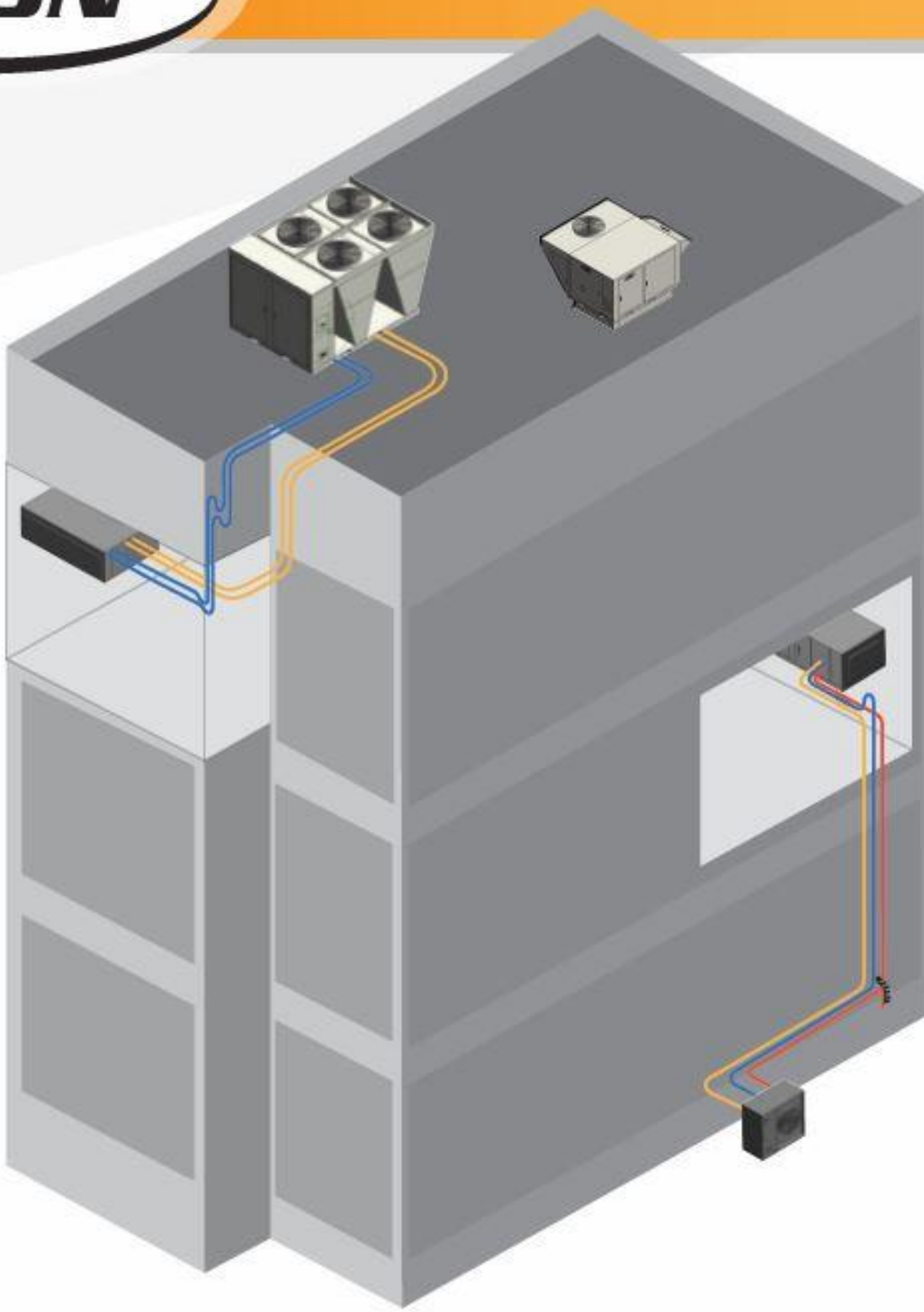

DX Handbook

1 Table of Contents

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Tables:

2 Fundamentals of HVAC Refrigeration

Heating, ventilation, and air conditioning systems (HVAC) provide the basic functions of cooling air, heating air, adding or removing water vapor, mixing and filtering air streams, and moving air. The purpose is to maintain a suitable environment for comfort, health, productivity of the occupants and processes within the environment. This manual focuses on the mechanical refrigeration system of the HVAC system only.

A large part of conditioning a space is accomplished by heat transfer. Heat transfer is always from the hotter substance to the colder substance. In an HVAC system, the majority of heat transfer occurs in the evaporator coil and the condenser coil. Both coils have a fan blowing air across them so that the two substances in heat transfer are air (blowing across the coil) and refrigerant (inside the coil). In cooling mode of operation, the hot return air (or outside air) being conditioned transfers its heat to the refrigerant inside the evaporator coil; and in a different part of the refrigerant cycle, the refrigerant in the condenser coil transfers its heat to the ambient air blowing through the coil. The arrows in the figure below show the pathway of the refrigerant in the system.

Another important aspect of the refrigeration cycle is pressure. While heat transfer takes place in the evaporator & condenser coils, considerable change in pressure takes place in the compressor and the thermostatic expansion valve (TXV). The compressor changes the low pressure vapor to a high pressure vapor. The TXV changes the high pressure liquid to a low pressure liquid, and it also controls the amount of refrigerant entering the evaporator coil.

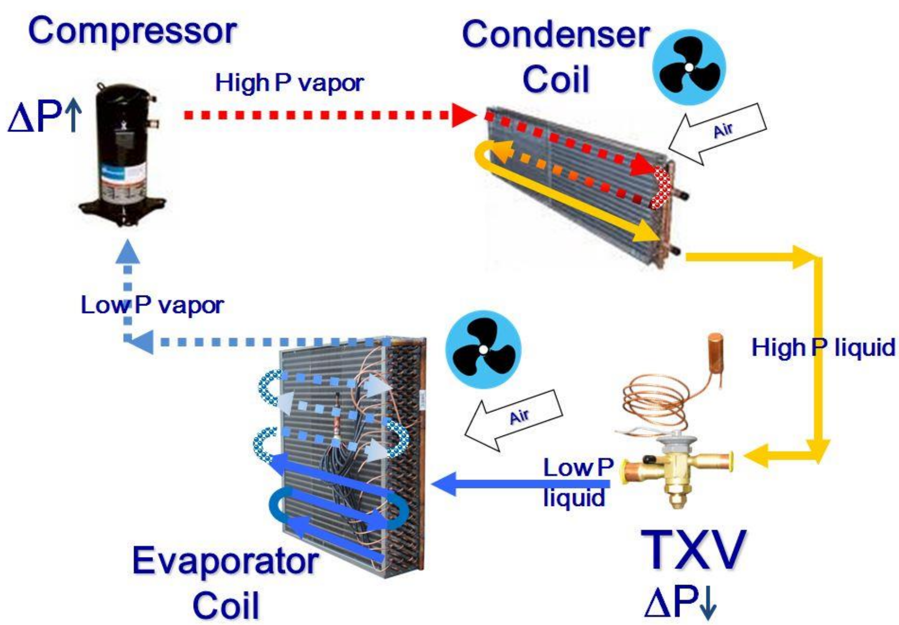


Figure 1 - Refrigeration Cycle

In AC Mode of Operation:

Evaporator coil - the heat transfers from the warm air to the refrigerant inside the evaporator coil. The air coming out of the evaporator coil is cooler than the air entering it. The values in the figure below are just one example situation.

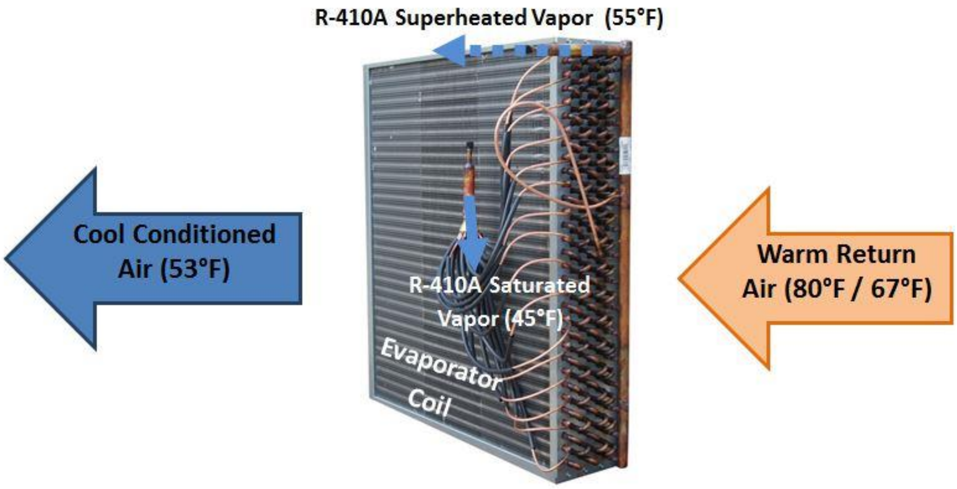


Figure 2 - Evaporator Coil Heat Transfer

Condenser coil - the heat transfers from the hot refrigerant inside the condenser coil to the air flowing through the coil. The air coming out of the condenser coil is warmer than the air entering it.

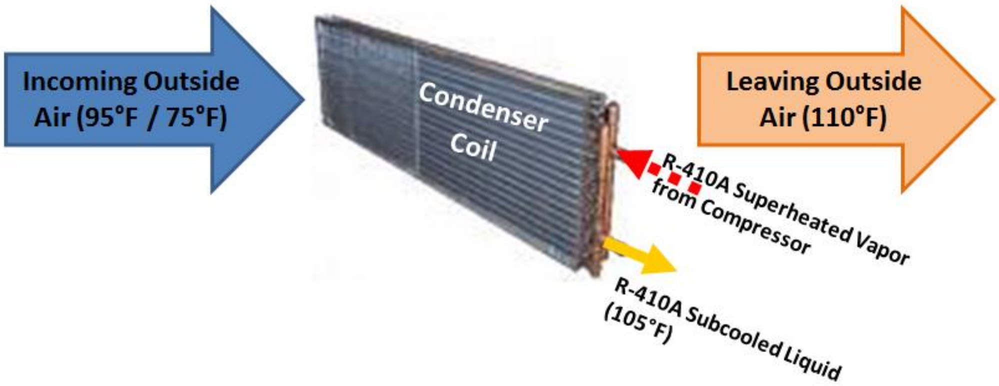


Figure 3 - Condenser Coil Heat Transfer

Latent Heat vs Sensible Heat:

A majority of the heat transfer in the refrigeration system is due to latent heat from the refrigerant changing states. Latent heat is the heat that it takes to change the state of a substance. For example, in the evaporator coil, the refrigerant changes from a liquid state into a vapor state (evaporation) and in the condenser coil, the refrigerant changes from a vapor state to a liquid state (condensation). The heat energy that it takes to change states (latent heat) is much larger than the heat it takes to change the temperature of that same substance within a state (sensible heat).

The properties of water, shown below, help to illustrate the principles of latent and sensible heat. Water exhibits all the properties of a refrigerant and all are familiar with the freezing and boiling points of water.

Sensible Heat - Specific heat - 1Btu raises 1 lb of water 1°F at 14.7 psi

Latent Heat of Fusion of water - 144 Btu changes 1 lb of 32°F water to 32°F ice at 14.7 psi

Latent Heat of Vaporization of water - 970 Btu changes 1 lb of 212°F water to 212°F vapor at 14.7 psi

Notice the changes of state occur at one temperature and one pressure - the freezing/melting point or the boiling/condensing point at a specific pressure

In the Evaporator - refrigerant liquid evaporates into a vapor - Latent Heat of Vaporization of R-410A - adding 92.7 Btu changes 1 lb of 40°F liquid to 40°F vapor at 118 psig

In the Condenser - refrigerant vapor condenses into a liquid - Latent Heat of Condensation of R-410A - removing 60.2 Btu changes 1 lb of 120°F vapor to 120°F liquid at 418 psig

Definition of one ton of refrigeration
1 ton of refrigeration = amount of heat required to melt 1 ton of ice in one day
Expressed in an hourly rate:

1 ~~ton ice~~	2000 ~~lb ice~~	144 Btu		=	12,000 Btu
	1 ~~ton ice~~	1 ~~lb ice~~	24 hr		1 hr

So 1 ton of refrigeration = 12,000 Btu/hr

R-410A Pressure-Enthalpy Diagram:

Understanding the Pressure-Enthalpy diagram can help to understand the different states that the refrigerant goes through in the refrigeration cycle. The shaded portion of the diagram is the saturated state. Within that envelope the refrigerant is saturated, meaning that it contains a mixture of both liquid and vapor. The edge of the envelope is where the refrigerant is either 100% liquid (on the left) or 100% vapor (on the right). The unshaded area to the left of the 100% liquid state is subcooled liquid. The unshaded area to right of the 100% vapor state is superheated vapor. Sub-cooled means the refrigerant is 100% liquid and the *amount* of sub-cooling is the difference in temperature between the saturated temperature at a specific pressure and the measured temperature of the liquid. Superheated means the refrigerant is 100% vapor and the *amount* of superheat is the difference in temperature between the measured temperature of the vapor and the saturated temperature at a specific pressure. The refrigeration cycle takes advantage of the latent heat energy so most of the refrigerant cycle is within the envelope where the refrigerant is changing states from either a liquid to a vapor or from a vapor to a liquid.

R-410A is a blend of R-32 and R-125. It is a near azeotrope because the bubble point and dewpoint are very close in temperature. So close that for all practical purposes they can and are treated as the same in refrigeration cycle analysis.

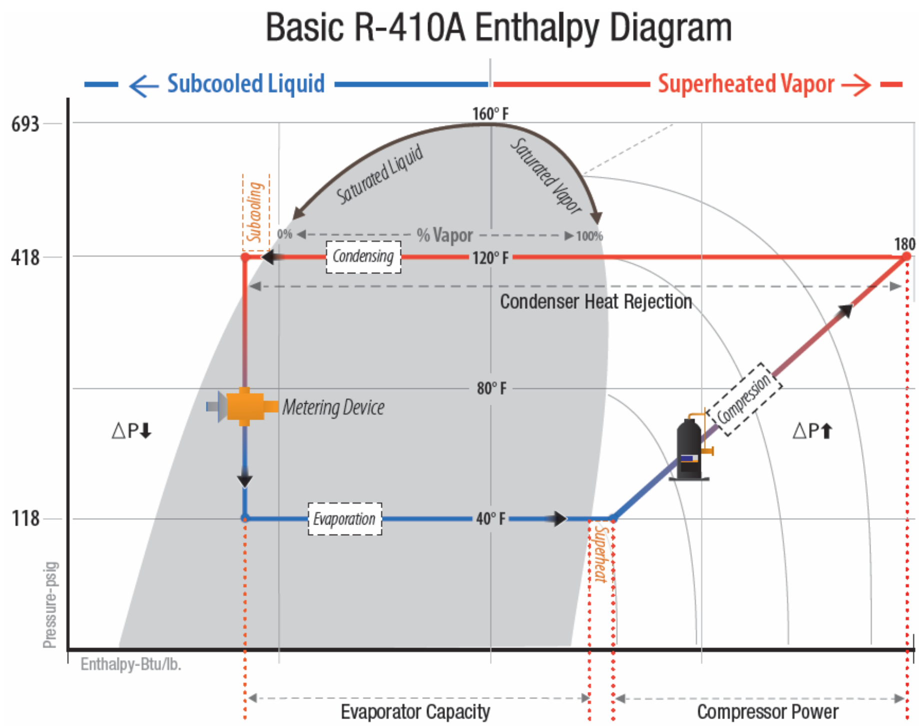


Figure 4 - Basic R-410A Pressure-Enthalpy Diagram

Saturation Temperature = boiling or condensing temperature at a specific pressure

(at 418 psig sat temp = 120°F and at 118 psig sat temp = 40°F)

Sub-cooling = saturation temperature - measured temperature of liquid = 10°F typical

Superheat = measured temperature of vapor - saturation temperature = 15°F typical

For a more in depth technical discussion, see Sporlan's Pressure-Enthalpy Chart document .

The numbers 1-4 indicate the different physical states of the refrigerant fluid as it moves through the system.

- **State 1** is a high pressure, high temperature superheated vapor. The compressor is designed to handle only vapor so low pressure, low temperature superheated vapor enters the compressor and high pressure, high temperature superheated vapor leaves the compressor.
- **State 2** is a sub-cooled liquid. In the condensing coil, enough heat transfers from the refrigerant that the superheated vapor turns to a saturated vapor (sensible heat), then to a saturated liquid (latent heat), and then to a sub-cooled liquid (sensible heat). As it exits the condenser coil, it is a high pressure, high temperature sub-cooled liquid.
- **State 3** is a low pressure, low temperature liquid. The thermostatic expansion valve must be fed sub-cooled liquid to operate properly, so high pressure, high temperature sub-cooled liquid enters the TXV and a low pressure, low temperature liquid-vapor mixture exits the TXV. Notice the TXV outlet in **Figure 4** is in the two phase region of the pressure-enthalpy diagram.
- **State 4** is a superheated vapor. In the evaporator coil, heat transfers from the air to the refrigerant such that the liquid-vapor mixture entering the evaporator turns to a saturated vapor (latent heat) and then to a superheated vapor (sensible heat). As it exits the evaporator coil, it is a low pressure, low temperature superheated vapor.

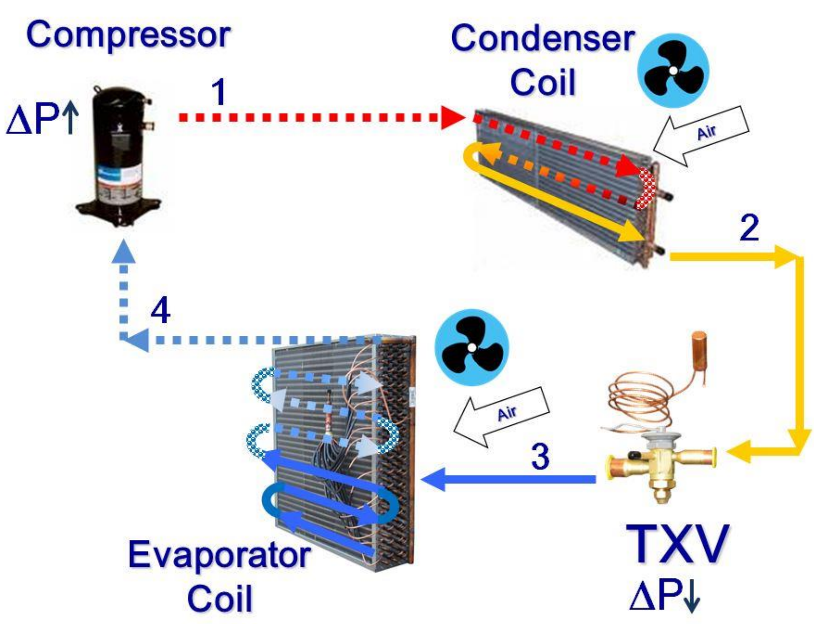


Figure 5 - Refrigerant Flow Path

3 Basic Components in a Refrigeration System

The high side of a refrigeration system is the discharge side (outlet) of the compressor. The high side includes the discharge line, the condenser coil, the liquid line, and the inlet to the TXV. It is called high side because the temperatures and pressures are higher on the discharge side of the compressor. The compressor pressurizes the refrigerant in the system.

The low side of a refrigeration system is the suction side (inlet) of the compressor. The low side includes the outlet of the TXV, the suction line, the evaporator coil, and the inlet to the compressor. It is called low side because the temperatures and pressures are lower on the suction side of the compressor.

One thing that can seem confusing is the vapor state is at a cold temperature and the liquid state is at a warm temperature. That goes against logic if the refrigerant were at the same pressure in both cases. The reason the vapor state is a cold temperature is because the pressure is low, so the refrigerant boils at a lower temperature. In a similar manner, the liquid condenses at a high temperature due to a high pressure and therefore high condensing temperature.

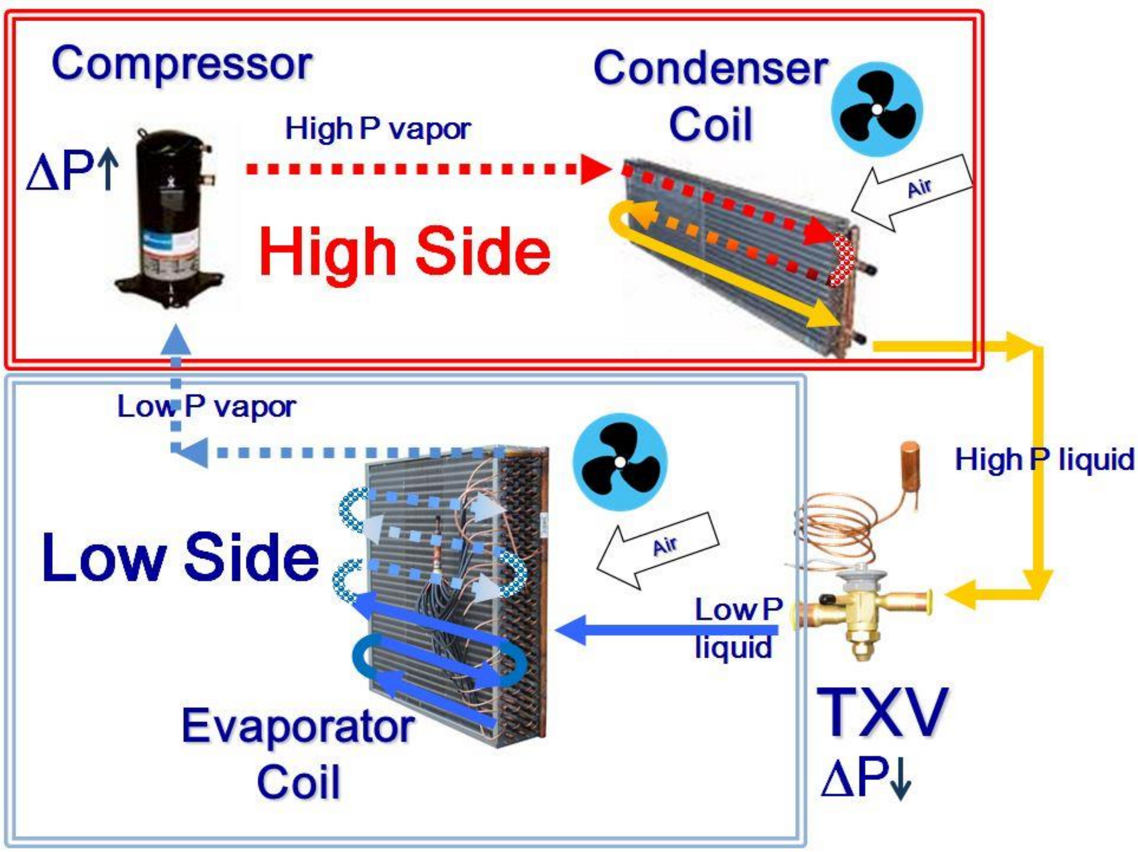


Figure 6 - High and Low Side

4 Compressor Types

The purpose of the compressor is to pump refrigerant vapor & increase the pressure of the refrigerant. A compressor is designed for vapor only; liquid entering a compressor is not only undesirable, but can result in compressor failure.

