid coil, bringing the system out of the heat reclaim mode. Proper refrigerant management requires transferring the residual refrigerant remaining in the idle reclaim coil back to the active part of the system. More importantly, if not removed, the remaining mixture of saturated liquid and vapor in the heat reclaim coil will be at the ventilating supply air temperature. When heat reclaim is required again, the superheated discharge vapor entering the reclaim coil will mix with the lower temperature liquid refrigerant remaining in the reclaim coil tubes, causing it to boil quickly. If the temperature difference between the entering discharge vapor and the idle saturated liquid is great enough, the resulting agitation may cause liquid hammer.

There are two ways in which the heat reclaim coil can be pumped out. Both methods transfer the high pressure refrigerant from the heat reclaim coil to the suction header on the compressor rack. Pump out is complete when the pressure in the heat reclaim coil is equal to the suction header pressure. There is always some refrigerant remaining in the idle heat reclaim coil. Pump out only allows the higher pressure refrigerant in the heat reclaim coil to vent to the suction manifold, resulting in a drop in heat reclaim coil pressure. A substantial amount of the refrigerant is removed in the process, however some vapor will remain.

Method #1: Pumping out the refrigerant from the idle heat reclaim coil can be accomplished with the Type “B” 3-way valve, which has a bleed orifice in the main piston (refer to Figure 1). De-energizing the Type “B” pilot solenoid coil will cause the main piston to shift upwards, stopping the refrigerant flow to the heat reclaim port. The refrigerant in the heat reclaim coil will flow through the bleed orifice, into the upper pilot port, and proceed to the suction header. The bleed orifice provides the necessary flow path for the refrigerant to re-enter the system, eliminating the need for a dedicated pump-out solenoid valve.

Method #2: The Type “C” 3-way valve does not have a bleed orifice in the main piston. If it is used, a dedicated pump out solenoid will be required. This would typically be a normally open solenoid valve, wired in parallel with the 3-way valve, and piped from the outlet of the reclaim coil to the suction header. With the 3-way valve de-energized (no heat), the normally open pump out solenoid would be de-energized as well, allowing refrigerant to flow from the heat reclaim coil to the suction header.

In most instances the inlet to the heat reclaim coil is located at the top, with the outlet located at the bottom. This allows for the free draining flow of refrigerant, and reduces the possibility of any oil logging. If any liquid refrigerant is present in the heat reclaim coil during the pump out mode it will drain to the bottom, or outlet fitting. With the Type “B” 3-way valve, pump out allows refrigerant to flow from the valve’s outlet fitting (inlet of the heat reclaim coil) to the suction header. Because of the free draining nature of the heat reclaim coil, liquid refrigerant and/or lubricant should never be present at this point. Therefore, it would be unusual to experience any liquid return to the suction header with this method of pump out. Nevertheless, it is always a good practice to install a restriction to the pump out line upstream of the suction header. The resulting pressure drop will expand any liquid refrigerant that might be present into a vapor before entering the compressors.

With the Type “C” 3-way valve, pump out allows refrigerant flow from the reclaim coil’s outlet fitting (bottom of the heat reclaim coil) to the suction header.