Compression ratio = absolute discharge pressure / absolute suction pressure

4.1 Copeland ScrollTM - R-410A

Figure 7 - Copeland Scroll Compressor

See HowScrollWorks.swf for an explanation of how the scroll compressor operates. A brief summary is the refrigerant gas enters the outer spiral of the scroll and as it travels through to the inner part of the spiral, it gets more and more pressurized until it is finally discharged through the center of the spiral.

Top fixed scroll Bottom orbiting scroll

Figure 8 - Scrolls

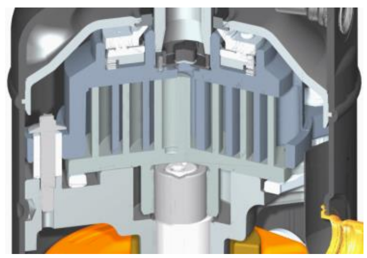


Figure 9 - Copeland Scroll Compressor Schematic

Each compressor has its own operating envelope. The operating envelope represents operating conditions that are safe for the compressor motor.

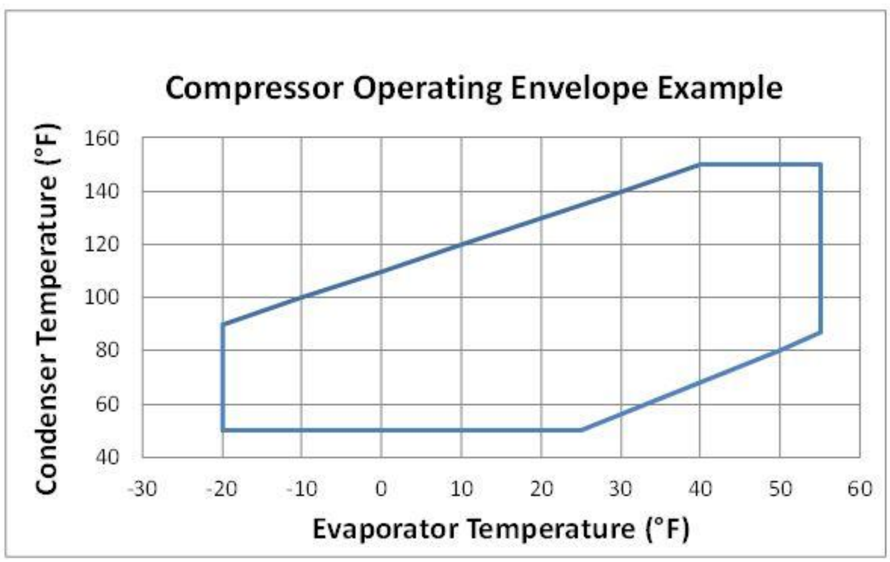


Figure 10 - Example Compressor Operating Envelope

4.2 Copeland Scroll™ Two-Stage - R-410A

Two-stage scroll compressors provide two stages of capacity, typically 67% and 100%, for more energy efficient part load operation.

The Copeland Scroll two-stage compressor modulates between a part load and a full load capacity setting. Two internal bypass ports enable the compressor to run at the part-load capacity during times when only part-load heating or cooling is needed. When demand increases, the modulation ring is activated, sealing the bypass ports and instantly shifting capacity to 100%. Running for longer periods at part load capacity can lower the humidity inside the building and also allows the HVAC system to operate more quietly.

[Copeland marketing for two-stage compressor]

[How Two-Stage Scrolls Work]

Figure 11 - Copeland Scroll Two-Stage Compressor

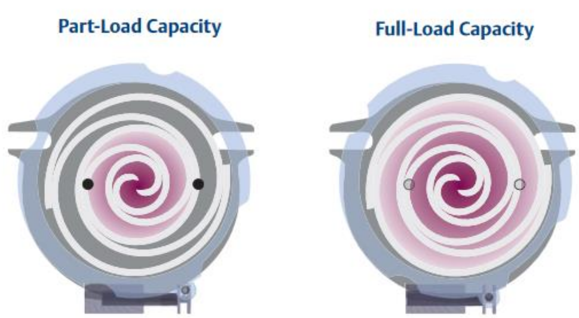


Figure 12 - Copeland Scroll Two-Stage Bypass Ports
Figure source

See Copeland Bulletin AE4-1311 for more detailed information.

4.3 Copeland Scroll™ Digital - R-410A

Digital or variable capacity scroll compressors provide 10-100% modulating capacity for load matching cooling and heating and more energy efficient part load operation.

These compressors require a 1-5 VDC control signal to control compressor capacity modulation. The capacity is modulated by a solenoid unloader. The signal tells the solenoid how many seconds per cycle to unload the compressor. In the unloaded state, the scroll elements separate and thus create 0% capacity. When using customer provided controls any variable capacity compressors should run at 100% for 1 minute when starting.

Figure 13 - Copeland Scroll Digital Compressor

The capacities start at 3 tons and go to 15 tons. After that, tandem compressors can be provided up to 30 tons. ZPD34 through ZPD91 compressors require external piping for the solenoid valve, which is factory installed (See **Figure 15**). ZPD103 through ZPD182 compressors include an oil sight glass and oil service connection (See **Figure 14**).

See Copeland Bulletins AE4-1395 & AE8-1328 for more detailed information.

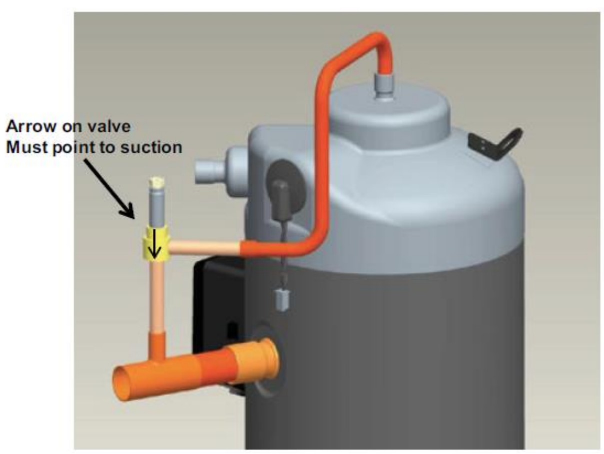


Figure 15 - Copeland Scroll™ Solenoid Valve Piping for ZPD34-ZPD91

Figure 14 - Compressor Oil Sight Glass & Oil Service Connection

Figure 16 - Digital Compressor Solenoid

Unloader Solenoid Valve is the part of the variable capacity compressor that modulates the capacity using a cycle time with varying times of loaded state verses unloaded state. This external piping is only necessary on compressors ZPD34 through ZPD91. The larger digital compressors will have the bypass capability internal to the compressor.

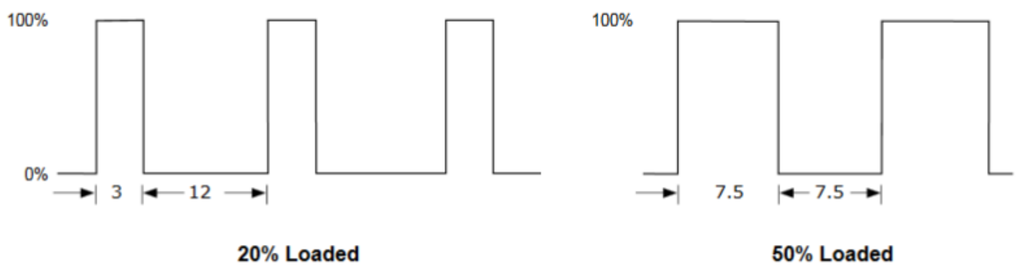


Figure 17 - Concept of Modulating Compressor Cycle Time

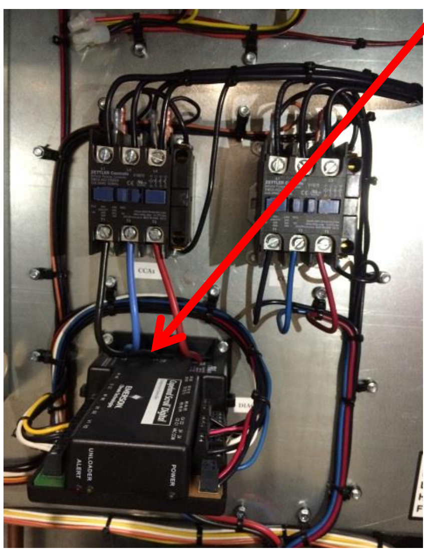


Figure 18 - Copeland Scroll Digital Compressor Diagnostics

Copeland Scroll Digital Compressor Diagnostics modulates the solenoid based on capacity demand signal. It also protects against excessive discharge temperatures, low flow conditions, and operation under fault conditions. The power LED shows green to indicate a 24V signal. The Unloader LED shows yellow to indicate the solenoid is energized. The Alert LED has a flash code that is interpreted by counting the number flashes (1-9). The flash code guide is mounted on the access door next to the wiring diagram. When the AAON Touchscreen Controller or WattMaster VCB-X & VCC-X controls are used, this component is not necessary because its functionality is built into the controllers.

4.4 Copeland Scroll™ Variable Speed Compressor - R-410A

This is a scroll compressor that modulates capacity using a permanent magnet motor control drive connected to the compressor. The speed of the compressor modulates from 1500-7200 rpm. That corresponds to a capacity turn down of about 21%.

The motor control drive has an oil boost feature that increases the motor speed for a period of time to ensure the compressor is getting oil return.

Figure 19 - Copeland Scroll Variable Speed Compressor

These are ZPV compressors with EV variable speed drives.

Copeland Variable Speed Compressor Information

4.5 Copeland Scroll™ Tandem - R-410A

Tandem compressors are two compressors piped together with only one suction line and one discharge line. The Copeland Tandem Scroll compressors are typically two compressors with the same capacity. There are different variables possible in tandem compressors. In a tandem set there are a couple possibilities:

1. Both standard on/off scroll compressors
2. One standard on/off and one digital scroll compressor

Figure 20 - Copeland Scroll Tandem Compressor

4.6 Bitzer Orbit for Variable Speed Drive - R-410A

This is a scroll compressor that modulates capacity using a variable frequency drive (VFD) and changing the speed of the orbiting scroll. The speed modulates down to 35 hertz from either 60 Hz (208/230V) or 75Hz (460/575V with a few exceptions). That corresponds to a capacity turn down of about 58% (60Hz) and 47% (75Hz). Since the volume of refrigerant and speed decrease, the velocity in the refrigerant piping must be high enough to return oil back to the compressor. Each Bitzer compressor has an oil sight glass and an oil service connection.

Figure 21 - Bitzer Orbit Compressor for Variable Speed Drive

208 & 230V Applications can modulate between
35-60 Hz - 58-100% capacity control range

460 & 575V Applications can modulate between
35-75 Hz - 47-100% capacity control range

See Bitzer ESP-133-3 for more detailed information.

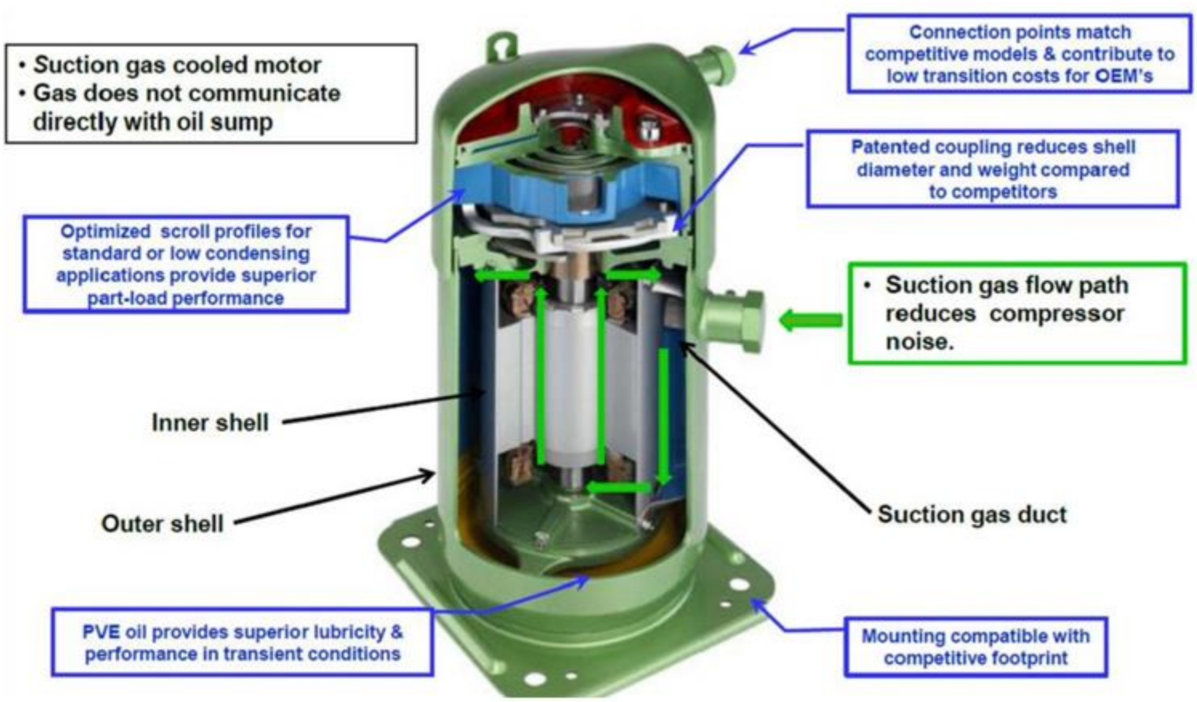


Figure 22 - Bitzer Orbit Compressor Schematic

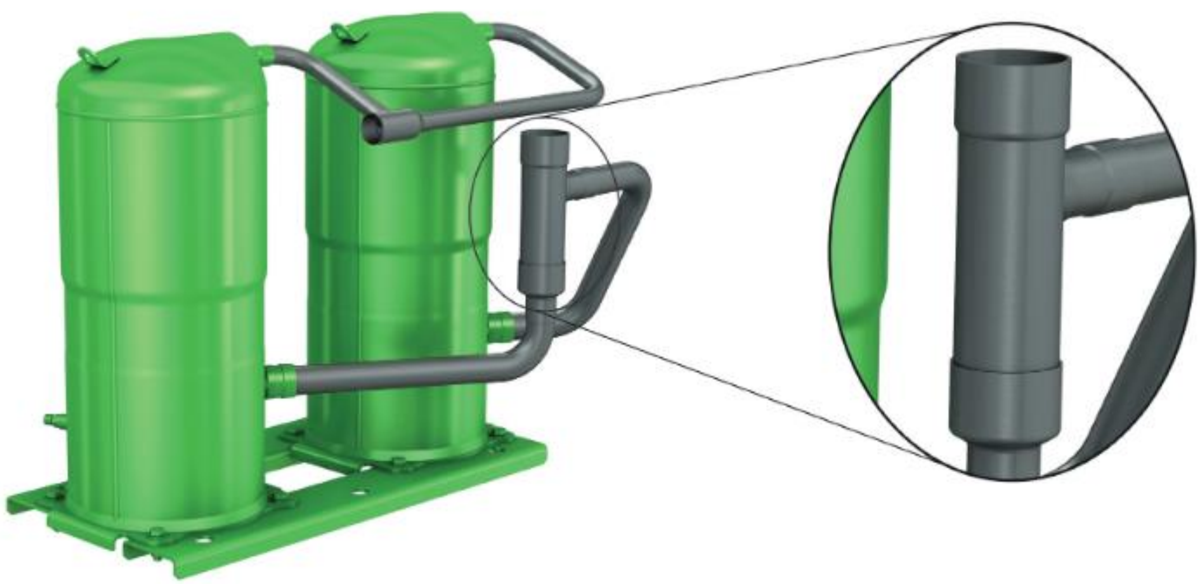
4.7 Bitzer Orbit Tandem VFD Controlled - R-410A

Tandem compressors are two compressors piped together with only one suction line and one discharge line. The Bitzer Tandem VFD Controlled Scroll compressors can be two different capacity compressors. These compressors use a VFD to control compressor capacity modulation. There are different variables possible in tandem compressors. In a tandem set these are the possibilities:

Figure 23 - Bitzer Orbit Tandem Compressor

1. Both standard on/off scroll compressors
2. Both VFD controlled scroll compressors
3. One standard on/off and one VFD controlled scroll compressor
4. Scroll compressors with two different capacities

See Bitzer ESP-131-6 for more detailed information.



4.8 Danfoss Inverter Scroll Compressor - R-410A

This is an inverter scroll compressor that modulates capacity using a permanent magnet motor control drive connected to the compressor. The speed of the compressor modulates from 900-6000 rpm. That corresponds to a capacity turn down of about 15%.

If the compressor runs below 2400 rpm for more than 120 minutes, the CDS will increase the compressor speed to 3600 rpm for 1 minute to facilitate oil return.

These are VZH compressors with CDS variable speed drives.

Danfoss Inverter Scroll Compressor Information

Figure 24 - Danfoss VZH Inverter Scroll Compressor

4.9 Danfoss TurboCor TT - R-134a

The TurboCor compressors are oil free, magnetic bearing, variable speed centrifugal compressors.

See Danfoss website for more detailed information.

Figure 25 - Danfoss TurboCor TT Compressor

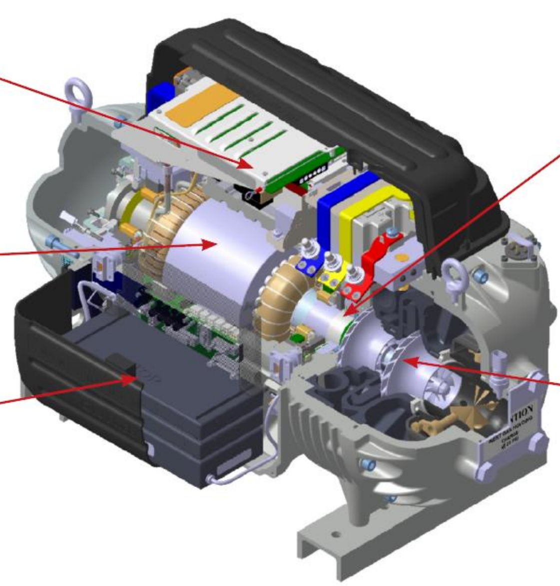


Figure 26 - Danfoss TurboCor TT400 Compressor Schematic

4.10 What can cause a Compressor Failure?

- Short cycling the compressor causing oil loss
- Excessive liquid refrigerant entering the compressor
- Insufficient oil returning to the compressor (not applicable in TurboCor compressors which are oil-free)
- Compressor operating outside of the operating envelope (temps, pressures, electrical limits). The more critical issue is temperatures above the limits rather than below. Excessive temperatures can cause the oil to break down.
- Eventual wear from too many on/off cycles

4.11 Variable Speed vs Digital Comparison

Variable speed drives and digital capacity modulation are somewhat different unloading methodologies in terms of how partial mass flow of refrigerant is achieved, but the outcome is actually fairly similar. The work done by the compressor to increase the pressure of a volume of suction refrigerant to the discharge pressure can be reduced when conditions are favorable to operate the refrigerant cycle at a reduced discharge (condensing) pressure. The closer together the suction and discharge pressure are, the less work must be done by the compressor, the strategy typically referred to as low lift.

The main difference between a variable speed compressor and a digital scroll compressor is the methodology whereby the low lift unloading is achieved. The variable speed reduces mass flow by decreasing the angular speed of the motor and thus the frequency of the scroll set (positive displacement of refrigerant) is reduced to correspond with the load. The digital scroll compressor loads and unloads in a cyclic fashion. The axial compliance mechanism for maintaining proper scroll tolerance during compression is pressure driven, with the top (fixed scroll) being held in place by back pressure from the discharge refrigerant. During unloading periods, a valve opens and the back pressure on the fixed scroll is relieved to the suction side of the compressor, which lifts the fixed scroll away from the orbiting scroll. This result in cessation of compression and the discharge valve closes due to insufficient pressure in the discharge chamber of the scroll set and positive displacement is suspended. This is akin to turning the compressor off in conventional cycling, but the compressor motor continues to spin freely without a load torque during this period so there is a nominal power draw (roughly 10% of full load power). The time period for the loaded and unloaded state is 15 seconds for both cycles. This means the compressor at 10% capacity would operate loaded for 1.5 seconds and unloaded for 13.5 seconds. The loading and unloading operation consists of four distinct phases: compression with full flow, transition from full flow to no flow, no flow, and transition to full flow.

The variable speed compressor will maintain constant, steady state conditions at the operating speed (a steady condensing temperature and a steady suction temperature). The digital scroll approximates those temperatures, but due to the pulsing nature of the flow, the process causes the condensing and evaporator temperatures to vary about the mean temperatures that the variable speed compressor achieves.

Below is an example of what can be seen in a short cycling condenser and evaporator, versus the mean temperature that the variable speed will hold. During the compression process, the digital scroll compressor maintains efficiency, as volumetric and isentropic efficiency of the compressor are observed to remain near that achieved during full flow. Additional heat transfer is also achieved during the off cycle due to the evaporator temperature remaining below ambient (an advantage that is a feature of short cycling due to the fast transition from on to off).

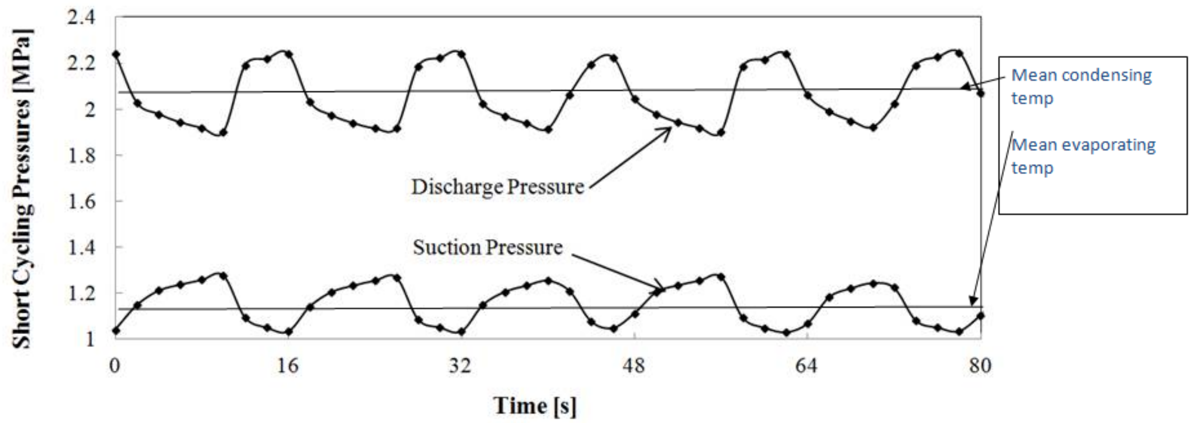


Figure 2. Measured suction and discharge pressures for R-410A during a short cycling operation with the a 50% on-cycle fraction

Generally speaking the digital scroll compressors are efficient at high to moderate loads. At low load fractions the compressor on time fraction is low and it requires more energy input to achieve the required cycle (not unlike startup for traditional cycling). At high to moderate loads however, the off time duration affords adequate power reduction. This is actually similar to the behavior of the variable speed scroll with hot gas bypass at lower loads to maintain flow and ensure continuous flow of oil. The variable speed compressors are typically efficient in a 50 to 100% regime, but will be less efficient below this point due to the bypass. However, the permanent magnet variable speed compressors are not limited to the 50% turndown, instead they can have as low as 15% turndown.

Oil return due to full flow of refrigerant during the on period and robustness due to minimal on/off transients in the digital scroll are advantages of the compressor.

5 Thermostatic Expansion Valve

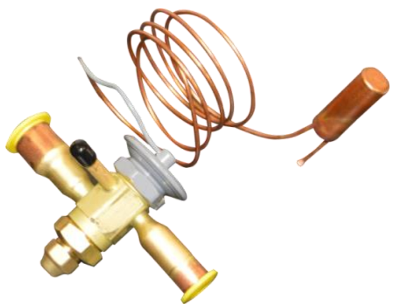


Figure 27 - TXV

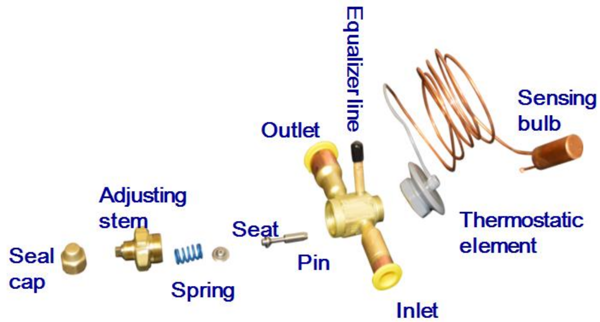


Figure 28 - TXV Internal Components

The thermostatic expansion valve (TXV) controls the amount of refrigerant entering the evaporator by maintaining a constant superheat value of the refrigerant vapor at the outlet of the evaporator.

$$Superheat = (Measured\ Temperature) - (SST)$$

Where:

Measured Temperature = refrigerant temperature at the evaporator outlet measured by the sensing bulb (see **Figure 31** - TXV Bulb Location for the best measured temperature results)

SST = *Saturation Suction Temperature of Refrigerant* is the temperature at which the refrigerant boils at a specific pressure (i.e. the saturation temperature of water is 212°F or 100°C at 1 atm pressure). See **Table 13** - R-410A Refrigerant Temperature-Pressure Chart for saturation temperatures at specific pressures. SST is typically between 40°F - 50°F.

Figure 30 - TXV in Air Handling Unit

It is critical to ensure superheat leaving the evaporator coil because the compressor is not designed to handle liquid. The refrigerant leaving the evaporator must be a superheated vapor in order to protect the compressor from damage. The amount of superheat is set using the adjusting stem. Turn the adjusting stem ½ to 1 full turn every 15 minutes until the correct superheat is reached. (Sporlan Form 10-143) Typical superheat values for HVAC are 8-15°F.

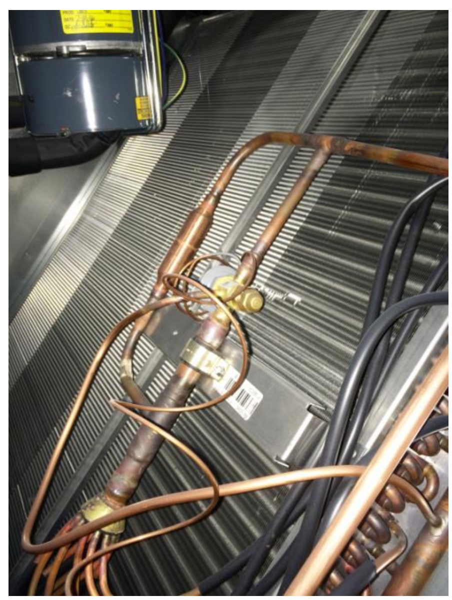


Figure 29 - TXV in Condensing Unit

Thermostatic expansion valve bulbs should be mounted with good thermal contact on a horizontal section of the suction line close to the evaporator, but outside the cabinet, and well insulated. On suction lines less than or equal to 7/8" OD, mount in the 12 o'clock position. On suction lines greater than 7/8" OD, mount in either the 4 o'clock or 8 o'clock position.

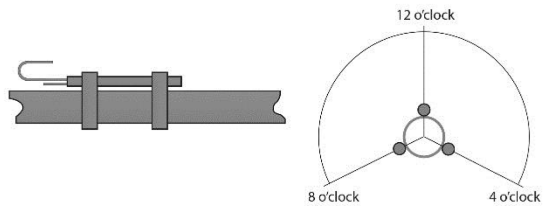


Figure 31 - TXV Bulb Location

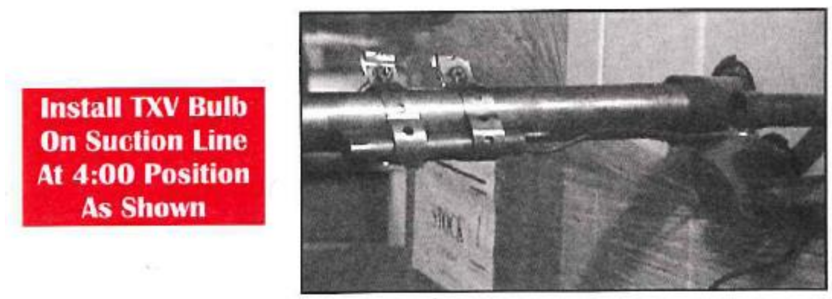


Figure 32 - TXV Bulb Installed on Suction Line

The 4:00 or 8:00 position is the best place to get a good representative temperature of the refrigerant inside. The bottom of the tube might have liquid oil which could cause erratic feeding. If the bulb cannot be located on a horizontal section of pipe, the next best option is to mount it on a descending vertical line. The bulb should never be located in a trap or downstream of a trap in the suction line.

Knowing the superheat in a refrigerant system is a big help in servicing the unit. A high superheat can indicate that the evaporator is not getting enough refrigerant, while a low superheat can indicate the evaporator is getting too much refrigerant. A rule of thumb for the proper amount of superheat is 8-15°F. See Sporlan Bulletin 10-11 for troubleshooting based on the superheat reading.

Sporlan TXV Nomenclature

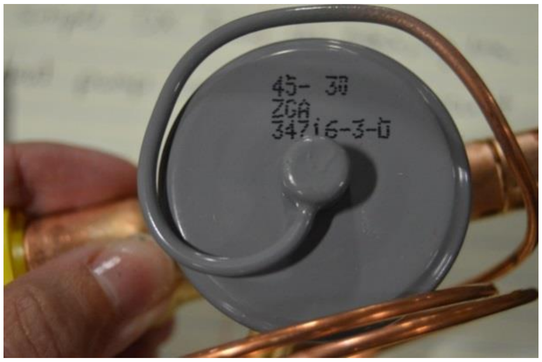


Figure 34 - TXV Element ZGA

Figure 33 - TXV Element ZCP160

- 45 = Element Size
- 30 = Length of capillary tube (30in)
- 34716-3-G = Date code
- Z = Refrigerant Code, R-410A

- GA = common AC charge inside the sensing bulb
- CP160 = Charge inside the sensing bulb
    - ZCP160 broken down further:
        - Z = R-410A application
        - C = By itself is medium temperature charge, but coupled with a P is typically AC charge
        - P160 = Pressure limiting charge with 160 psi nominal pressure. If the bulb senses pressures above 160 psi (57°F) it will not open any further. This is a safety feature to prevent floodback during long periods of shutdown.

BBI - Body type, no internal check valve
CBBI - Body type, with internal check valve
E = External Equalizer

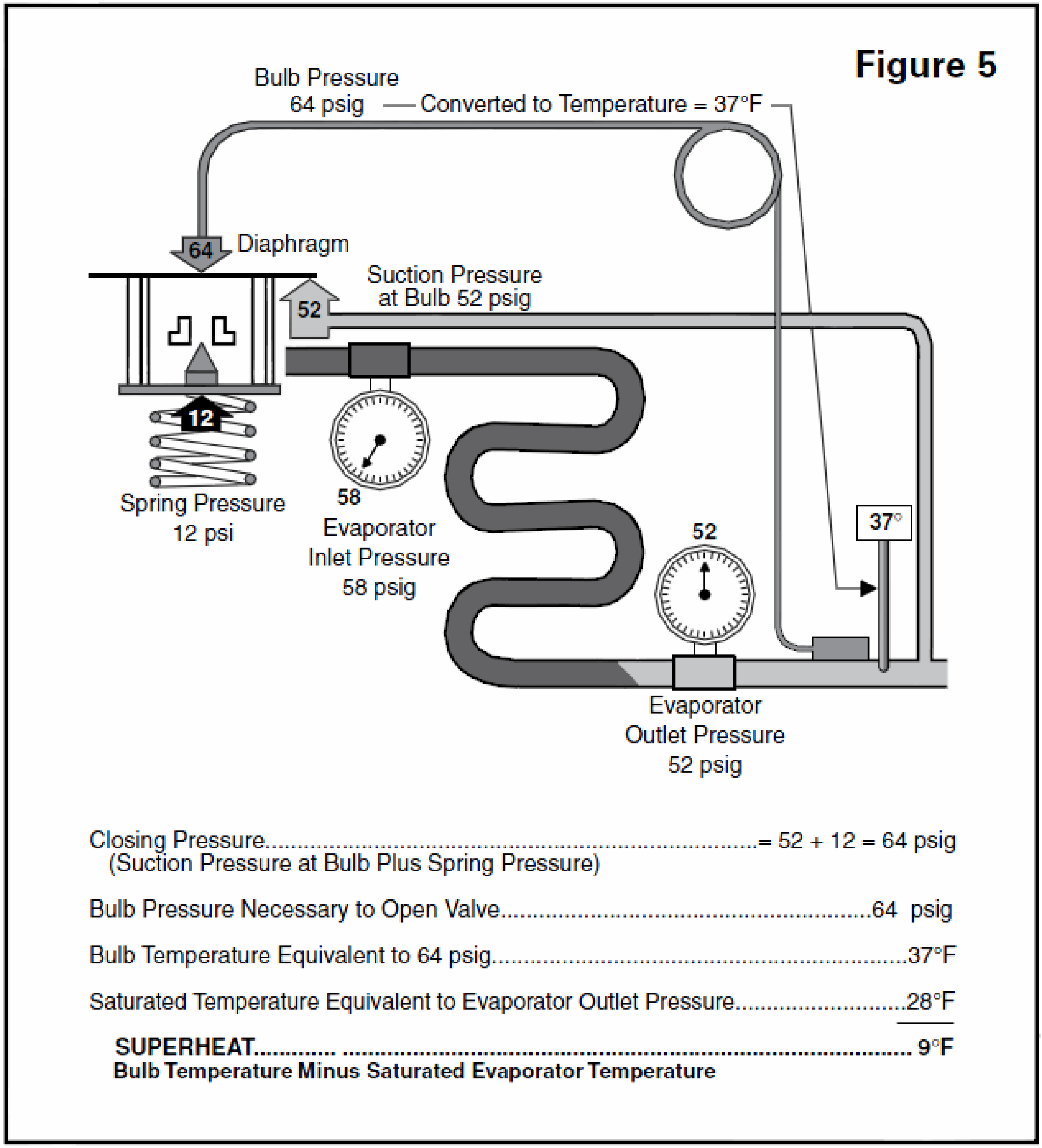


Figure 35 - Sporlan TXV Pressures Example
Courtesy of Sporlan Division – Parker Hannifin Corporation

The figure above shows how the TXV works and how the different pressures affect the operation of the valve. Notice the equalizing line, which gives the true suction pressure (52 psig), and the spring pressure (12 psi) work together to close the valve. If the bulb pressure is high enough to overcome the two closing pressures, then the valve will open allowing refrigerant to the evaporator coil. This example is based on R-22 refrigerant, if it were R-410A, the bulb pressure at 37°F would be 111 psig instead of 64 psig.

5.1 Electronic Expansion Valve

The electronic expansion valve serves the same purpose as the thermostatic expansion valve in that it controls the amount of refrigerant entering the evaporator by maintaining a constant superheat value of the refrigerant vapor at the outlet of the evaporator. It is a more sophisticated and precise operation using electronic signals that operate a 2500 step motor. These small steps allow the valve position to change in very small increments, changing the refrigerant flow in small amounts per step. The electronic signals sent to the EEV are a suction line temperature sensor and suction pressure transducer.

Figure 36 - Electronic Expansion Valve

See Sporlan Bulletin 100-20 for more detailed information on the valve and MCS Controls website on the controls.

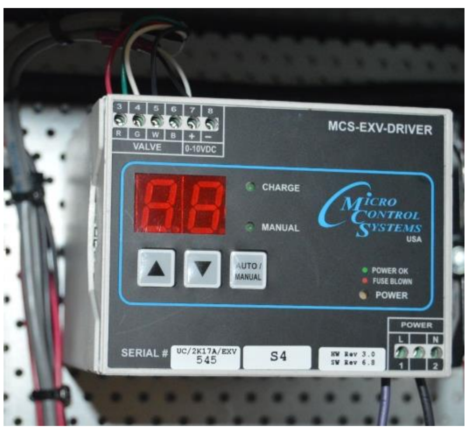


Figure 38 - EXV Driver

Figure 39 - Suction Line Sensors

Figure 37 - EXVs in AHU

6 Evaporator (DX) Coil

The role of the evaporator coil is to cool the air that blows across it for the comfort of the occupants of a space. That heat transfer from the air turns the liquid refrigerant into a superheated refrigerant vapor. This vapor leaves the DX coil and enters the compressor. The compressor must have a superheated refrigerant vapor in order to operate smoothly, so another important role of the evaporator coil is to produce a superheated refrigerant vapor.

Figure 42 - DX Coil

Figure 43 - Distributer Assembly

Figure 41 - Distributer Inlet

Figure 40 - Distributer Outlet

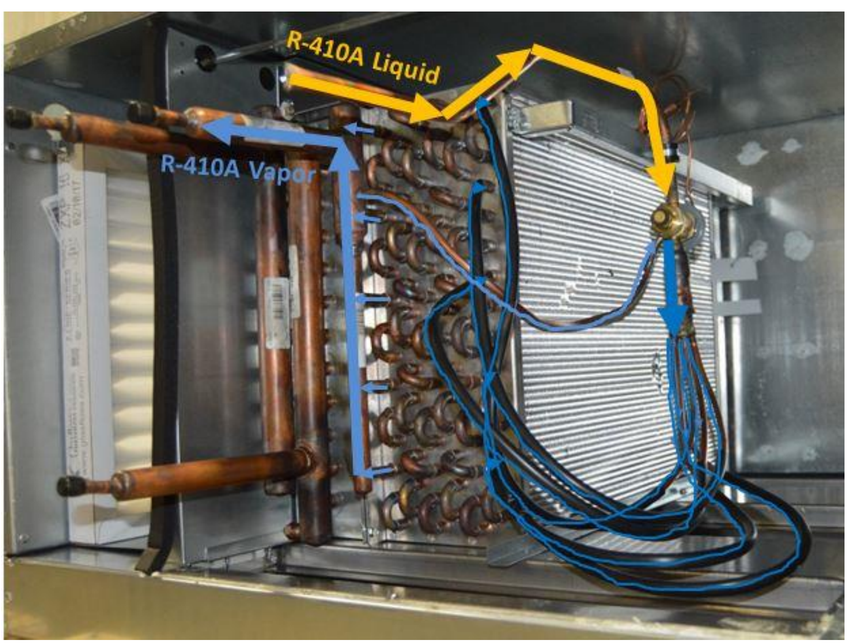


Figure 44 - Refrigerant Flow through a DX Coil

The picture shows a hot water coil upstream of an evaporator (DX) coil in an air handling unit. Ignoring the hot water coil, the letters below show the path of the refrigerant in the DX coil.

The liquid refrigerant travels into the entering liquid line (A) and then the solid column of liquid enters the TXV (B). The refrigerant then goes through the distributer (C) and into the five distributer tubes (D). It travels through the straight evaporator tubes, through return bends on the opposite side of the coil, back through straight tubes, through return bends (E) on this side of the coil and continues through the coil until it exits into the manifold header (F) and then into the suction line (G). Notice also the external equalizer line (H) and the TXV sensing bulb (I). The refrigerant vapor from the manifold header (F) travels through the external equalizer line (H) to an isolated passageway to the underside of the TXV diaphragm. The external equalizer allows the TXV to read the pressure at the evaporator outlet so that when the superheat is calculated, it is based on the actual pressure and temperature at the evaporator outlet. It removes the inaccuracy of the pressure drop through the coil. The sensing bulb (I) in this picture is not yet mounted and will be mounted in the field onto the suction line (G). It transmits bulb pressure to the top of the valve's diaphragm, based on the temperature it reads at the evaporator outlet.

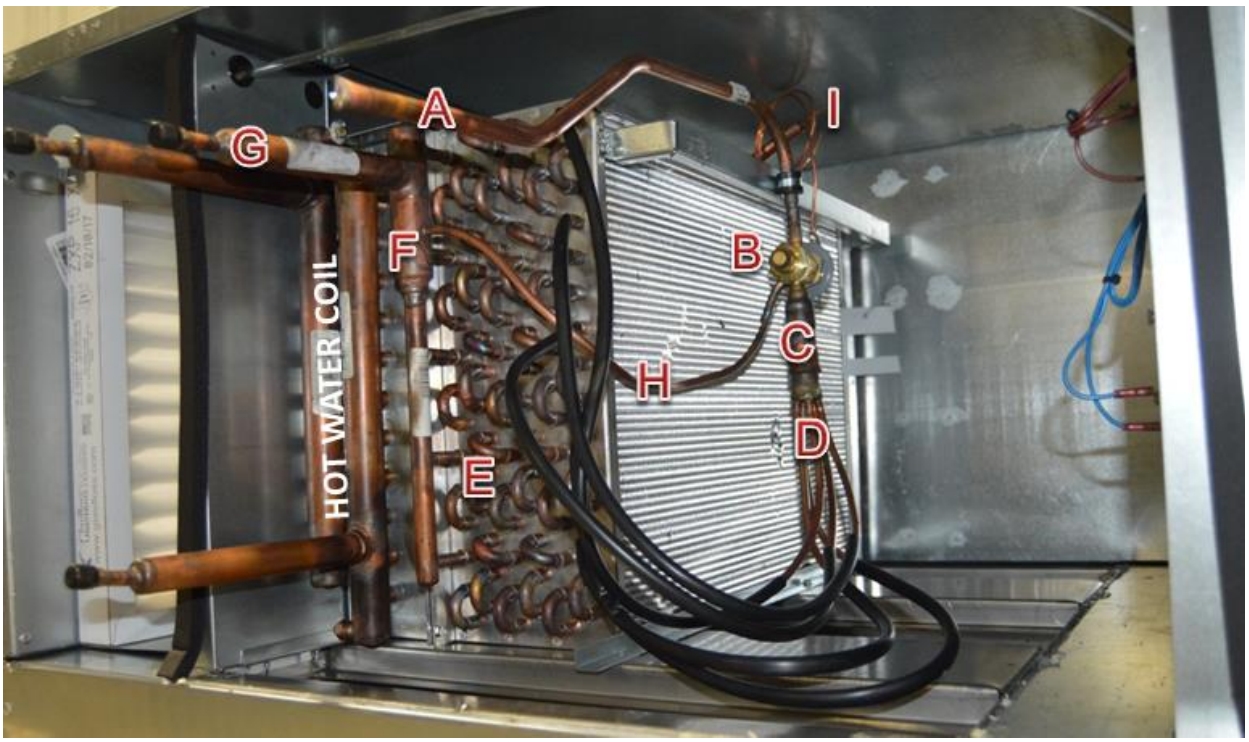


Figure 45 - Refrigerant Flow Steps through a DX Coil

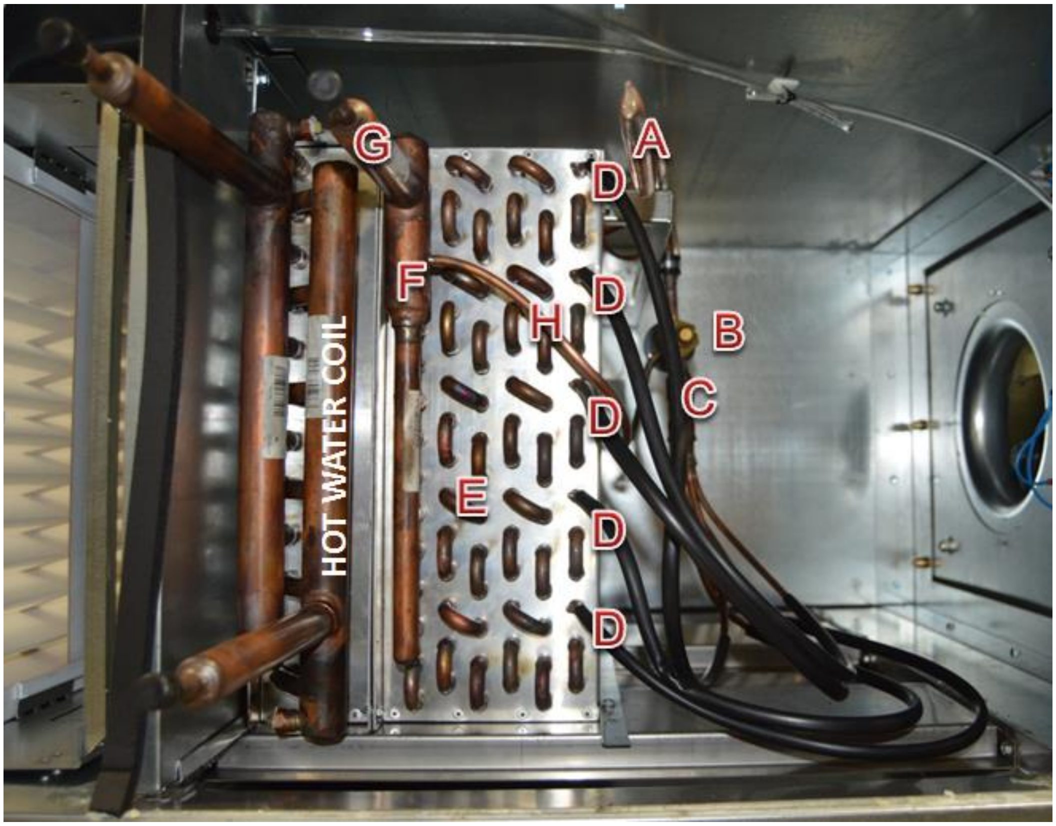


Figure 46 - Manifold View of DX Coil from Previous Figure

6.1 Polymer E-Coat

Polymer E-coating (aka ElectroFin E-Coat) can be applied to a coil intended for use in coastal saltwater conditions under the stress of heat, salt, sand and wind and can be used in other corrosive environments where a polymer e-coating is acceptable. The coating exceeds a 6,000 hour salt spray test per ASTM B117-97 requirements, yet is only 0.6-1.2 mils thick and has excellent flexibility.