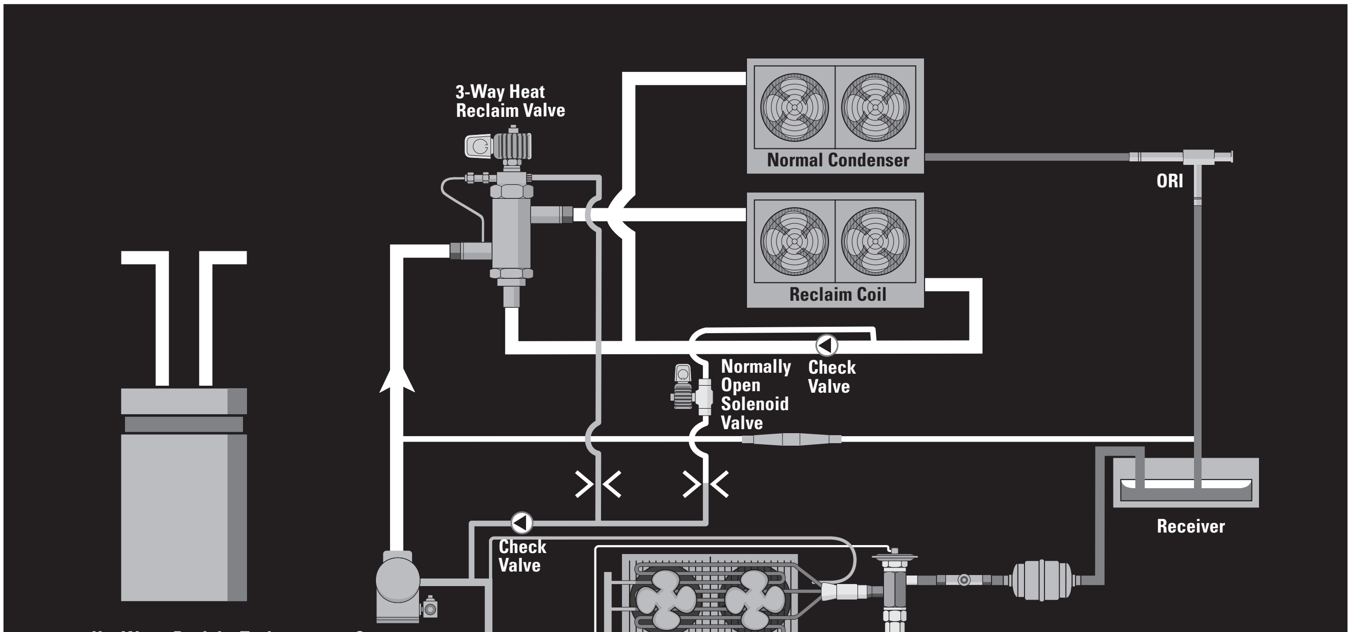

Figure 2
Series Heat Reclaim

If there is any liquid present in the heat reclaim coil, it will accumulate at this location during the pump out mode. Therefore, with this method of pump out, a restriction in the pump out line is a necessity.

If there is no refrigerant flow between two points in the system, the refrigerant will migrate to the colder of those two points. For example: once the idle heat reclaim coil has been pumped out, there is a connection between the suction header and the heat reclaim coil through the pump out line. If there is a possibility that the temperature of the idle heat reclaim coil (or hot water reclaim tank) would be lower than the saturated suction temperature of the compressor rack, the refrigerant will migrate to the colder heat reclaim coil. A tight seating check valve should be added to the pump-out line to reduce refrigerant migration.

Note: you may think that a check valve would completely eliminate the possibility of refrigerant migration. In theory it should. In reality, all check valves will experience some amount of leakage, albeit minor. The type of seating material used will also influence the check valve's ability to provide as close to a leak free seal as possible. A metal to metal seat may have an accepted leak rate of 750 cc/min, while a synthetic to metal seat may be as low as 10 cc/min.

Series Heat Reclaim

Figure 2 shows the series method of heat reclaim piping. The finned tube heat exchanger used in this method is properly termed a heat reclaim coil; it is not a heat reclaim condenser. Its purpose is to de-superheat the discharge vapor only, and therefore its Btu capacity is much less than that of the outdoor condenser; somewhere between 30% to 50% of the total heat of rejection. In the series method, when the 3-way valve shifts into the heat reclaim mode, the refrigerant discharge vapor flows from the compressor, through the tubes in the heat reclaim coil, and finally to the inlet of the normal condenser. Heat from the refrigerant discharge vapor is transferred to the ventilating air, and is subsequently distributed through the ducting system, providing heat for the building. Because the heat reclaim coil only removes some portion of the superheat content of the refrigerant discharge vapor, the normal condenser must remove the remainder of heat content necessary to convert the vapor into liquid refrigerant.

A hot water reclaim tank (also shown in Figure 2) can be substituted for the heat reclaim coil, and will serve to preheat the water supply before entering the hot water heater. Up to 60% of the heat normally rejected at the condenser can be transferred to the water, resulting in an 80°F degree temperature increase, to approximately 65 gallons per hour (per 10 tons of refrigeration capacity).

Note: Some applications are using plate heat exchangers in place of hot water reclaim tanks.