See the Luvata website for more detailed information.

Figure 47 - E-coated DX Coil

7 Air-Cooled Condenser Coil

The role of the condenser coil is to reject heat from the superheated discharge refrigerant vapor so the refrigerant changes phases and becomes a sub-cooled refrigerant liquid. From the outlet of the condenser coil, the refrigerant travels through the liquid line to the inlet of the TXV.

For the TXV to operate correctly, the refrigerant entering the valve must be a sub-cooled liquid. Vapor in the liquid line, even in small quantities, will reduce valve capacity. The condenser coil is typically designed for a minimum of 10°F of refrigerant sub-cooling.

$$Subcooling = (SCT) - (Measured\ Temperature)$$

Where:
Measured Temperature = liquid refrigerant temperature at the condenser outlet

SCT = *Saturation Condensing Temperature* of refrigerant is the temperature at which the refrigerant condenses at a specific pressure (i.e. the saturation temperature of water is 212°F or 100°C at 1 atm pressure). See **Table 13 -** R-410A Refrigerant Temperature-Pressure Chart for saturation temperatures at specific pressures.

These are the types of cooling that occur in the condenser coil:

- Desuperheating - sensible cooling of discharge vapor
- Condensing - latent cooling of condensing vapor to liquid
- Sub-cooling - sensible cooling of sub-cooling the liquid

Temperature difference between SCT & ambient air entering condenser can range between 10-30°F. So if ambient is 90°F, then SCT will be around 110°F and if ambient is 80°F, then SCT will be around 100°F.

Figure 48 - Condensing Unit

7.1 Fin and Tube Coil

Figure 49 - Fin and Tube Condenser Coil

Fin and tube condenser coils are constructed of copper tubes with aluminum fins mechanically bonded to the tubes and aluminum end casings with a sine wave rippled fin design.

Coils are designed for a minimum of 10°F of refrigerant sub-cooling.

7.2 Microchannel Coil

Tubes and fins are both aluminum with a copper stub out.

In general, microchannel coil use less refrigerant charge, have increased efficiency, high corrosion resistance, compact design, and less weight compared to fin and tube coils.

Coils are designed for a minimum of 10°F of refrigerant sub-cooling.

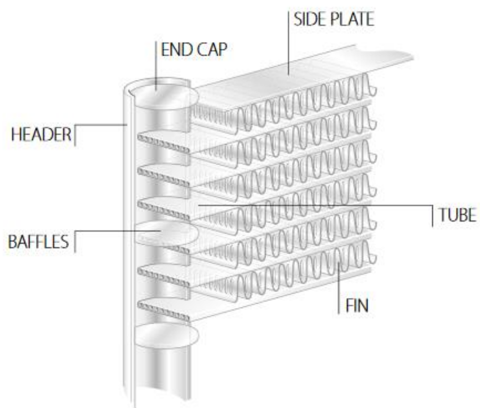


Figure 50 - Danfoss Microchannel Schematic

Source - Danfoss Microchannel Brochure

Figure 51 - Microchannel Coil

Microchannel coils include a heat shrink around the copper-aluminum connection. This protects the joint from ambient conditions and moisture and thus prevents corrosion arising due to dissimilar metals. The copper and aluminum is hand brazed after the coil comes out of the oven. After brazing the heat shrink is applied.

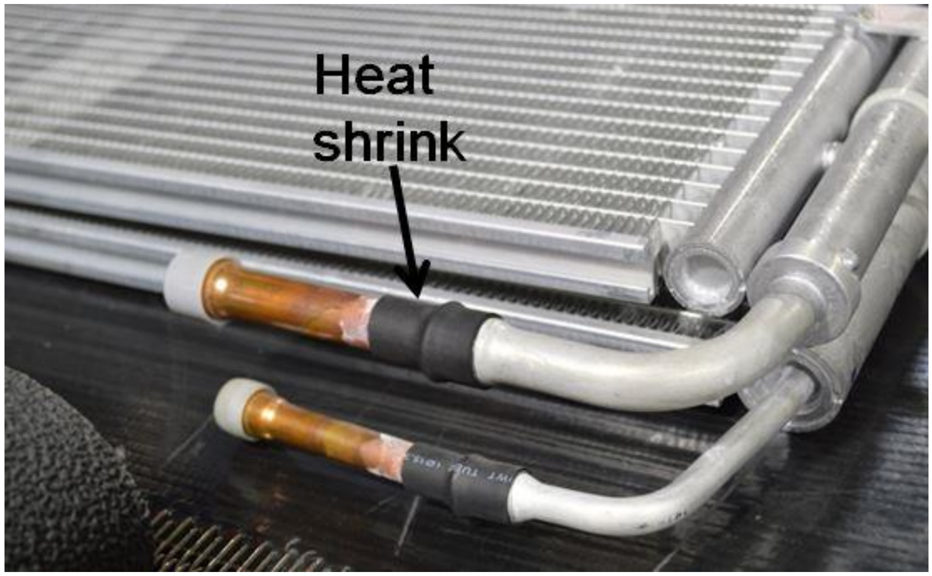


Figure 52 - Microchannel Copper-Aluminum Connection

7.3 Wrap Around Coil

Fin and tube wrap-around, single row, high efficiency condenser coils are constructed of copper tubes with aluminum fins mechanically bonded to the tubes and aluminum end casings with a sine wave rippled fin design.

Used only on the CB Series condensing units.

A single row wrap-around coil is easier to clean than a multi-row wrap-around coil.

This particular coil is a heat pump coil evidenced by the TXV valve installed.


Figure 53 - Wrap Around Condenser Coil

8 Evaporative-Cooled Condenser

In the evaporative-cooled condensing process, water is sprayed over the condenser coil as the condenser fans draw air across the coil to evaporate the spray and cool the refrigerant tubes toward the ambient wet bulb temperature. Unlike an air-cooled condenser which rejects heat from the refrigerant to the air at the ambient **dry bulb** temperature, an evaporative-cooled condenser rejects heat from the refrigerant to the water at the **wet bulb** temperature which can be 15° to 25°F lower than dry bulb. The lower condensing temperature means that the evaporative-cooled condenser can reject more heat than an air-cooled condenser, while requiring less compressor work and consuming less energy. As a result an evaporative-cooled condenser can be 20% to 40% more efficient than a comparable air-cooled condenser.

Figure 54 - Evaporative-Cooled Condenser

The patented AAON evaporative-cooled condenser design employs a de-superheater coil above the wetted section to reduce the potential for scale formation (Patent #6,715,312). The de-superheater rejects heat through forced air convection (via the condenser fans), reducing the temperature of the refrigerant to saturation before it enters the condenser coil, and reducing the probability that the water will rapidly evaporate and leave mineral deposits on the coil. This increases the life and efficiency of the condenser coil, and also requires significantly less water for cooling. At ambient temperatures below 33°F DB all of the heat of rejection can be transferred through the de-superheater, offering 100% water savings. By requiring less cooling water than conventional evaporative-cooled condensers at all ambient conditions, less make-up water is needed, reducing both water and treatment costs.

Another benefit of the de-superheater coil is it removes some humidity out of the air before it goes through the condenser fan. With the fan motor in a non-condensing environment there are fewer tendencies for any water to infiltrate the motor.

The air-cooled de-superheater coil is a fin and tube coil with polymer e-coat, but the evaporative-cooled condensing coil uses copper tubes without fins.

Figure 55 - Evaporative-Cooled Condenser Coil

While the AAON design minimizes the scale deposits on the coil, there is a possibility of scale if the water is not treated properly based on the local water conditions. The evaporative-cooled condensing unit includes a factory installed water treatment system. The following components are installed:

- Recirculating Water Pump
- Three Chemical Pumps & Tanks
- Conductivity Sensor
- Water Bleed Valve
- Water Meter
- DDC Controller

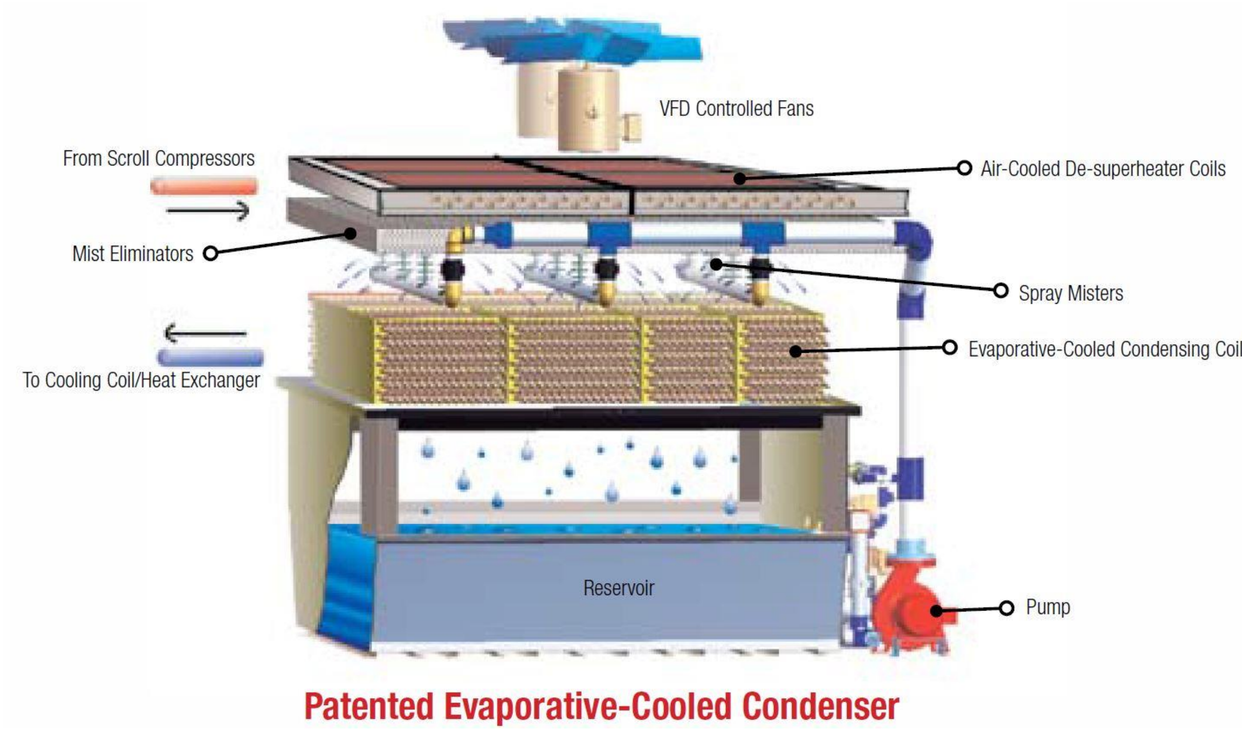


Figure 56 - Evaporative-Cooled Condenser Schematic

See The Patented AAON Evaporative-Cooled Condenser Brochure for more details.

9 Split System - Standard

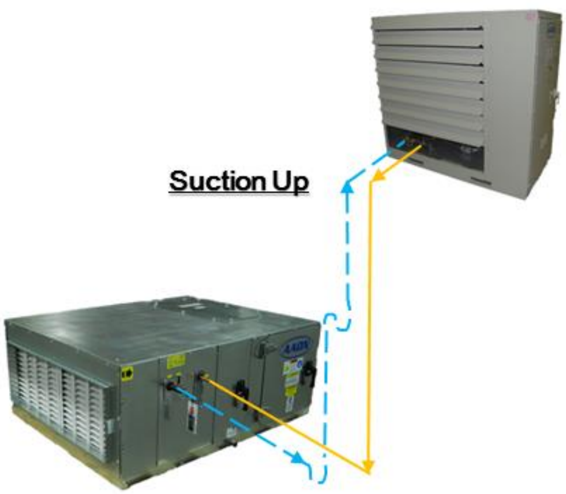


Figure 57 - Split System AHU & CU

A split system is two different pieces of equipment, each containing only part of the four basic components needed in refrigeration. Typically one unit will be outside of the building (condensing unit) and one unit will be inside of the building (air handling unit). A typical condensing unit will contain the condensing coil, condenser fan, liquid line filter-drier, and the compressor. A typical air handling unit will contain the evaporator coil, a supply fan, and TXV.

Copper piping must be run between the air handling unit copper stub outs and the condensing unit copper stub outs. The suction line and liquid line must be field piped. The discharge line is factory piped inside the condensing unit.

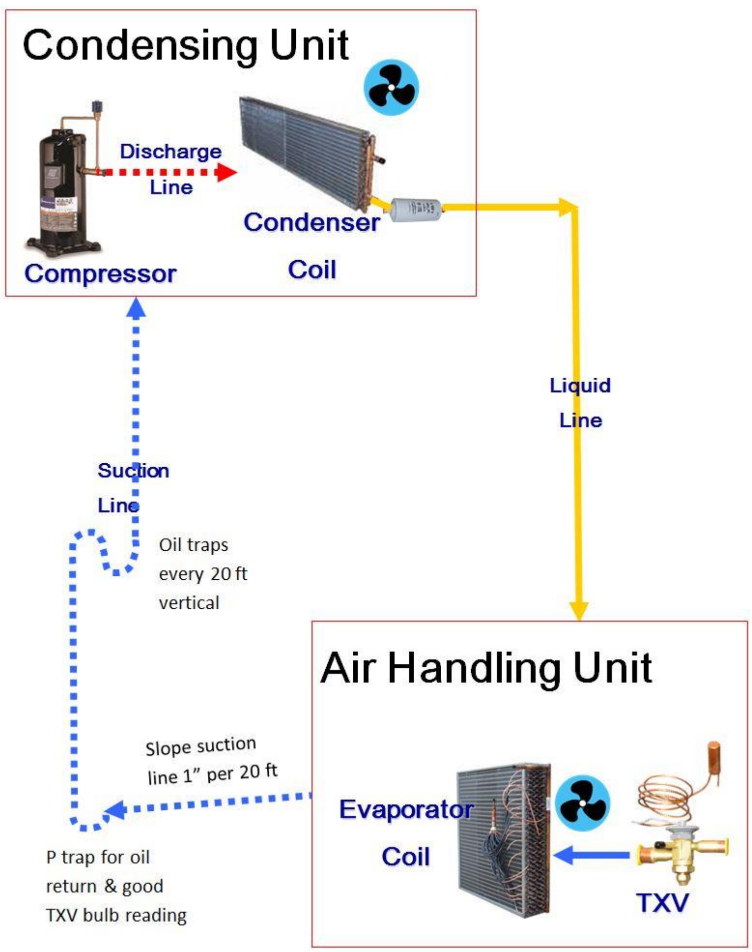


Figure 58 - Split System Standard Schematic Condensing Unit Above AHU

This is the piping diagram for a standard split system with the condensing unit above the air handling unit. (Suction Up, Liquid Down).

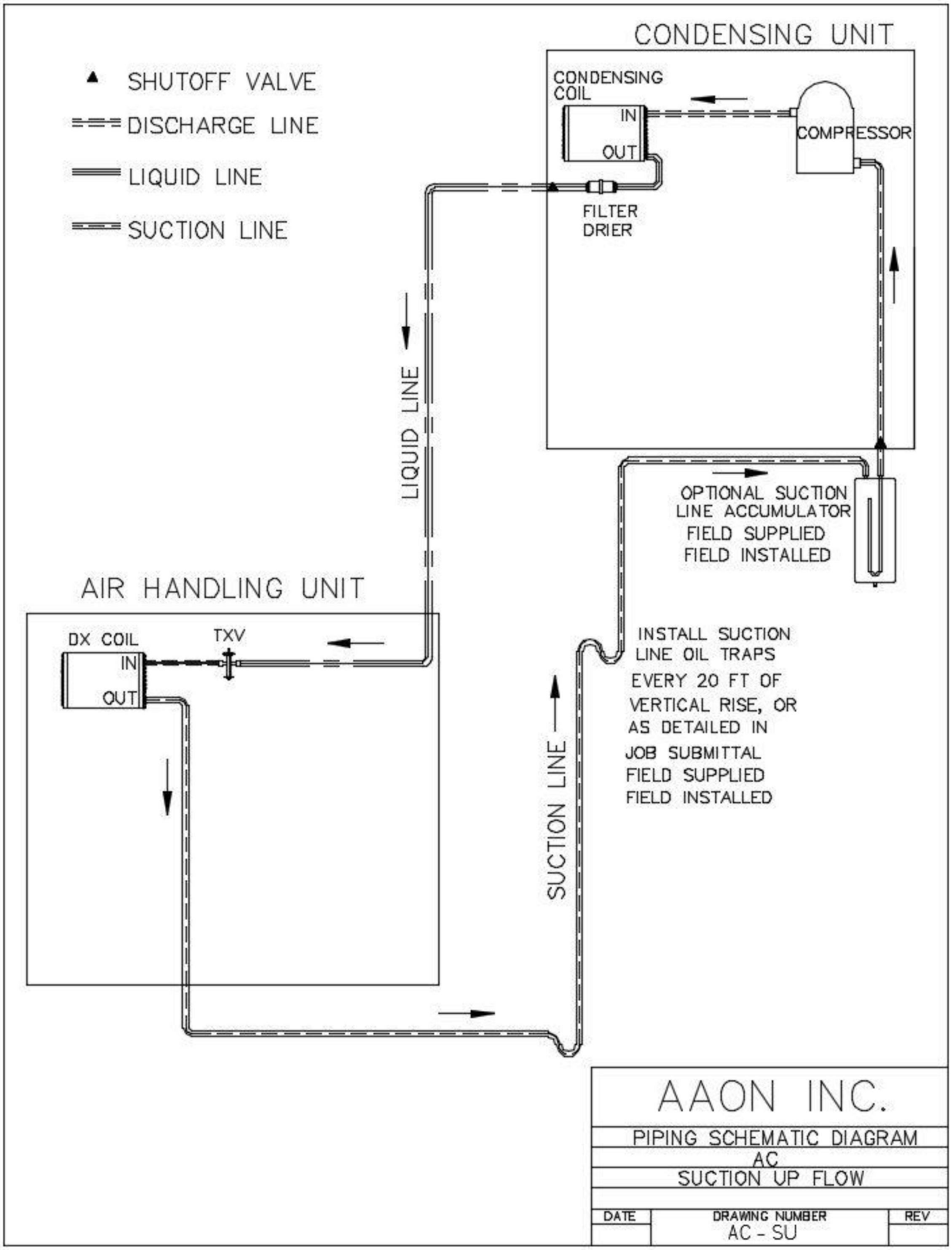


Figure 59 - Split System Suction Up Piping Diagram

10 Split System - Remote Condenser

A remote condenser will contain the condensing coil, condenser fan, and liquid line filter-drier. The matching air handling unit will contain the evaporator coil, a supply fan, TXV, and compressor.

Copper piping must be run between the air handling unit copper stub outs and the condensing unit copper stub outs. The discharge line and liquid line must be field piped. The suction line is factory piped inside the air handling unit.

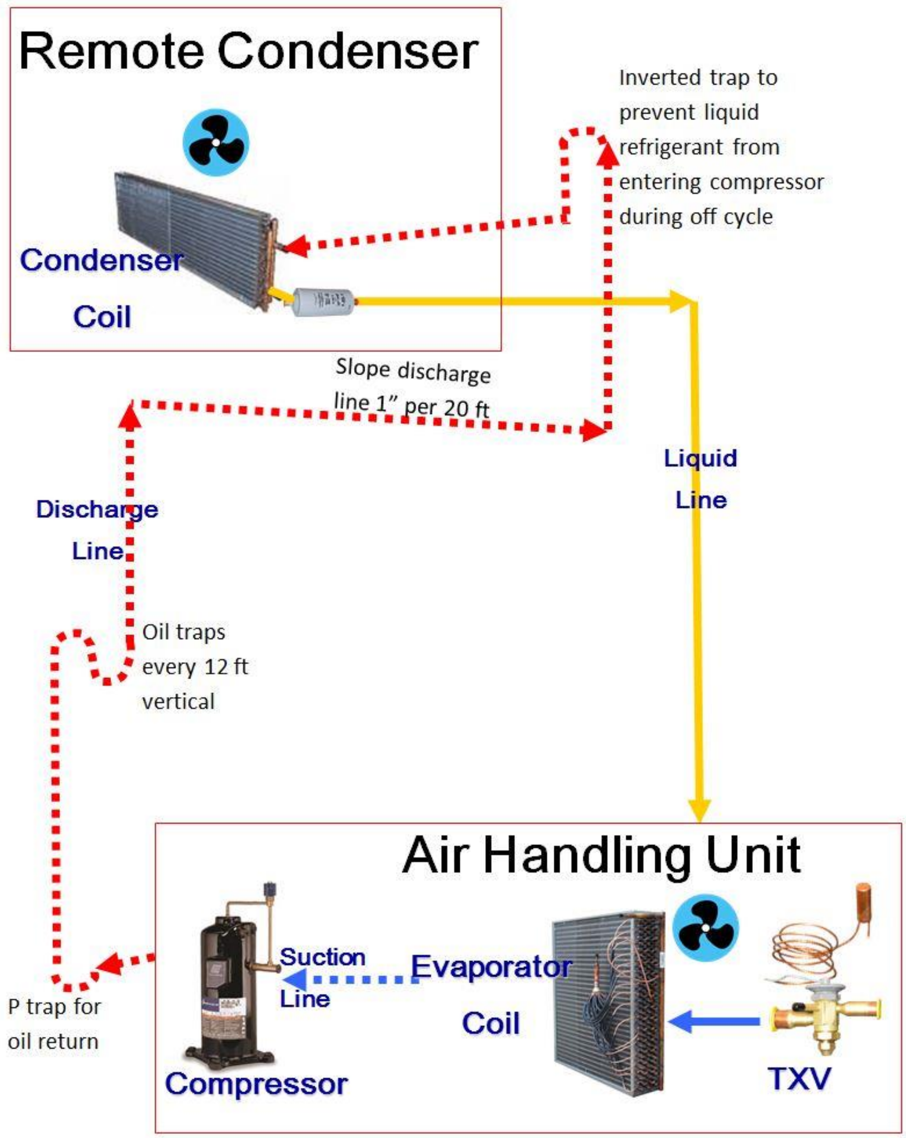


Figure 60 - Split System Remote Condenser Schematic

11 Split System - Modulating Hot Gas Reheat

Some split systems include modulating hot gas reheat as an option. The condensing unit will contain the condenser coil, condenser fan, the compressor, a liquid line receiver, liquid line filter-drier, and a modulating hot gas reheat valve. The air handling unit will contain the evaporator coil, a supply fan, the TXV, the hot gas reheat coil and two check valves.

Copper piping must be run between the air handling unit copper stub outs and the condensing unit copper stub outs. The suction line, liquid line, and hot gas reheat line must be field piped. The hot gas reheat line must include a purge circuit (see Purge Circuit). The discharge line is factory piped inside the condensing unit.

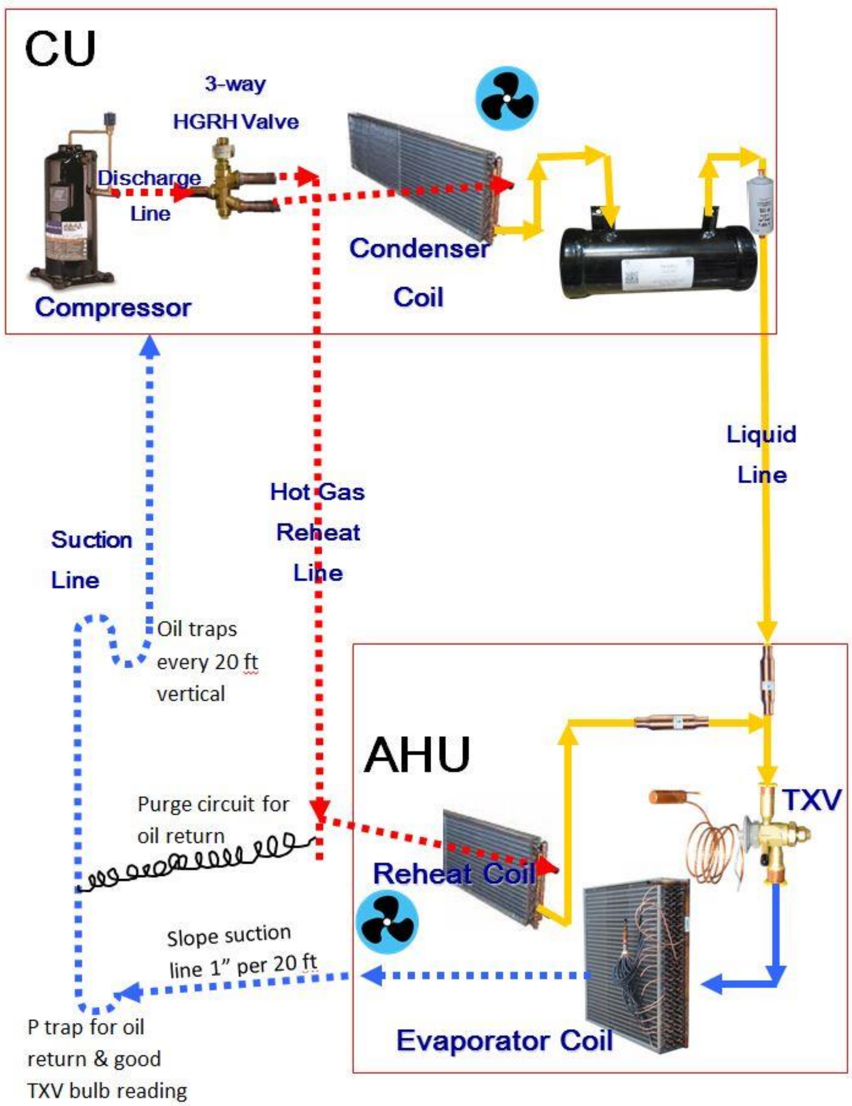


Figure 61 - Split System 3-Pipe (Parallel) Modulating Hot Gas Reheat Schematic

How Does Modulating Hot Gas Reheat Work?

The role of the hot gas reheat coil is to bring supply air temperature back up to a comfortable temperature after supply air leaves the DX coil. In order to remove moisture from the supply air for humidity control, the air temperature leaving the DX coil needs to fall below the dew point of the air so that when the air blows across the DX coil, the water droplets in the air will condense onto the DX coil and result in a less humid, cold air leaving the DX coil. The hot gas reheat coil warms the less humid, cold air back up to a comfortable temperature so that cold air is not blowing on the occupants of the conditioned space.

Typically a high capacity 6-row DX coil would be used to maximize the ability to remove moisture from the air. A hot gas reheat coil is basically a second condenser coil in the supply air stream. The piping schematic shows a 3-way modulating hot gas reheat valve controlling the flow to the condenser coil and the hot gas reheat coil. If 70% of refrigerant flow goes through the condenser coil, then only 30% will go through the reheat coil. The valve modulates the flow to the reheat coil in order to maintain the supply air temperature. The space humidity sensor reads the humidity in the air and sends a signal to enable dehumidification if the value is above the setpoint.

Heating and cooling calls usually override a dehumidification call (unless the controls are set up with dehumidification as the priority); however, when the dry bulb temperature is satisfied but the humidistat is not satisfied, the compressors will continue to operate.

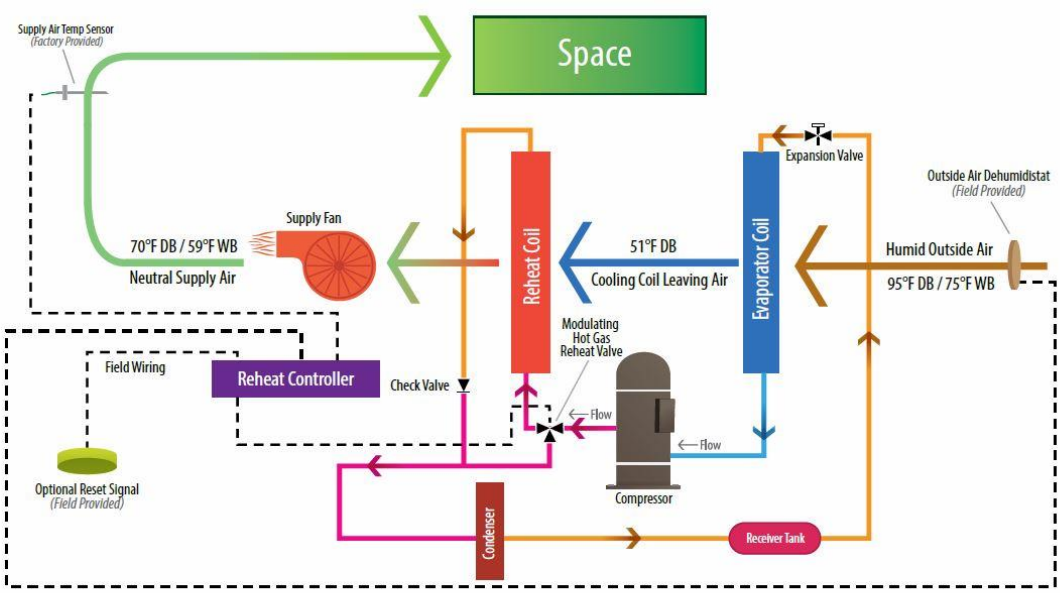


Figure 62 - 4-Pipe (Series) Modulating Hot Gas Reheat (Unitary Systems) Schematic

The only difference between the split system (3-pipe) and unitary system (4-pipe) schematics is the reheat coil outlet piping. The split system reheat coil outlet feeds into the inlet of the TXV so that only one hot gas reheat line must be field piped. The unitary piping does not have to be field piped so the reheat coil outlet feeds into the inlet of the condenser coil.

See the psychrometric chart below for the air as it goes through the evaporator coil and then through the reheat coil.

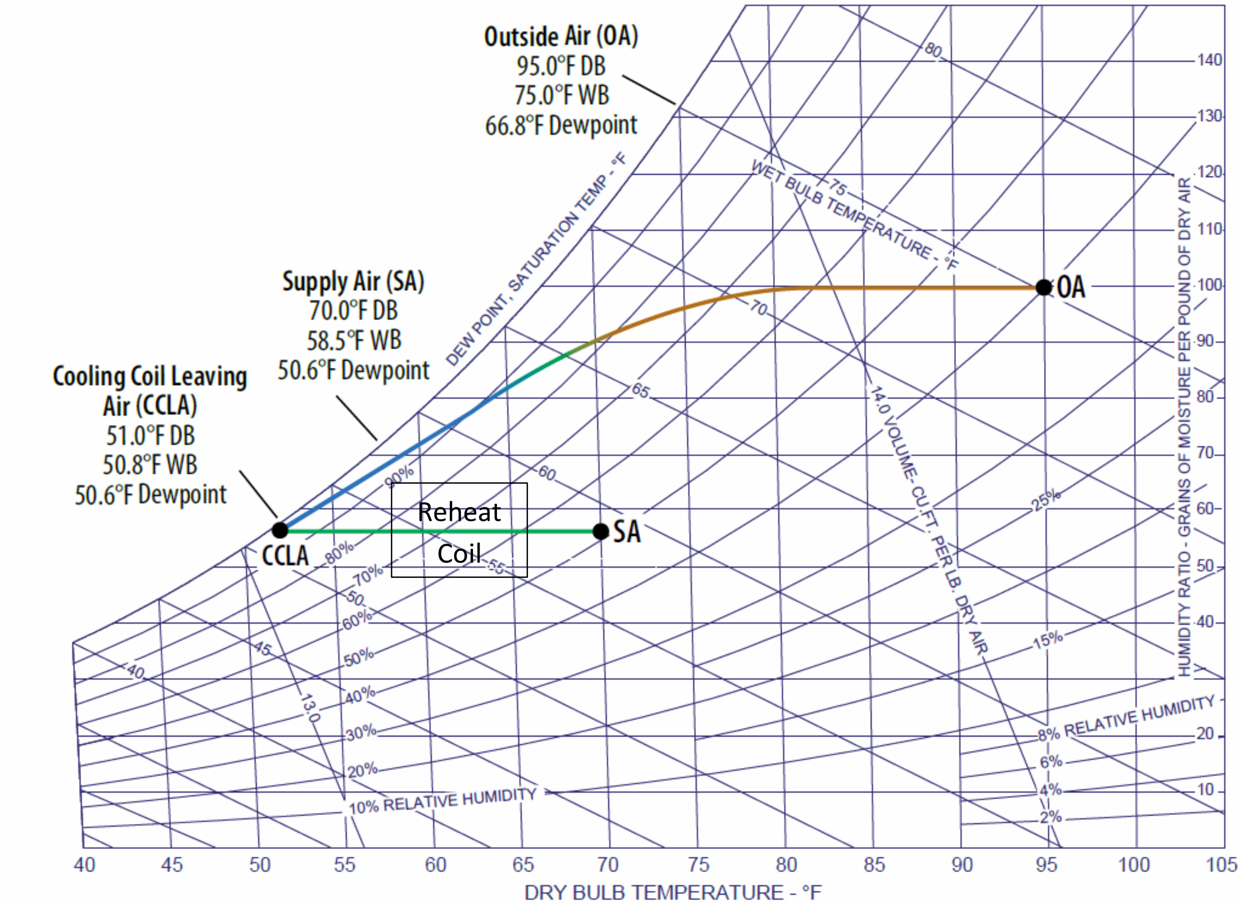


Figure 63 - Outside Air Modulating Hot Gas Reheat System Psychrometric Chart

11.1 Modulating Hot Gas Reheat Valve

The modulating hot gas reheat operation has been accomplished in two different ways. Either a 3-way modulating valve or two 2-way modulating valves can be used to accomplish the same goal. When two 2-way modulating valves are used, one of the valves modulates the refrigerant discharge gas entering the condenser coil, and the other valve modulates the refrigerant discharge gas entering the reheat coil. The valves modulate opposite of each other so that as one allows more flow through, the other allows less flow through. Basically, the condenser coil and reheat coil are sharing the same refrigerant discharge gas stream. Both coils are condensing the refrigerant discharge gas into a liquid, but the condenser coil transfers the heat to the ambient air, and the reheat coil transfers heat into the supply air stream. In split systems, the two streams of refrigerant eventually end up in the same place, as a liquid entering the TXV. In unitary systems, after the refrigerant passes through the reheat coil, it then passes through the condenser coil. The 3-way valve works exactly the same way; it is simply in one valve instead of two.

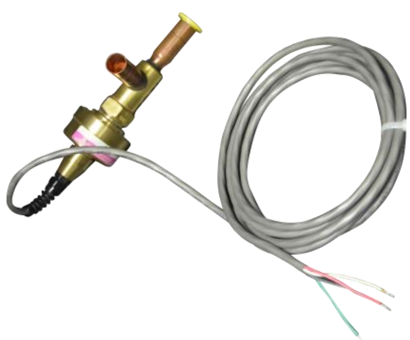


Figure 64 - Modulating Hot Gas Reheat 2-Way Valve

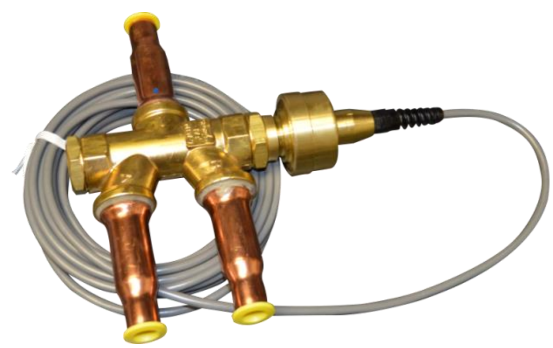


Figure 65 Modulating Hot Gas Reheat 3-Way Valve

Why add heat to a room I am trying to cool? If the humidity in a room is high, the room can become very uncomfortable. So even if the supply air temperature setpoint has been met, the room still feels uncomfortable. In order to get the humidity out of the air, the air across the evaporator coil must drop below the dew point temperature of the air so the moisture can come out of the air and condense onto the evaporator coil. The supply air temperature that is needed to remove humidity is too cold to blow directly into a room, so the reheat coil warms the supply air temperature back up to a comfortable supply air temperature. See Sporlan Bulletin 100-40-3 for more detailed information on the 3-way valve and Sporlan Bulletin 100-40 for more detailed information on the 2-way valves.

Why Select Modulating Hot Gas Reheat?

Indoor air quality has many benefits both to the occupants of the space and to the well-being of the space. The study of indoor air quality has pointed to moisture as one of the main causes of poor indoor air quality. High humidity can cause many problems including mold growth, condensation, wood rot, paper deterioration, increase in sickness and allergic reactions, and a variety of other indoor air quality and physically damaging issues.

In many geographic locations and in some building applications, the standard moisture removal ability of the cooling system cannot match the humidity control demand since it has been designed to satisfy a dry bulb temperature setpoint. When the setpoint has been satisfied, the supply fan continues to provide fresh air while the cooling system is shutoff until the temperature rises to some level above the setpoint. During that off time all moisture removal capability is lost. Moisture on the cooling coil evaporates back into the supply air stream, increasing the relative humidity.

In some applications, situations can arise where the space cooling load is low, and the space temperature setpoint is satisfied without the supply air being cooled to the dew point to remove moisture. This can result in a space humidity that is greater than desired. For such situations, a reheat dehumidification strategy is implemented.

On/off reheat solves most moisture related problems in a return air system, but in systems conditioning ventilation air, the following problems continue:

- Poor control of the amount of reheating, which wastes energy
- Uncomfortable supply air temperature swings during operation
- Unacceptable supply air temperatures and poor temperature control when used in make-up air applications, especially 100% outside air

Figure 66 - Modulating Hot Gas Reheat 3-way Valve Installed in Condensing Unit

Figure 67 - Reheat Coil in AHU

Modulating hot gas reheat provides consistent supply air temperature during dehumidification.

- Since the amount of hot refrigerant gas passing through the reheat coil is modulated, the system delivers only the amount of reheating that is required for space comfort.
- Occupant comfort is uniform and consistent because there will be no drastic swings in the supply air temperature that are inherent with on/ off solenoid valve control systems.

See The Humidity Control Solution Brochure for more details.

This is the piping diagram for a typical split system with modulating hot gas reheat when the air handling unit is above the condensing unit (Suction Down, Hot Gas Reheat Up, Liquid Up).

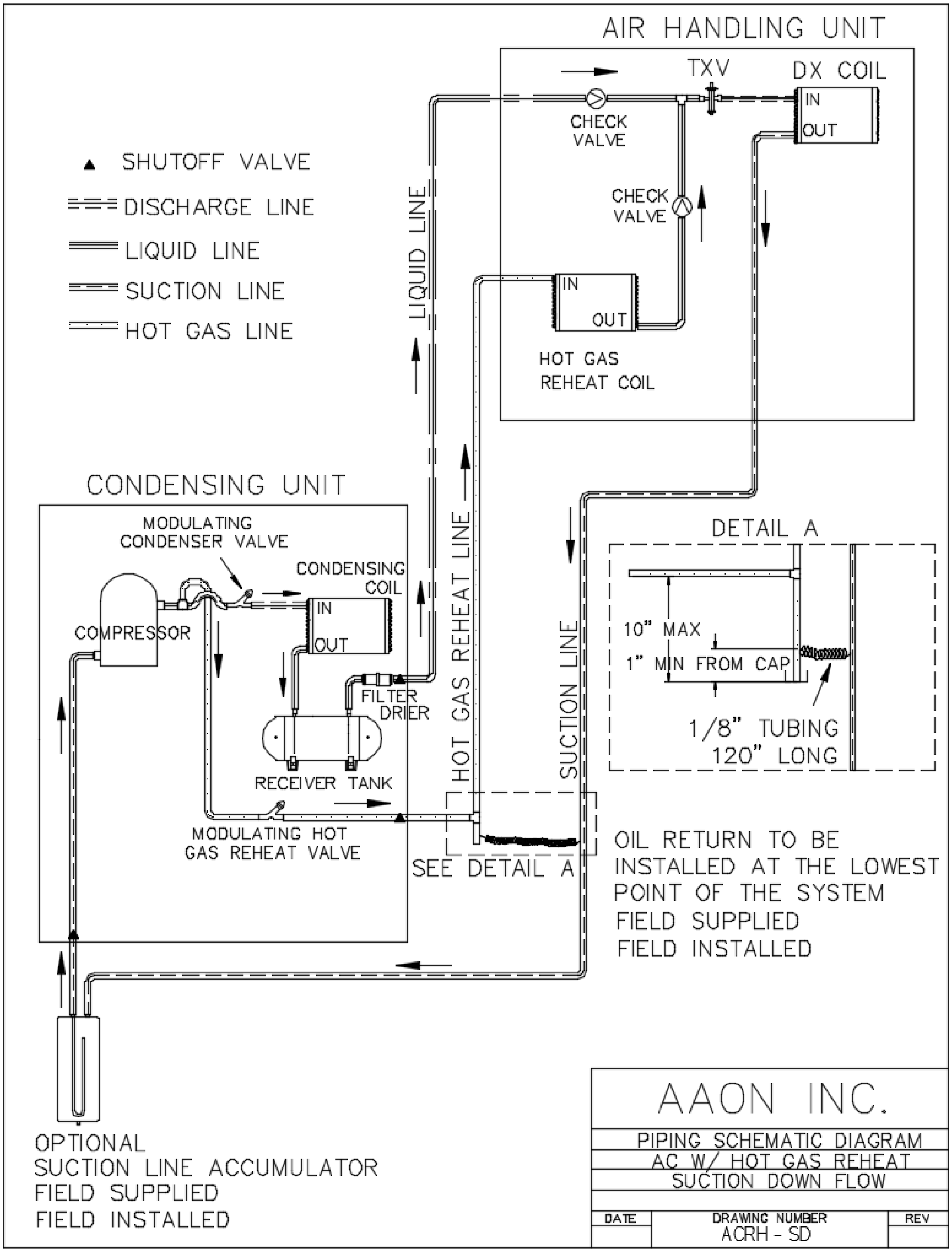


Figure 68 - Split System Modulating Hot Gas Reheat Piping Diagram

12 Split System - Hot Gas Bypass

Some split systems include hot gas bypass as an option. The condensing unit will contain the condensing coil, condenser fan, the compressor, liquid line filter-drier, and a hot gas bypass valve. The air handling unit will contain the evaporator coil, a supply fan, the TXV and a check valve.

Copper piping must be run between the air handling unit copper stub outs and the condensing unit copper stub outs. The suction line, liquid line, and hot gas bypass line must be field piped. The hot gas bypass line must include a purge circuit (see Purge Circuit). The discharge line is factory piped inside the condensing unit.

How Does Hot Gas Bypass Work?

A field adjustable, pressure activated hot gas bypass valve diverts hot compressor discharge gas to the evaporator coil if pressure on the evaporator side of the valve drops below 105 psi for R-410A (34°F at sea level). The bypass valve is at full capacity after six degrees of differential (28°F at sea level). This option helps prevent coil freezing during periods of low airflow or cold entering coil conditions. This option is used for refrigerant system protection only and cannot be used for cooling capacity modulation.

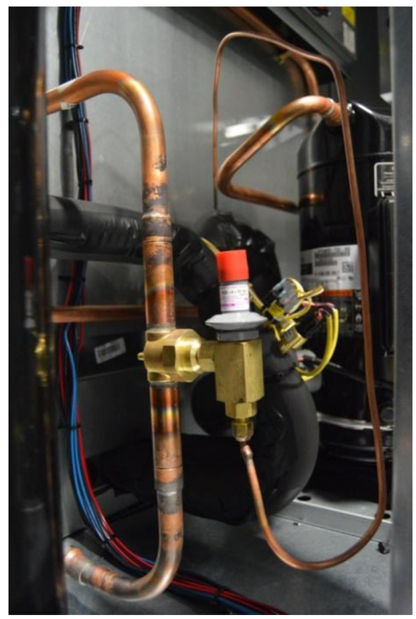


Figure 69 - Hot Gas Bypass Valve Installed in Condensing Unit

Why Select Hot Gas Bypass?

Hot gas bypass is required on all Variable Air Volume (VAV) and Makeup Air (MUA) units without digital scroll compressors. Hot gas bypass on the lag circuits is recommended on all VAV and MUA units with digital scroll compressors on only the lead circuits. Do not use hot gas bypass on circuits with digital scroll compressors.

12.1 Hot Gas Bypass Valve

Hot gas bypass is becoming more and more obsolete, and that is because digital scroll compressors are being used more and more. Hot gas bypass basically allows the system to continue to run at lower loads by bypassing discharge gas from the compressor outlet to the inlet of the evaporator coil. It false loads the evaporator so the system can keep running at lower loads. A digital scroll compressor is a much more efficient solution to get a wider range of operation.