Since the series method of heat reclaim utilizes a heat exchanger sized to de-superheat the discharge vapor only, there should never be any liquid refrigerant in the heat reclaim coil. While some amount of vapor

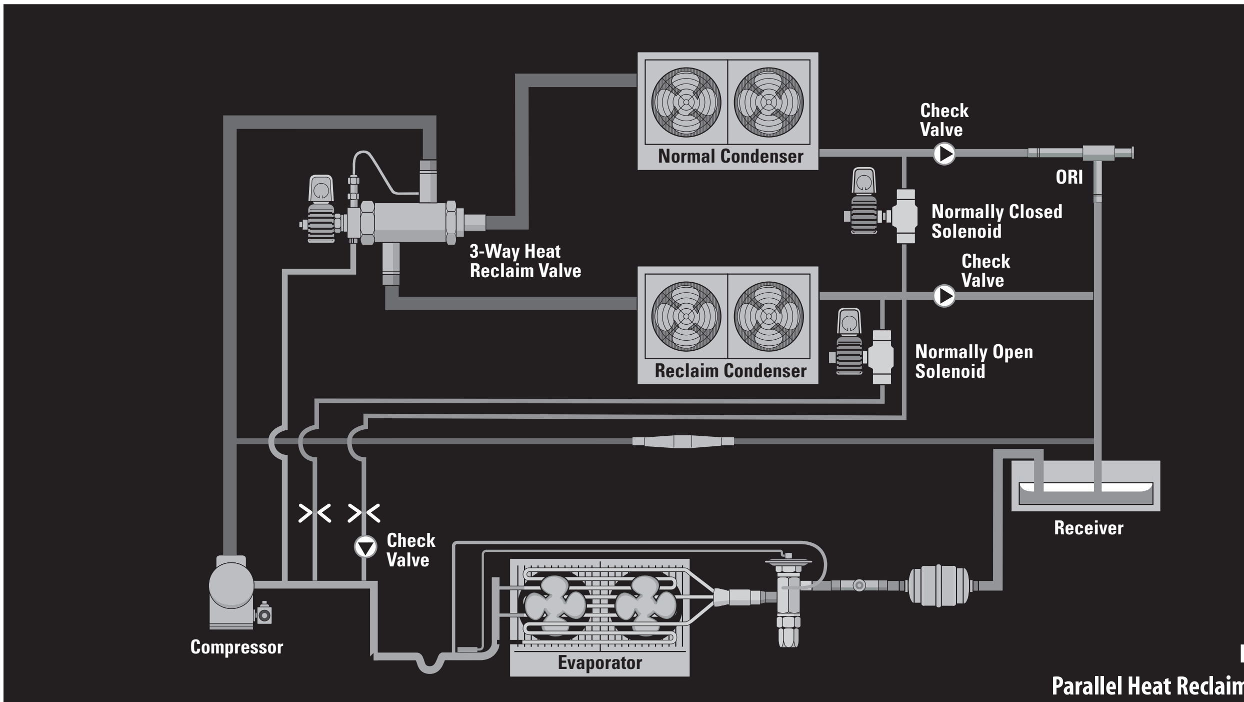

Figure 3
Parallel Heat Reclaim Piping

might be expected to condense during the pump-out period, it should never be present at the point of pump-out (heat reclaim coil inlet). As mentioned above, a restriction in the pump-out line should not be required; however it is always a good practice to install one.

Parallel Heat Reclaim

There are some applications where a parallel piped heat reclaim system may offer some benefit. This is an either/or method which employs two condensers of equal capacity; one outdoors, and the other in the ventilation system (see Figure 3). They are both sized for the entire heat of rejection load. During the normal condensing mode the outdoor condenser provides the heat transfer necessary to handle the entire heat of rejection. Whereas the smaller heat reclaim coil in the series application only de-superheats the refrigerant, the full sized reclaim condenser in the parallel application completely condenses the refrigerant. The entire heat of rejection load is transferred to circulating ventilation air. This will yield a larger heating capacity for the building, possibly eliminating the need for a second stage of heat.

The very thing that gives the parallel method a greater heating capacity offers its greatest challenge; being an either/or application. During the outdoor condensing mode the heat reclaim condenser is idle. This means that it receives no refrigerant flow, and it contains only a minimal amount of refrigerant vapor and liquid. When the requirement for heat requires the 3-way valve to shift, simultaneously the refrigerant flow ceases to the normal condenser, and starts to the reclaim condenser. There will be some time lag before the relatively empty reclaim condenser contains enough refrigerant to resume the steady supply of liquid refrigerant to the receiver.