Figure 70 - Hot Gas Bypass Valve Style 1

Figure 71 - Hot Gas Bypass Valve Style 2

This is the piping diagram for a typical split system with hot gas bypass when the air handling unit is above the condensing unit. (Suction Down, Hot Gas Bypass Up, Liquid Up).

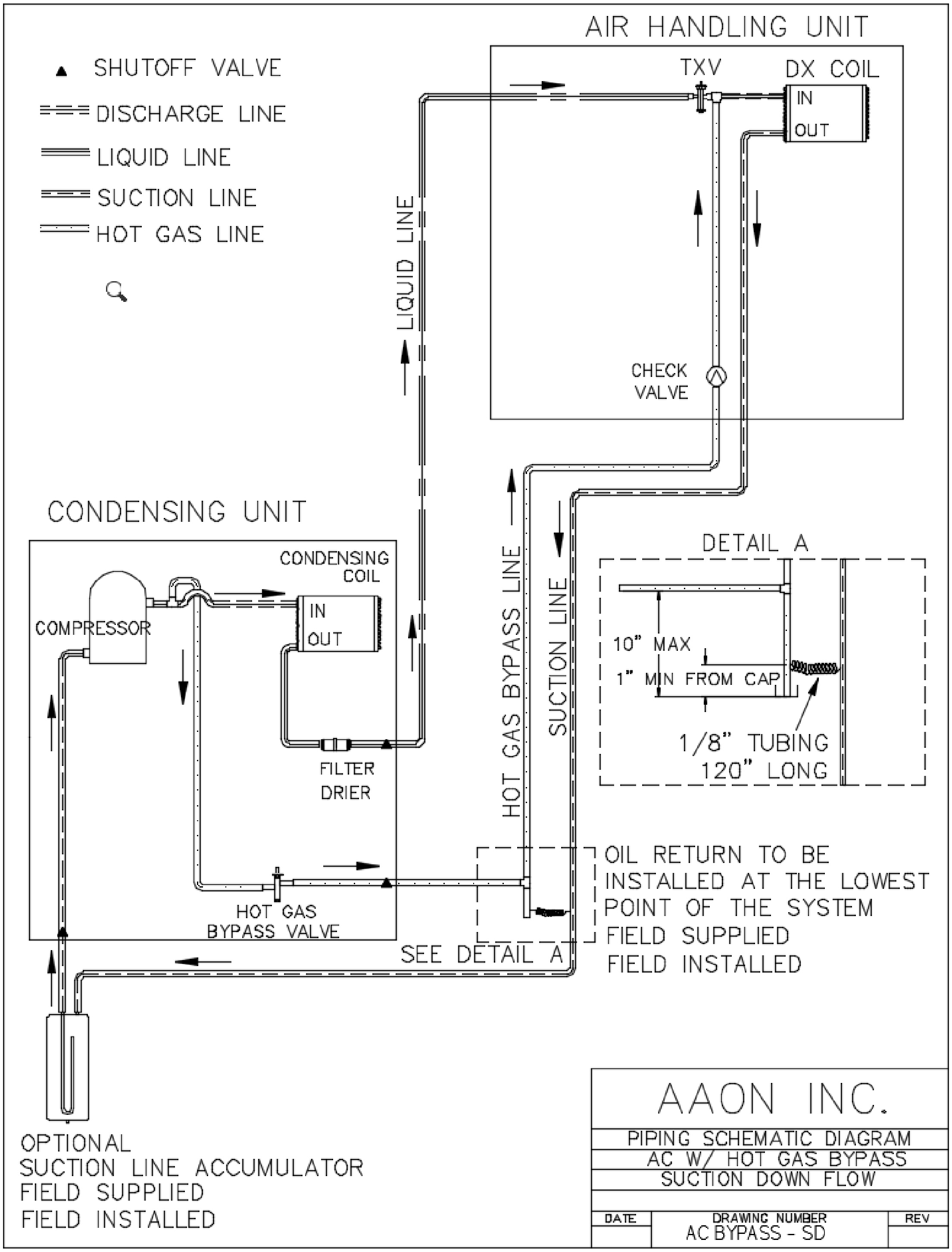


Figure 72 - Split System Hot Gas Bypass Piping Diagram

13 Split System - Air-Source Heat Pump

Some split systems include air-source heat pump as an option. The condensing unit will contain the outdoor coil, condenser fan, the compressor, liquid line receiver, suction line accumulator, reversing valve, TXV with either an internal check valve or an external check valve loop around it, and either a bi-directional filter-drier, or two single direction filter-driers with external check valve loop around them. The air handling unit will contain the indoor coil, a supply fan, and TXV with either an internal check valve or an external check valve loop around it. The check valve around the TXV allows refrigerant to bypass the TXV when the coil is being used as a condenser coil instead of an evaporator coil.

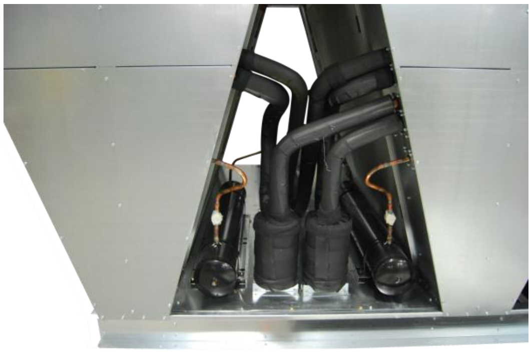


Figure 73 - Air-Source Heat Pump Condensing Unit

Copper piping must be run between the air handling unit copper stub outs and the condensing unit copper stub outs. The suction line and liquid line must be field piped. The suction line serves as both the suction line in cooling mode and the discharge line in heating mode. Similarly, the discharge line (cooling mode) that is factory piped in the condensing unit serves as a suction line in heating mode.

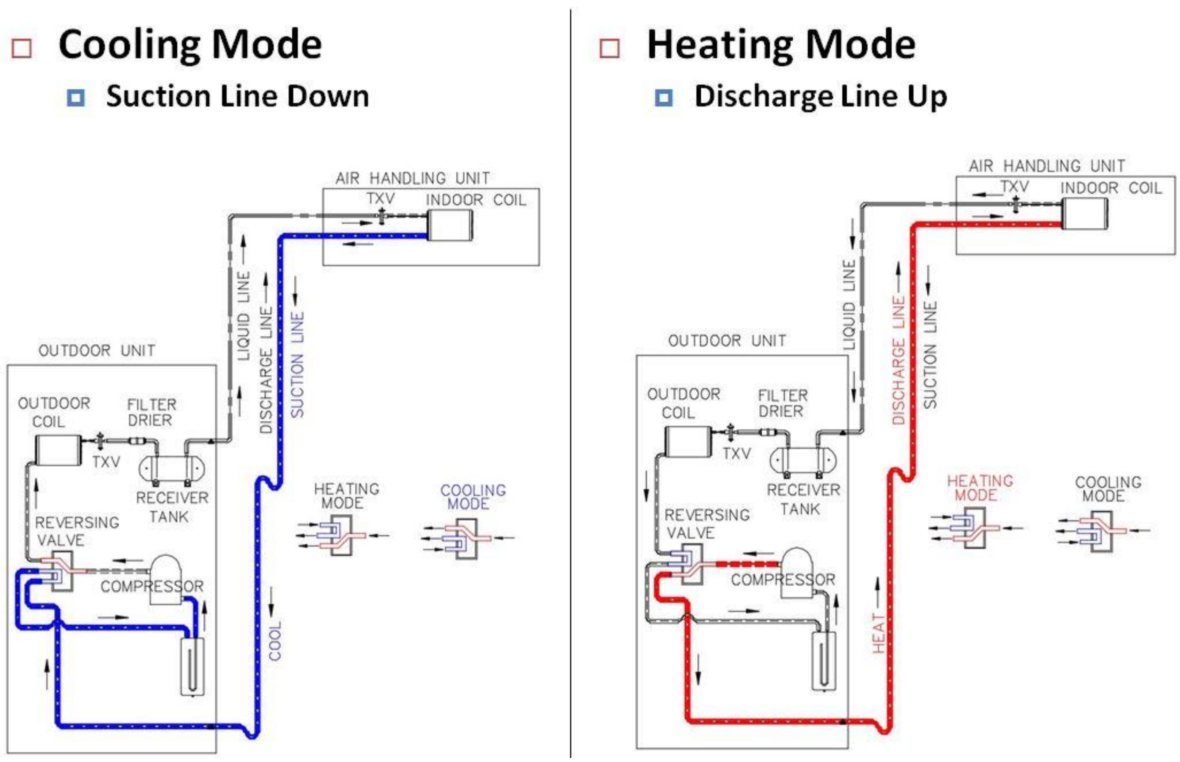


Figure 74 - Split System Heat Pump Suction/Discharge Line

How Does Air-Source Heat Pump Work?

An air-source heat pump split system is an operating system that is able to use the indoor coil as either a cooling coil in summer or a heating coil in winter. The reversing valve changes the direction of the refrigerant flow from the compressor discharge. When the reversing valve is energized, it is in cooling mode and the refrigerant flow discharges from the compressor to the outdoor coil. When the reversing valve is de-energized, it is in heating mode and the refrigerant flow discharges from the compressor to the indoor coil, which at that point acts as a condenser coil. Both the indoor and outdoor coils are designed with distributing tubes and TXVs because in cooling mode the indoor coil acts as a DX coil and in heating mode, the outdoor coil acts as a DX coil. The TXVs have check valves so that when the coils operate as a condensing coil, the refrigerant can bypass the TXVs. The refrigerant flow in the liquid line also changes directions and that is why the filter-driers are bi-directional. The liquid line receiver and the suction line accumulator protect the compressor especially against issues that arise when the system switches from heating to cooling or from cooling to heating.

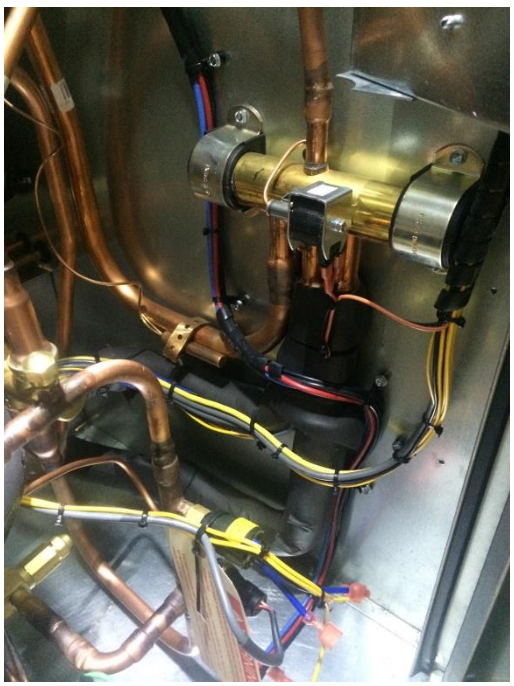


Figure 75 - Reversing Valve Installed in Condensing Unit

Why Select Air-Source Heat Pump?

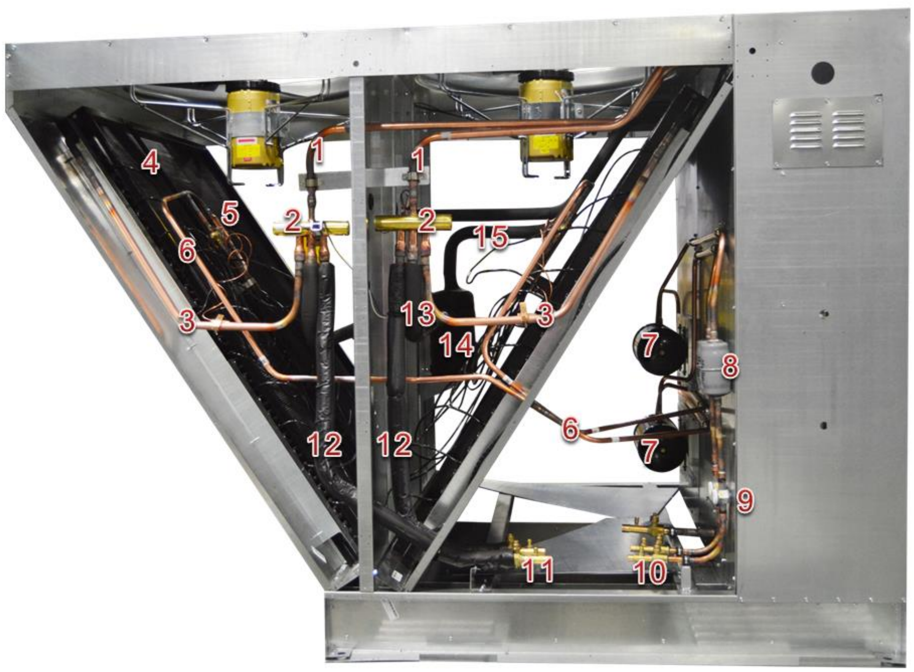
Air-source heat pumps provide energy efficient cooling and heating through the unit's refrigeration circuit. Efficiency is measured by coefficient of performance (COP) which is a measurement of the amount of useful heat supplied by the system (Q) divided by the work required to run the system (W).

COP = Q/W

Because heat pump systems take advantage of the refrigeration cycle, the COP can be up to 4.0. By comparison, electric heat is 1.0 and gas heat can be as high as 0.98. Heat pump units may incorporate another source of auxiliary heat in situations where more heat is needed.

Cooling Mode of Operation

The picture can help us to follow the path of refrigerant and to identify components within the heat pump system. Refrigerant discharges from the compressor through (1) discharge line, and enters (2) the reversing valve. In cooling mode of operation, the discharge gas will travel through (3) to (4) the condenser coil. The liquid refrigerant exits through (5) the bypass check valve around the TXV and then into (6) the liquid line. The liquid refrigerant enters (7) the liquid line receiver, then the (8) liquid line filter-drier, then the (9) sight glass, and finally to (10) the liquid line shut-off valves. The low pressure suction gas, which is coming from the evaporator coil of the indoor air handling unit travels through field installed piping and then enters through (11) the shut-off valves. From there it travels through (12) the suction line and into the (2) reversing valve. It will make the U bend in the (2) reversing valve and travel through (13) into (14) the suction line accumulator. The suction gas leaves (15) the suction line accumulator and enters into the suction side of the compressor.



Heating Mode of Operation

Refrigerant discharges from the compressor through (1) discharge line, and enters (2) the reversing valve. In heating mode of operation, the discharge gas will travel through (12) and to (11) the shut-off valves. The field installed discharge/suction line will carry the discharge gas to the indoor air handling unit to be condensed in the indoor coil. Liquid refrigerant enters the condensing unit through (10) the liquid line shut off valves, goes through the (9) sight glass, (8) liquid line filter-drier, and (7) liquid line receiver. The liquid refrigerant then enters the (5) TXV, goes through the distributer tubes and into (4) the coil which is now acting as an evaporator coil. The low pressure refrigerant gas now travels through (3) and into (2) the reversing valve. It will make the U bend in the (2) reversing valve and travel through (13) into (14) the suction line accumulator. The suction gas leaves (15) the suction line accumulator and enters into the suction side of the compressor.

13.1 Reversing Valve

The reversing valve uses a solenoid coil for operation. The solenoid coil is wired to a relay output and it is in cooling mode when energized, and heating mode when de-energized (aka fail to heat). The call moves the U-bend inside the valve to switch the flow of refrigerant. The first picture below is how the valve currently looks (with the top pipe in the center). The cut-away pictures are an older version of the valve that show internal parts of the valve.

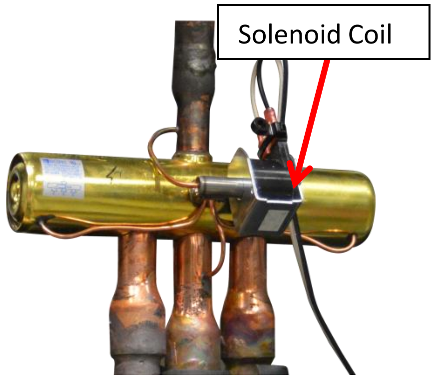


Figure 77 - Reversing Valve

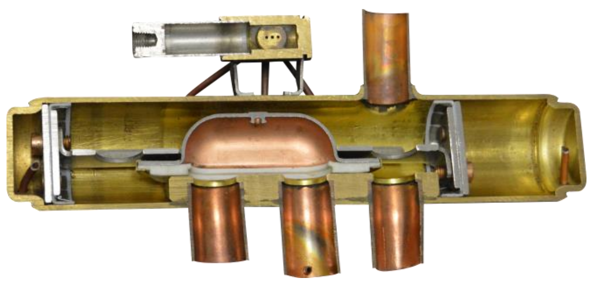


Figure 78 - Reversing Valve Internal Position 1

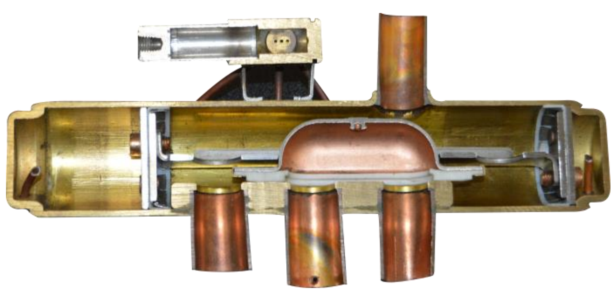


Figure 79 - Reversing Valve Internal Position 2

13.2 Heat Pump Protection

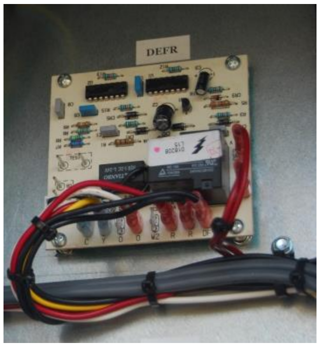


Figure 80 - Defrost Control Board

Defrost Control Board is included with heat pump units without factory provided controls to prevent frost accumulation on the outdoor coil during heat pump heating operation. If the freeze-stat mounted on the condenser coil reads a temperature below 32°F, the board starts a timer. When the timer reaches the field selected time (30/60/80 minutes), the defrost cycle will begin and last for 10 minutes or end when the outdoor coil temperature is above a fixed set point. When factory provided controls are used, the defrost cycle time is optimized. During defrost cycle all compressors energize, reversing valve energizes, and auxiliary heat energizes.

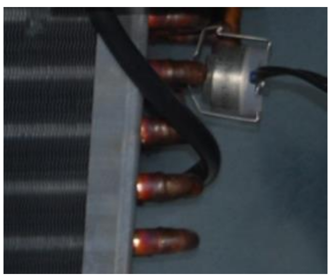


Figure 81 - Freeze-Stat

Freeze-Stat is used for defrost control of the coil in heat pump mode. It is mounted on the return bend of the coil and if it signals a temperature below setpoint, the unit will switch from heat pump mode to cooling mode to defrost the coil.

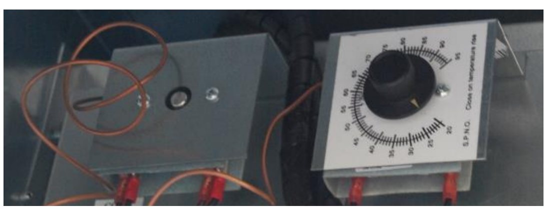


Figure 82 - Compressor Lockouts

Compressor Lockouts are included with heat pump units without factory provided controls. The cooling mode uses a non-adjustable compressor lockout set to 55°F and the heating mode uses an adjustable compressor lockout with a range from 20°F to 95°F. When outdoor conditions reach the lockout ambient temperature, the compressors will shut down and some other source of heating would be necessary.

This is the piping diagram for a typical split system air-source heat pump when the condensing unit is above the air handling unit.

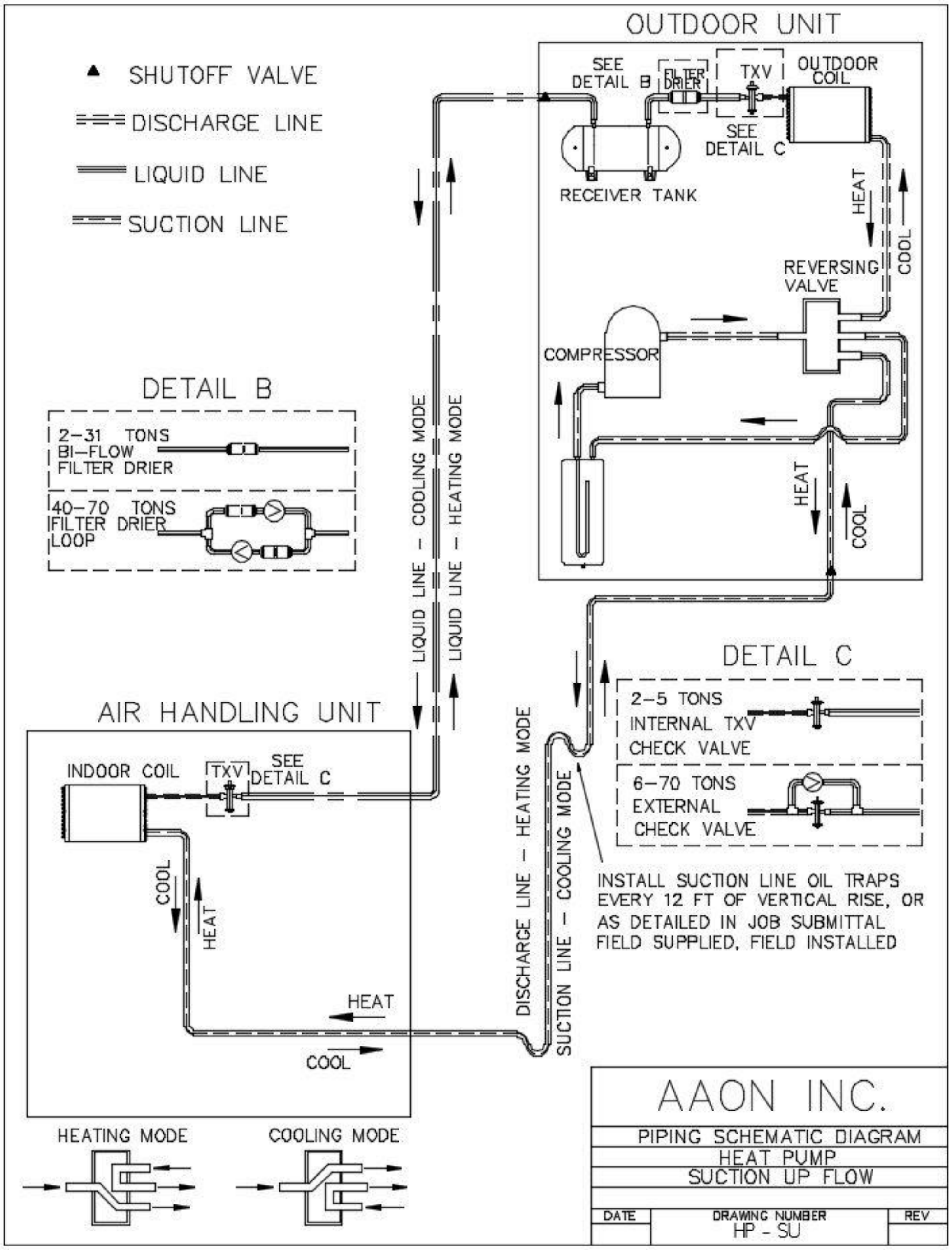


Figure 83 - Split System Heat Pump Piping Diagram

14 Split System - Flooded Condenser Low Ambient Controls (LAC)

Some split systems include flooded condenser low ambient controls as an option. The condensing unit will contain the condensing coil, condenser fan, the compressor, liquid line filter-drier, liquid line receiver, and a low ambient control valve. The air handling unit will contain the evaporator coil, a supply fan, and the TXV.