Because of this lag, the ability to quickly pump out the idle condenser is desirable. To accomplish this it is recommended that the Type C 3-way valve be used, along with a dedicated pump-out solenoid valve for each condenser. Again, a restriction should be used in the pump-out line to prevent liquid floodback. Even with this arrangement the compressor(s) will fill the now active reclaim condenser much faster than the pump out solenoid can drain the idle normal condenser. During the transition from one condenser to the other, the level of refrigerant in the receiver will be drained faster than the functioning condenser can replenish it. Without a sufficient refrigerant charge the receiver may lose its liquid seal before system equilibrium is restored, temporarily compromising the ability to provide vapor free liquid to the TEVs.

There is yet another potential problem with the parallel method; the accumulation of cold saturated liquid in the idle roof condenser during the heat reclaim mode. Upon resumption of the normal condensing mode, superheated discharge vapor will be re-introduced into the outdoor condenser. If the quantity of low temperature saturated liquid is excessive, the resulting rapid expansion from the sudden temperature increase may cause liquid hammer. Again, there will be the possibility of piping and/or component failure. The potential is

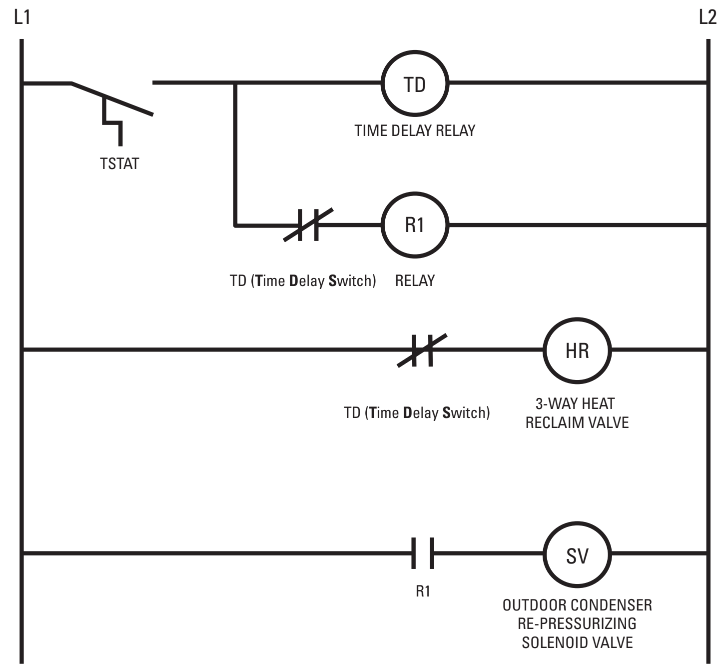

Figure 4
Wiring for Condenser Pressurizing Valve

more severe than in the series reclaim method, as there will be areas of the country that will experience extremely low ambient temperatures during the winter months. The amount of liquid hammer will be magnified as the difference between the discharge vapor temperature and ambient temperature increases.

In the parallel method there are two possibilities for liquid refrigerant to be present prior to resuming normal condensing:

1) Any time the ambient (outdoor condenser location) is colder than the suction vapor temperature the suction vapor will want to migrate to the colder condenser. The colder ambient will cause some of the migrated vapor to condense, resulting in an idle condenser that has a substantial amount of cold liquid in it. A tight seating check valve in the pump-out line is required to reduce the potential for migration.

2) Pump out of the idle condenser is complete when its pressure is reduced to compressor rack suction pressure. When pump out is complete, some refrigerant vapor will remain in the idle roof condenser. If the ambient temperature is lower than the compressor rack's saturated suction temperature, some of the remaining vapor will condense as its temperature is reduced in the colder ambient location. Since it will be at the pressure corresponding to the ambient, the pressure will be lower than the common suction pressure; hence it will remain in the condenser.

One method of eliminating the potential for any liquid hammer is to slowly re-pressurize the roof condenser with an auxiliary solenoid valve before shifting back to the normal condensing mode. This can be accomplished with a time delay relay (TD), a standard relay (R1), and a small normally closed solenoid valve (SV).

The store thermostat would close on a rise of temperature, energizing the time delay relay (TD) and relay (R1) when the store temperature reaches its set point (see Figure 4). A normally open switch in relay R1 will close, energizing the pressurizing solenoid valve. After the time delay relay TD times out (a minute or two), it will open the two normally closed TD switches, de-energizing the 3-way valve and R1. This will bring the system back into the normal condensing mode and de-energize the pressurizing solenoid valve. The slow introduction of discharge vapor to the condenser will eliminate rapid temperature shock and liquid hammer as the system reverts to the normal condenser mode.

Service Issues

When using the Type B 3-way valve, there may be rare occasions where the pilot connection to the suction header is frosted. Liquid refrigerant experiencing a reduction in pressure through the small pilot port will flash, producing a refrigeration effect. If the saturated suction temperature (or corresponding pressure) is at or below 32°F, the flashing will result in the pilot line frosting. First inclination would be to suggest that this is the result of the refrigerant in the idle heat reclaim coil pumping out through the orifice in the main valve piston.

When using the Type B 3-way valve, the refrigerant flowing through the main piston orifice during pump-out is coming from what would be the inlet of the heat reclaim coil. In the series method, the heat reclaim coil's function is to de-superheat the discharge vapor only. As such, there should not be any liquid present in this coil. Even if some small amount of liquid refrigerant were present during pump-out, the free draining nature of the heat reclaim coil (inlet at top, outlet at bottom) would prevent it from accumulating near the point of pump-out (the inlet). Therefore; if any liquid refrigerant is present during pump-out, gravity will force it to accumulate at the bottom of the heat reclaim coil, away from the point of pump-out.

Normal operation of the series heat reclaim application during pump-out will not allow for any liquid refrigerant at the point of pump-out; only vapor should be present at the point of pump-out. Expanding a high pressure vapor to a low pressure vapor does not produce a refrigeration effect. Therefore, a normal operating series heat reclaim system should never have frost on the pilot connection to the suction header.

So what is the answer? Frosting of the pilot line can only happen if the heat reclaim coil is nearly full of liquid refrigerant during pump-out. There is a check valve at the outlet of the heat reclaim coil to retard refrigerant backflow into the heat reclaim coil during the condensing mode. If the check valve developed substantial seat leakage, it would provide the reclaim coil with an unlimited source of refrigerant vapor which could completely fill the HR coil with liquid refrigerant over time. Because the air handling fan is normally on, air at approximately 75ºF will be flowing through the heat reclaim coil. This will cause the superheated discharge vapor leaking into the heat reclaim coil to condense, and experience a drop in pressure. This then makes room for additional vapor to enter the heat reclaim coil, which too will condense and experience a drop in pressure, making room for additional vapor to enter the heat reclaim coil, which too will condense and experience a drop in pressure…until it is nearly completely full of liquid refrigerant.

There is another potential problem when using the series heat reclaim method. Many refrigeration systems use condenser flooding valves to maintain a minimum head pressure during low ambient conditions. An ORI or A8 inlet pressure regulator (holdback valve) at the condenser outlet maintains a constant condensing pressure. A second valve (receiver pressurization valve) is located between the discharge line and the receiver, and maintains a constant receiver pressure. This can either be an A9 outlet pressure regulator, which maintains a constant receiver pressure based upon its set point, or an ORD differential check valve which maintains a constant differential between the discharge pressure and receiver pressure. A tee in the main discharge line will supply discharge vapor to the receiver pressurization valve (see Figure 2).

There is a natural pressure drop that exists between the outlet side of that tee and the inlet of the holdback valve; a combination of pressure drop in the piping and components (oil separator, ball valves, 3-way valve, ORI/A8 valve, etc.), and pressure drop in the condenser. The A9 valve's setpoint (receiver pressure) is independent of the setting of the holdback valve, or the accumulated pressure drop between the outlet of the tee and the inlet of the holdback valve. The ORD's function in maintaining receiver pressure is very dependent on this pressure drop and the holdback valve's setpoint.

For example: The ORI/A8 valve is set to maintain a 185 psi pressure at its inlet. Receiver pressurization is achieved with an ORD-4-20 differential check valve. It begins to open at a 20 psi differential, and flows its rated capacity at a 30 psi differential. It begins to close at a 14 psi differential. If the pressure drop between the outlet of the tee and the inlet of the holdback valve during the normal condensing mode is 5 psi, then the pressure at the ORD inlet would be 190 psi (ORI/A8 setting plus the 5 psi pressure drop). The receiver pressure would be at 170 psi (190 psi discharge pressure minus the 20 psi differential at which the ORD opens).