Copper piping must be run between the air handling unit copper stub outs and the condensing unit copper stub outs. The suction line and liquid line must be field piped. The discharge line is factory piped inside the condensing unit.

How Does Flooded Condenser Low Ambient Controls Work?

Flooded condenser low ambient control maintains normal head pressure during periods of low ambient. When the ambient temperature drops, the condensing temperature and therefore pressure drops. Without ambient control, the system would shut down on low discharge pressure or low suction pressure.

Table 1 - Condenser Flooding

PERCENTAGE OF CONDENSER TO BE FLOODED								
Ambient Temperature (°F)	Evaporating Temperature (°F)							
	0°	**10°**	**20°**	**30°**	**35°**	**40°**	**45°**	**50°**
70°	40	24	0	0	0	0	0	0
60°	60	47	33	17	26	20	10	4
50°	70	60	50	38	45	40	33	28
40°	76	68	60	50	56	52	46	42
30°	80	73	66	59	64	60	55	51
20°	86	77	72	65	69	66	62	59
0°	87	83	78	73	76	73	70	68

In order to maintain head pressure in the refrigeration system, liquid refrigerant is backed up in the condenser coil to reduce condenser coil surface. The following chart shows the percentage that a condenser must be flooded in order to function properly at the given ambient temperature.

During higher ambient temperatures the entire condenser is required to condense refrigerant. During these higher ambient temperatures, a receiver tank is used to contain the refrigerant that is required to flood the condenser during low ambient operation. The receiver must be sized to contain all of the flooded volume otherwise there will be high head pressures during higher ambient conditions.

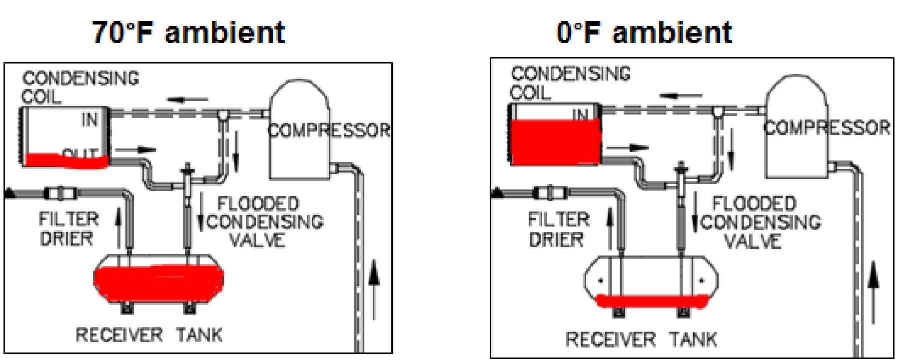


Figure 84 - Flooded Condenser Low Ambient Control Schematic

Why Select Flooded Condenser Low Ambient Controls?

Flooded condenser low ambient control allows cooling operation with ambient temperatures down to 0°F. The most common application for this kind of system is data centers because they still have a need for cooling even in winter conditions due to the heat load inside the room.

14.1 LAC Valve

The low ambient control (LAC) valve is a modulating 3-way pressure activated valve. It can either be activated by the discharge pressure (LAC-4) or the receiver pressure (LAC-5, LAC-10). In either case, when the ambient temperatures drop, the valve allows discharge gas to bypass around the condenser and into the receiver tank. Mixing of the discharge gas with liquid inside the receiver creates a high pressure at the condenser outlet, reducing the flow and causing liquid to back up into the condenser coil.

Figure 85 - LAC-10 Valve

See Sporlan Bulletin 90-30 for more detailed information.

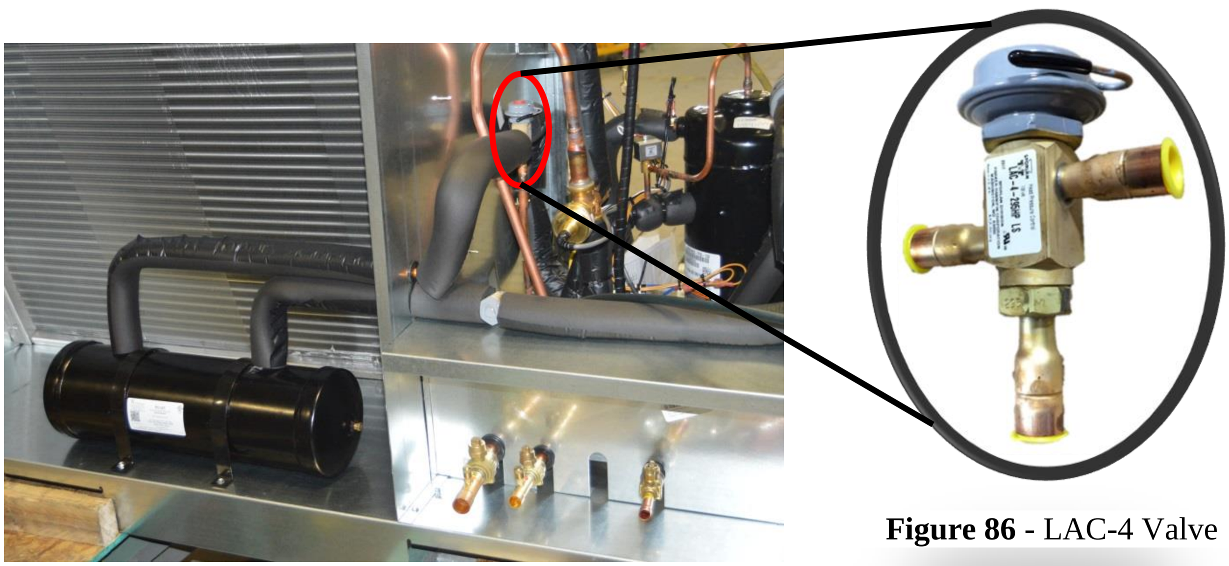


Figure 86 - LAC-4 Valve

This is the piping diagram for a typical split system with low ambient controls when the condensing unit is above the air handling unit. (Suction Up, Liquid Down)

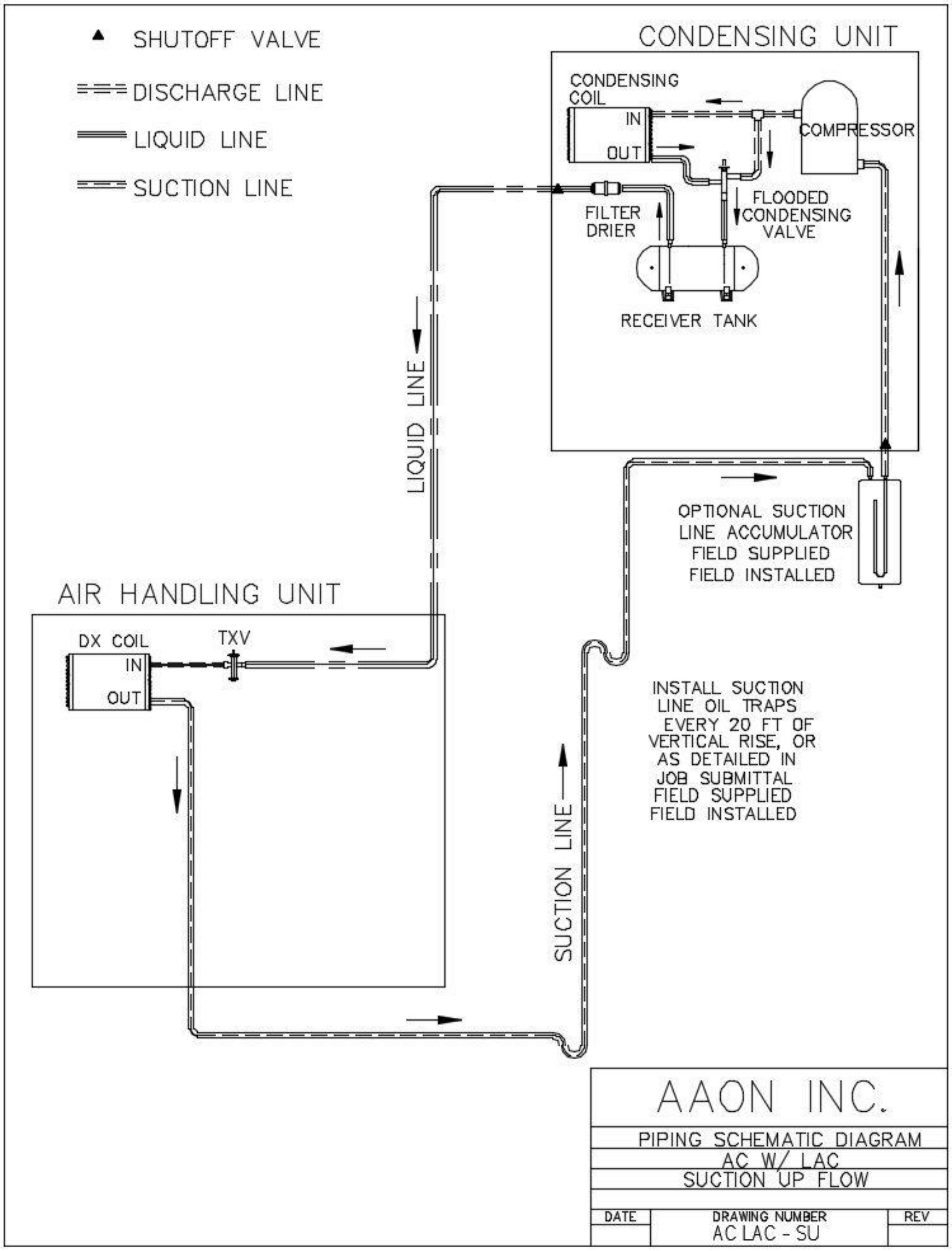


Figure 87 - Split System with LAC Piping Diagram

15 General Line Sizing Discussion

Refrigerant line sizing is critical for protecting the compressors and ensuring smooth TXV operation.

Piping from the condensing unit to the air handler is the responsibility of the installing contractor.

Use only clean type "ACR" copper tubing that has been joined with high temperature brazing alloy.

The pipe or line sizes must be selected to meet the actual installation conditions and NOT simply based on the connection sizes at the condensing unit or air handler.

Care must be taken not to cross the circuits on multiple circuit systems.

Upon completion of piping connection, the interconnecting piping and air handler MUST BE evacuated to 500 microns or less; leak checked and charged with refrigerant.

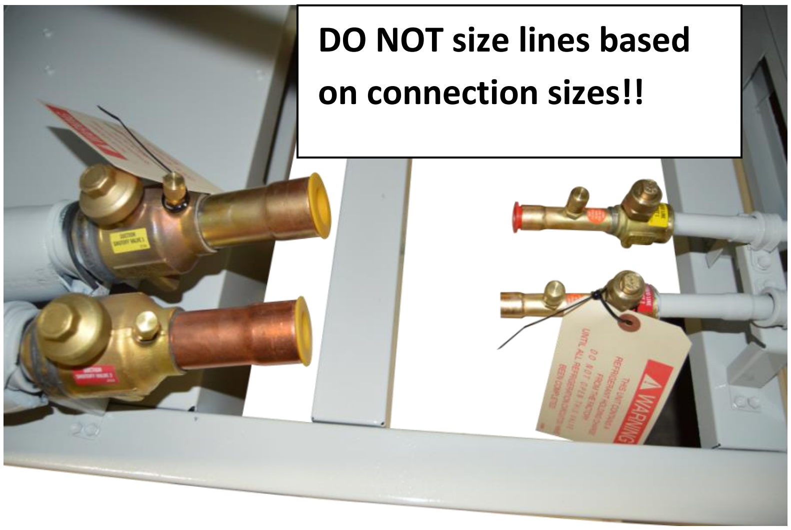


Figure 88 - Condensing Unit Connections

When sizing the piping between the condenser and air handling unit the following must be considered:

1. Continuous oil return, **and**

2. Minimize pressure drop, **and**

3. Prevention of liquid refrigerant slugging, or carryover

In sizing refrigerant lines, cost considerations favor keeping line sizes as small as possible (less refrigerant in the system and lower material costs). However, suction and discharge line pressure drops cause loss of compressor capacity and increased power usage. Excessive liquid line pressure drops can cause the liquid refrigerant to flash, resulting in faulty expansion valve operation.

The following pages discuss the logic behind sizing refrigerant lines properly. It is extremely important and beneficial to understand. The AAON ECat Split System selection program uses this logic when it provides line sizes. Knowing the following information helps to select lines wisely, based on the specific application and line routing.

Liquid lines - The most important considerations when sizing the liquid lines are minimizing the refrigerant charge and avoiding flashing in the line before the thermostatic expansion valve. Minimizing refrigerant charge can be achieved by minimizing the liquid line diameter. However, reducing the pipe diameter will increase the velocity of the liquid refrigerant which increases the frictional pressure drop in the liquid line, and causes other undesirable effects such as noise.

Managing the pressure loss in the liquid line is critical to ensuring sufficient sub-cooling, avoiding flashing upstream of the TXV, and maintaining system efficiency. Pressure losses through the liquid line due to frictional contact, installed accessories, and vertical risers are inevitable. Maintaining adequate sub-cooling to overcome these losses is the only method to ensure that liquid refrigerant reaches the TXV.

If the air handling unit is above the condensing unit, liquid refrigerant travels upwards in a riser and loses head pressure. The higher the vertical rise, the higher the pressure loss, and the greater risk of flash gas in the liquid line.

If the air handling unit is below the condensing unit, with the liquid line flowing down, the gravitational force will increase the pressure of the liquid refrigerant. This will allow the refrigerant to withstand greater frictional losses without the occurrence of flashing prior to the TXV.

Suction lines - The suction line is more critical than the liquid line from a design and construction standpoint. Suction lines with vertical risers must be trapped every 20 ft to facilitate oil return. More care must be taken to ensure that adequate velocity is achieved to return oil to the compressor at minimum loading conditions. However, reducing the piping diameter to increase the velocity at minimal load can result in excessive pressure losses, capacity reduction, and noise at full load.

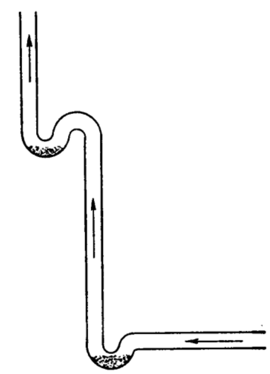


Figure 89 - Suction Line Trap

It is important to consider part load operation when sizing suction lines. At minimum capacity, refrigerant velocity may not be adequate to return oil up the vertical riser. Decreasing the diameter of the vertical riser will increase the velocity, but also the frictional loss. At points where small pipe size can be used to provide sufficient velocity to return oil in vertical risers at part loads, greater pressure losses are incurred at full loads. This can be compensated for by over sizing the horizontal runs, while using the smaller pipe size on the vertical risers. Transition to the smaller diameter at the horizontal pipe right before the trap, and then install an expander at the top elbow that transitions to horizontal again.

Make sure to provide support to maintain suction line positioning.

Suction Line Double Risers - For difficult line routing applications, a double suction riser can be applied to the situation of part load operation with a suction riser. A double suction riser is designed to return oil at minimum load while not incurring excessive frictional losses at full load. A double suction riser consists of a small diameter riser in parallel with a larger diameter riser, and a trap at the base of the large riser. At minimum capacity, refrigerant velocity is not sufficient to carry oil up both risers, and it collects in the trap, effectively closing off the larger diameter riser, and diverting refrigerant up the small riser where velocity of the refrigerant is sufficient to maintain oil flow. At full load, the mass flow clears the trap of oil, and refrigerant is carried through both risers. The smaller diameter pipe should be sized to return oil at minimum load, while the larger diameter pipe should be sized so that flow through both pipes provides acceptable pressure drop at full load. The inverted trap at the top is to prevent back flow into the inactive line in the double suction riser.

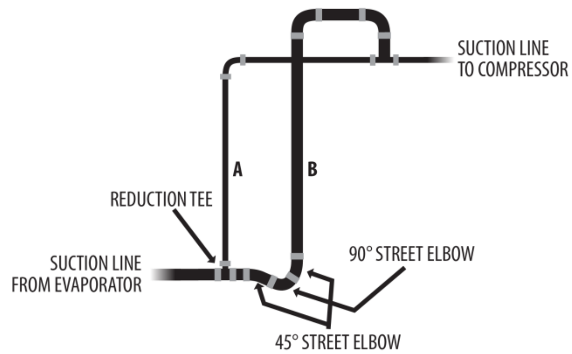


Figure 90 - Double Suction Riser Construction

Suction Line Double Riser for Heat Pumps - A double riser used for heat pump operation is different than described above. The specific volume (ft^3/lb) of refrigerant at the discharge temperature (heating mode line conditions) is significantly lower than the specific volume at the suction temperature (cooling mode line conditions). To compound the issue, the capacity in heating mode is lower than the capacity in cooling mode. The discharge velocity in the riser during heating mode is much lower than the suction velocity during cooling mode. A double riser may be necessary to get acceptable velocities for the discharge mode and acceptable velocities for the suction mode. In **Figure 91** & **Figure 92**, the cooling mode will use both lines, and the heating mode will use only one.

Discharge lines - The discharge line is the least critical line to size. Pressure losses in this line do not have a large effect on capacity, and since the temperatures are higher the oil viscosity is lower and requires less drag force (pressure difference) to return the oil.

Discharge lines should be sized to ensure adequate velocity of refrigerant to ensure oil return, avoid excessive noise associated with velocities that are too high, and to minimize efficiency losses associated with friction.

Heat Pump Cooling Mode

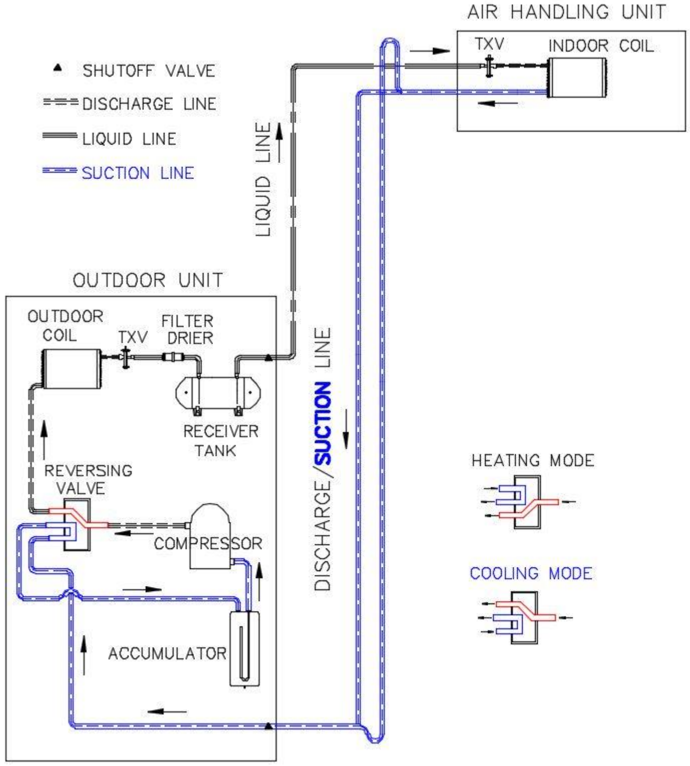


Figure 91 - Heat Pump Piping Schematic of Suction Vapor Flow Down in Double Riser

Heat Pump Heating Mode

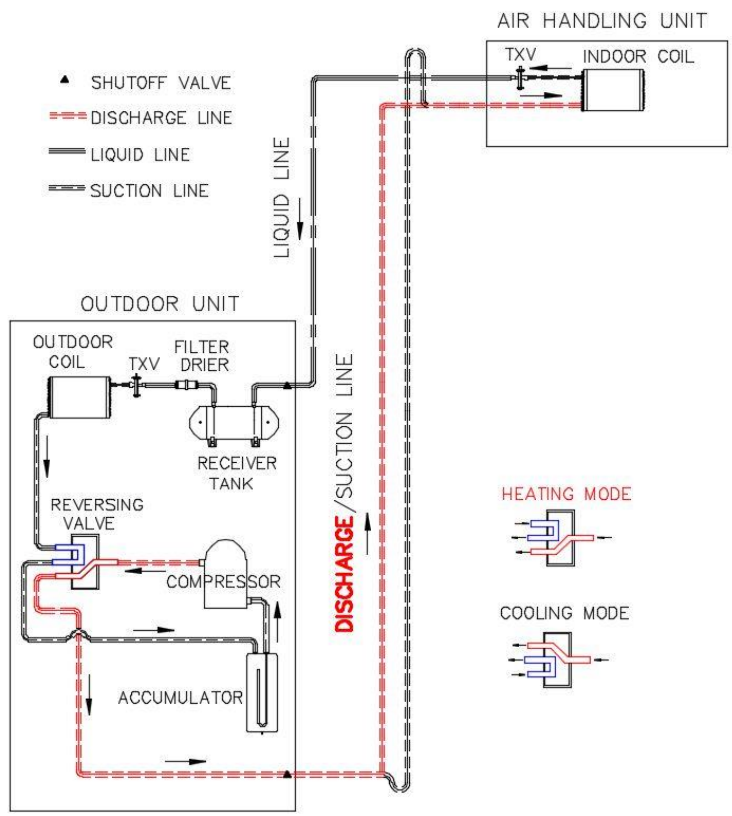


Figure 92 – Heat Pump Piping Schematic of Discharge Vapor Flow Up in Double Riser

Hot Gas Reheat line - The hot gas reheat line is a discharge line with modulating refrigerant flow based on the need for hot gas reheat. The AAON modulating hot gas reheat system diverts hot discharge gas from the compressor to the reheat coil in the air handling unit through the hot gas line. Field piping between the condensing unit and the air handler is required.