The series method of heat reclaim offers the greatest potential for total pressure drop between the discharge line and the inlet of the holdback valve. It includes the combined total pressure drop from the reclaim coil and the normal condenser, plus the associated piping and components for both. In an application with long piping runs between the heat reclaim coil and the condenser, the possibility exists where the total pressure drop could exceed the point at which the ORD-4-20 differential check valve begins to close. In extreme cases, the total pressure drop might exceed the point at which the valve would start to open, causing it to remain open indefinitely.

For example: The holdback valve is again set for 185 psi, and the pressure drop between the discharge line and the inlet of the holdback valve is 17 psi. The pressure at the ORD inlet would be 202 psi (ORI setting plus the 17 psi pressure drop), and the receiver pressure would be 182 psi (202 psi discharge less the 20 psi differential at which the ORD opens). Because of the pressure drop, there will always be a minimum 17 psi differential between inlet and outlet of the ORD during the heat reclaim mode. As the need for head pressure control subsides, the holdback valve would cease holding back (wide open operation) and the receiver pressurization valve would throttle closed. However, with a 17 psi pressure drop, the minimum 14 psi differential at which the ORD-4-20 starts to close will never be realized. The differential check valve would continually bypass hot gas to eliminate the 3 psi difference between the available pressure drop (17 psi) and the point at which it starts to close (14 psi). This would only be a problem when operating conditions do not require the holdback valve to maintain a minimum condensing pressure, yet heat reclaim is required. This would be an infrequent condition when using heat reclaim to provide the building's heat. Because of hot water reclaim's year around demand, it would offer more potential for this circumstance to occur.

To realize the negative impact of this situation requires an understanding of the dynamic condition that exists within the receiver. It is true that in a static condition a vessel containing refrigerant will be at a saturated condition relative to the ambient temperature. In a dynamic condition such as an operating system, it is common for the entering refrigerant to be subcooled. Given the fact that a receiver is a poor heat transfer device, where does the required heat gain come from to bring this refrigerant to a saturated condition? It doesn't…in a dynamic system there isn't enough time for the required amount of heat to transfer through the receiver wall and heat the liquid to a saturated condition. The point of interface between the liquid and vapor present in the receiver will be at a saturated condition. Below this point there will be a stratification of liquid, with the very bottom of the receiver containing the coolest liquid. The closer the liquid temperature is to the evaporating temperature, the more efficient it will be in transferring heat from the refrigerated space. Unnecessarily adding hot gas to the receiver will increase the leaving liquid temperature, resulting is a less efficient use of the liquid. This will require an increase in refrigerant mass flow, resulting in lower equipment efficiency and higher operating costs.

If receiver pressurization is maintained with an ORD differential check valve, it should be selected with a rated closing point which is greater than the combined pressure drop between the discharge tee outlet and the receiver inlet. If an A9 outlet regulator is used for receiver pressurization, its normal operation will not be impacted by excessive pressure drop between the discharge line and the receiver inlet. As an outlet pressure regulator, its function is to provide a constant receiver pressure, and its ability to maintain this is not impacted by the pressure difference between its inlet and outlet.

Heat Reclaim: Is it Viable?

There are some who suggest that using heat reclaim to provide a source for building heat is not a huge energy saver. The argument goes something like this: higher condensing pressures are required to generate sufficient discharge temperatures from which heating can be achieved. While building heat reclaim will eliminate the need for the additional equipment and energy source necessary for heating, the required higher condensing pressures reduce compressor efficiency. In the winter, the energy saved from operating at low condensing temperatures will be greater than the gain from heating the store with heat reclaim at higher condensing temperatures. Perhaps there is some validity to this argument. However; it doesn't hold true when applied to a hot water heat reclaim application. While operating at the lowest possible ambient in the winter, there will still be some heat content that can be transferred from the discharge vapor to the water in a hot water reclaim tank. Because hot water is in demand year around, its benefit can be maximized during the warmer months.

Today's climate of energy conservation demands that every effort be made to increase system efficiency. The compression process, while an essential component of the vapor-compression cycle, is nothing more than a means to an end. It is a necessary evil which requires the expenditure of energy to accomplish. Some of that energy is converted to heat; a waste byproduct. Heat reclaim takes advantage of this existing situation and extracts some benefit from it. Using heat reclaim to heat a building may not be the most efficient choice; however there is demand for hot water year round, and heat reclaim converts waste into a reduction in energy consumption.

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