Hot Gas Bypass line - The hot gas bypass line is a discharge line with modulating refrigerant flow based on the need for hot gas bypass. Hot Gas Bypass is available for use with DX systems that may experience low suction pressure during the operating cycle. This may be due to varying load conditions associated with VAV applications or units supplying a large percentage of outside air. The system is designed to divert refrigerant gas from the compressor discharge to the low pressure side of the system in order to keep the evaporator from freezing and to maintain adequate refrigerant velocity for oil return at minimum load.

Hot discharge gas is redirected to the evaporator inlet via an auxiliary side connector (ASC) to false load the evaporator when reduced suction pressure is sensed. Field piping between the condensing unit and the evaporator is required.

15.1 General ECat Length Limits

The refrigerant line size calculations in ECat use the following limits. If ECat provides no acceptable lines, use EES Refrigerant Piping Calculator.

Some ECat limits:

1. Limit the maximum delta temperature loss in the suction line to 6°F.
2. Limit the maximum vertical distance to 70 ft.
3. Limit liquid line losses to less than 8°F.

15.2 Velocity and ΔTemperature Guidelines

Liquid Line Maximum Velocity Limits:

- From receiver to TXV - 500 fpm
- From receiver to TXV if solenoid valve is in the line - 300 fpm
- From condenser to receiver - 100 fpm

Liquid Line Δ Temperature Loss Due to Friction in Lines + Friction through Valves and Other Accessories + Liquid Head:

- Must be less than 8°F if available sub-cooling is 10°F. Try to ensure at least 2°F sub-cooling entering the TXV.
- Liquid line risers cause approximately 0.43psi/ft of static loss due to head pressure in R-410A systems at 100°F liquid temperature.
- See **Table 14** - Equivalent Length Table for pressure loss due to valves & other accessories.
- Available sub-cooling for an air cooled condensing system is typically around 10°F (because condenser coils are designed to operate at 10°F sub-cooling or more), but available sub-cooling for an air-source heat pump condensing system is only 2-4°F. In this case sub-cooling must be increased by 1°F per 10 ft of vertical liquid rise. A suction-to-liquid Heat Exchanger may be required.

Suction Line Δ Temperature Loss:

- Less than 6°F (a temperature loss also causes a loss in capacity so try to keep this as low as possible).

Suction Line Maximum Velocity Limits:

- For all compressor types, the full load velocities should not exceed 4000 fpm due to noise concern.

Suction Line Minimum Velocity Limits:

- See Table 2 - Suction/Discharge Line Minimum Velocity Limits & Table 3 - Minimum Velocity & Tons for R-410A Oil Return

Discharge Line Maximum Velocity Limits:

- For all compressor types, the full load velocities should not exceed 3500 fpm due to noise concerns.

Discharge Line Minimum Velocity Limits:

- See Table 2 - Suction/Discharge Line Minimum Velocity Limits & Table 3 - Minimum Velocity & Tons for R-410A Oil Return

Hot Gas Bypass Velocity Limits: maximum - 3500 fpm minimum - 2000 fpm

- When installing hot gas bypass line, a Purge circuit must be provided at the lowest point in the system

Hot Gas Reheat Velocity Limits: maximum - 3500 fpm minimum - 2000 fpm

- When installing hot gas reheat line, a Purge circuit must be provided at the lowest point in the system

Suction/Discharge Line Minimum Velocity Limits:

Table 2 - Suction/Discharge Line Minimum Velocity Limits

			Air-Cooled Mode		Heat Pump Mode*	
Compressor Type	Minimum Load	Horizontal (fpm)	Suction Down (fpm)	Suction Up (fpm)**	Discharge Down (fpm)	Discharge Up (fpm)
Digital	10% this is not what is used for velocity	Minimum velocity for oil return		1500 fpm	Minimum velocity for oil return	900 fpm
Digital Even Tandem	5% this is not what is used for velocity	Minimum velocity for oil return with only one compressor running		1500 fpm with only one compressor running	Minimum velocity for oil return with only one compressor running	900 fpm with only one compressor running
On/Off	100%	Minimum velocity for oil return for the specific line size as found in **Table 3**				
Even Tandem	50% for even tandems	Minimum velocity for oil return with only 50% of capacity for the specific line size as found in **Table 3**				
2-step	67%	Minimum velocity for oil return with only 67% of capacity for the specific line size as found in **Table 3**				
AC Motor VFD driven	50%	Minimum velocity for oil return with only 50% of capacity for the specific line size as found in **Table 3**				

Notes:

The minimum velocity for oil return values are absolute minimum velocities for oil return. Do not design lines at the minimum velocities due to the many possible operating conditions.

*Heat pumps use the suction line from cooling mode as the discharge line for heating mode. The line must be checked in both modes of operation and an acceptable line chosen that fits both criteria for maximum and minimum velocities. If none will work, a double riser must be considered.

**The digital compressor suction up minimum velocity limit of 1500 fpm is the minimum and requires traps. As a precaution raise system capacity to 100% for five minutes every 24 hours.

Table 3 - Minimum Velocity & Tons for R-410A Oil Return
This table does not apply to digital scroll compressors.

Line OD (in)	Minimum Velocity for Oil Return (fpm)		Minimum Tons for Oil Return (tons)	
	Suction Line*	Discharge Line**	Suction Line*	Discharge Line**
3/8	253	223	0.09	0.15
1/2	320	261	0.23	0.33
5/8	377	292	0.44	0.60
3/4	427	317	0.72	0.95
7/8	476	341	1.11	1.41
1-1/8	569	384	2.26	2.72
1-3/8	646	419	3.79	4.36
1-5/8	723	451	6.01	6.66
2-1/8	873	512	13.04	13.58
2-5/8	735	559	16.52	22.29
3-1/8	830	606	27.29	35.35
3-5/8	914	646	40.62	50.95
4-1/8	993	682	57.38	69.97
5-1/8	1142	748	102.80	119.6

These are absolute minimum velocities for oil return. Do not design lines at the minimum velocities due to the many possible operating conditions.

***Cooling Mode Suction Line Minimum Velocity Conditions:**

0.35psi/100ft for lines 2" ID and under
0.20psi/100ft for lines above 2" ID
Saturated Suction Temperature = 35°F
Superheat = 15°F

AAON does not allow cooling mode SST below 35°F.

****Heating Mode Discharge Minimum Velocity Line Conditions:**

Saturated Condensing Temp = 75°F
Sub-cooling = 15°F
Saturated Suction Temperature = -10°F
Superheat = 10°F

These are the most extreme conditions that AAON allows for heat pump heating mode operation.

15.3 Line Routing & Sizing - AHU above Condensing Unit (Cooling Only)

- Liquid line up -
  - No traps needed
  - No pitch needed, but can be pitched to match the other lines
  - When the system is shut down, gravity will pull liquid down the vertical column, and back to the condenser when it is below the evaporator. This could potentially result in compressor flooding. A check valve can be installed in the liquid line where the liquid column rises above the condenser to prevent this.
  - Size the liquid line for less than 500 fpm velocity
    - Less than 300 fpm velocity if solenoid or other electrically operated valves in the line
    - Less than 100 fpm velocity if line is between condenser and receiver
  - Liquid line risers cause static loss due to head pressure. This provides less sub-cooling, so be sure to check the pressure loss in the liquid line, including the accessories (filter-drier, valves, any other accessory installed in the liquid line that could cause pressure loss). Make sure the system has enough sub-cooling to overcome the frictional & head pressure losses. If it doesn't, add a Heat Exchanger to gain more sub-cooling in the liquid line.

- Suction line down -
  - An inverted trap that rises slightly above the top of the evaporator helps prevent liquid refrigerant from migrating to the compressor during the off cycle. **This practice is not necessary on AAON equipment because AAON designs the evaporator coils with the refrigerant exiting from the top of the evaporator coil. If the evaporator coil is not an AAON coil, use this piping practice.**
  - If a riser is taken directly upward from an evaporator, provide a short horizontal section of tubing and a trap ahead of the riser so the thermostatic expansion valve bulb can be installed in the short horizontal section. The trap serves as a drain area, and helps to prevent the accumulation of liquid under the bulb which could cause erratic expansion valve operation.
  - Pitch horizontal sections 1 inch per 20 feet in the direction of flow to maintain oil flow towards the compressor.
  - Recommended line sizes will keep the temperature loss less than 2°F to minimize capacity loss
  - Since suction flow is down, oil return is not as big of a concern as when suction flow is up. Keep velocities above the minimum velocity for oil return. **Table 3**

Line Routing & Sizing - AHU above Condensing Unit (Cooling Only) Continued

- Hot Gas Reheat up -
  - Purge circuit must be installed for oil return.
  - Pitch horizontal sections 1 inch per 20 feet in the direction of flow (this will be pitched in the opposite direction as the suction line)
  - Keep velocity between 2000 fpm - 3500 fpm

- Hot Gas Bypass up -
  - Purge circuit must be installed for oil return.
  - Pitch horizontal sections 1 inch per 20 feet in the direction of flow (this will be pitched in the opposite direction as the suction line)
  - Keep velocity between 2000 fpm - 3500 fpm

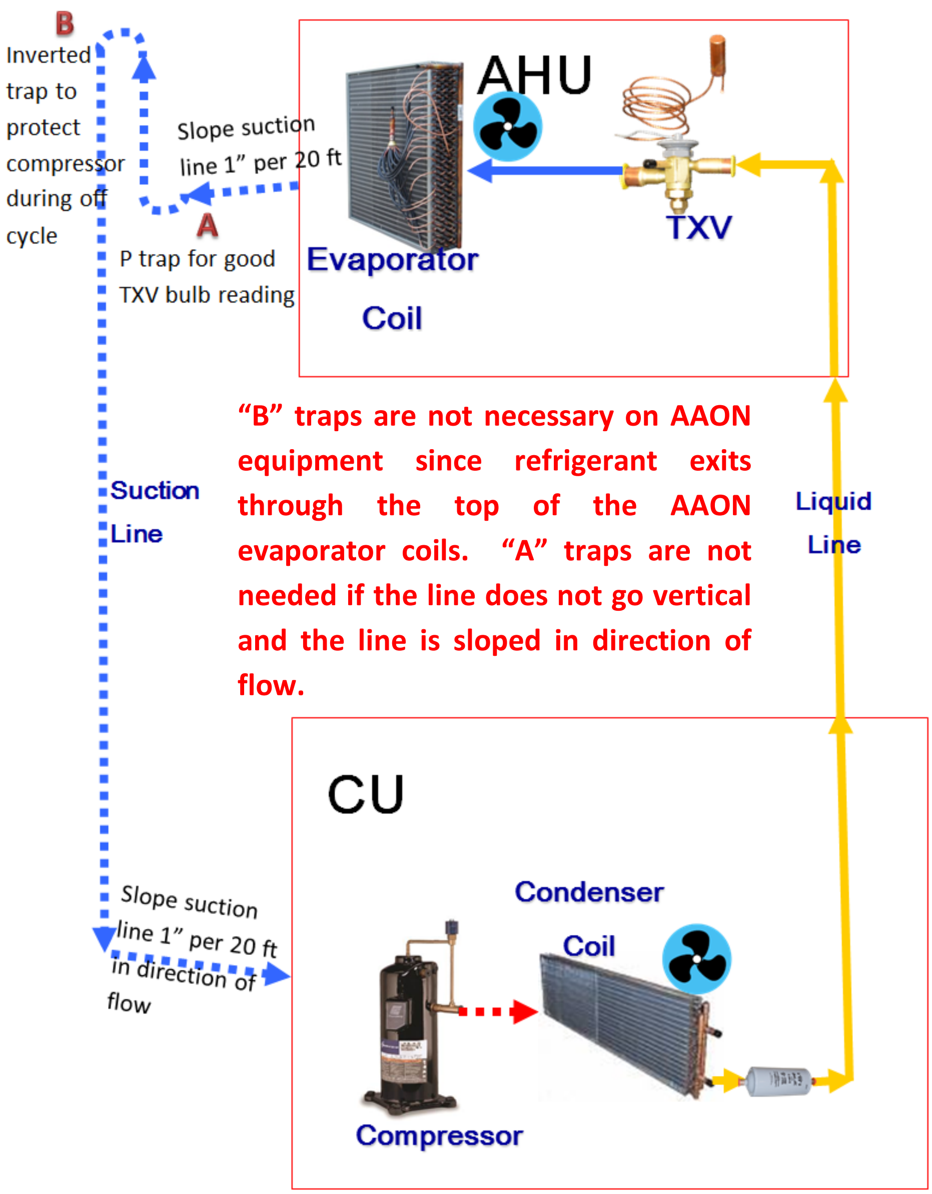


Figure 93 - Split System Line Routing AHU above CU

The purpose of Trap "A" is to keep liquid oil away from the TXV sensing bulb and to facilitate oil return up the riser. Liquid refrigerant and oil around the sensing bulb will cause the TXV to operate erratically.

The purpose of inverted Trap "B" is to prevent liquid refrigerant from draining back to the compressor by gravity during the off cycle. The inverted trap must rise to the top of the evaporator coil. This trap is not necessary on a pump-down system. **This practice is not necessary on AAON equipment because AAON designs the evaporator coils with the refrigerant exiting from the top of the evaporator coil. If the evaporator coil is not an AAON coil, use this piping practice.**

Line Routing & Sizing - AHU below Condensing Unit (Cooling Only)

- Liquid line down -
  - No traps needed
  - No pitch needed, but can be pitched to match the other lines
  - Size the liquid line for less than 500 fpm velocity
    - Less than 300 fpm velocity if solenoid or other electrically operated valves in the line
    - Less than 100 fpm velocity if line is between condenser and receiver
  - Since the liquid line will travel down, the liquid refrigerant will gain sub-cooling due to the head pressure. This is an advantage because more sub-cooling will ensure 100% liquid entering the TXV.
- Suction line up -
  - P-trap at the bottom of vertical risers for oil return.
  - If a riser is taken directly upward from an evaporator, provide a short horizontal section of tubing and a trap ahead of the riser so the thermostatic expansion valve bulb can be installed in the short horizontal section. The trap serves as a drain area, and helps to prevent the accumulation of liquid under the bulb which could cause erratic expansion valve operation.
  - Trap every 20 feet of vertical rise
  - Pitch horizontal sections 1 inch per 20 feet in the direction of flow to maintain oil flow towards the compressor
  - Recommended line sizes will keep the temperature loss less than 2°F to minimize capacity loss. (ECat allows up to 6°F)
  - Velocity is very important in suction line up systems. The line must be sized to return oil at minimum loads, but also to minimum pressure loss at full load.
    - Full load velocity < 4000 fpm
      - Digital Compressors - full load velocity > 1500 fpm
      - Tandem Digital Compressors - full load velocity > 3000 fpm or use double suction riser with 1500 fpm in each pipe
        - When circuits with digital compressors exceed 40 ft of vertical rise in the suction line, use controls to raise the system to full capacity for 5 minutes every 24 hours.

Line Routing & Sizing - AHU below Condensing Unit (Cooling Only) Continued

- Suction line up (continued) -
  - Minimum load velocity must be greater than the minimum velocity for oil return. **Table 3**
    - On/Off Compressors - minimum load velocity = full load velocity
    - Even Tandem Compressor - minimum load velocity = 50% of full load velocity
    - 2-step Compressor - minimum load velocity = 67% of full load velocity
    - Typical AC motor VFD Driven Compressor - minimum load velocity = 50% of full load velocity (As compressor technology changes, this turndown can also change. Check the specific compressor capabilities to determine the % of full load velocity).
    - Digital Compressors are only checked at full load velocity. The minimum load velocity table does not apply to Digital Compressors. (see full load velocity limits above)

- Hot Gas Reheat down -
  - Purge circuit must be installed for oil return.
  - Pitch horizontal sections 1 inch per 20 feet in the direction of flow (this will be pitched in the opposite direction as the suction line)
  - Keep velocity between 2000 fpm - 3500 fpm

- Hot Gas Bypass down -
  - Purge circuit must be installed for oil return.
  - Pitch horizontal sections 1 inch per 20 feet in the direction of flow (this will be pitched in the opposite direction as the suction line)
  - Keep velocity between 2000 fpm - 3500 fpm

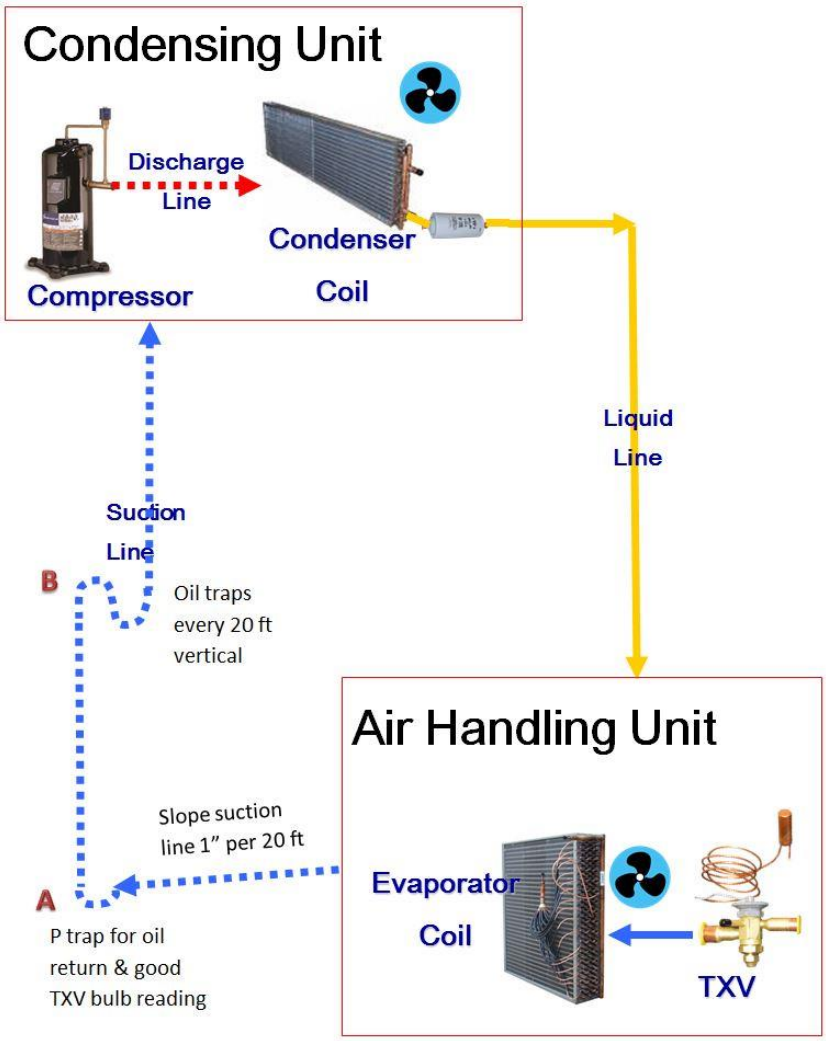


Figure 94 - Split System Line Routing AHU below CU

The purpose of Trap "A" is to keep liquid oil away from the TXV sensing bulb and to facilitate oil return up the riser. Liquid refrigerant and oil around the sensing bulb will cause the TXV to operate erratically.

The purpose of Trap "B" is to help the oil move in vertical rises over 20 ft. If refrigerant velocity is too low, oil drops out of the refrigerant into the trap. The oil will begin to fill the trap, making the internal area smaller and the refrigerant velocities at that location increase. The increased refrigerant velocities will help move the oil to the next level of traps until the oil makes it back to the compressor.

A disadvantage of adding traps is that every bend in the copper adds equivalent length and added pressure drop to the system.

15.4 Heat Pump Considerations

The suction / discharge line must be able to transport refrigerant in cooling mode and heating mode. The line must be sized for the cooling capacity tonnage as a suction line, and then it must be sized for the heating capacity tonnage as a discharge line. The best line size for the discharge line usually does not match the best line size for the suction line, but the same line is used for both scenarios so what is the solution?

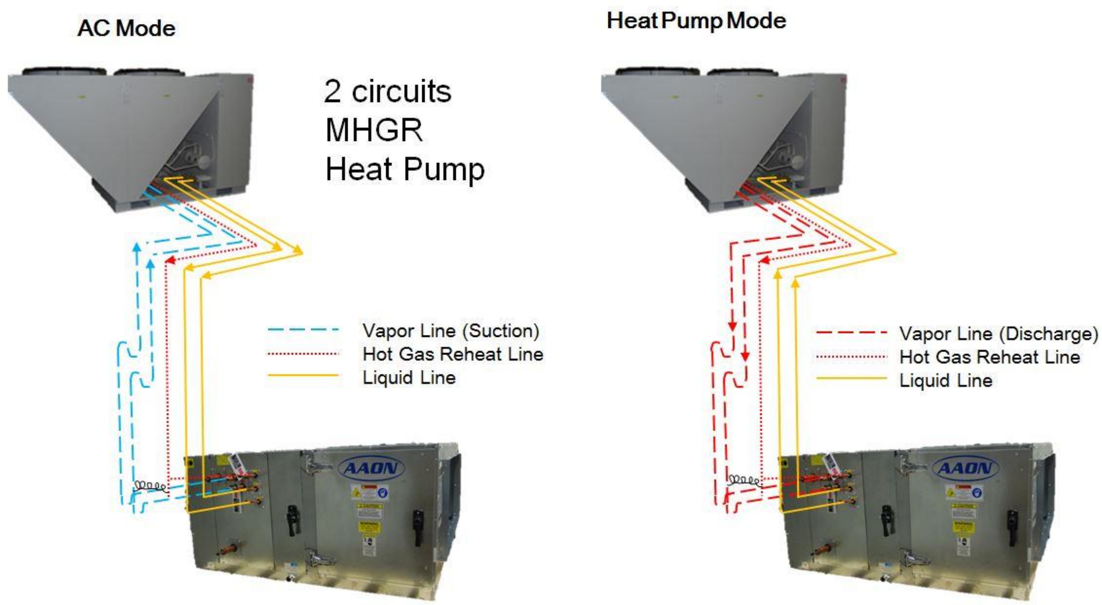


Figure 95 - Heat Pump Suction/Discharge Line

A suction line must be selected that meets the suction line maximum velocity limits at full load and also meets the suction line minimum velocity limits at minimum load. A heat pump unit must meet the suction line minimum and maximum limits and the discharge minimum and maximum line limits. Discharge minimum velocities can be lower since the viscosity of the oil is lower at the temperature of the discharge gas and thus more easily transported.

15.5 Line Routing & Sizing - AHU above Condensing Unit (Heat Pump)

- Liquid line up in cooling / down in heating -
    - No traps needed
    - No pitch needed
    - When the system is shut down, gravity will pull liquid down the vertical column, and back to the condenser when it is below the evaporator. This could potentially result in compressor flooding. A check valve can be installed in the liquid line where the liquid column rises above the condenser to prevent this.
    - Size the liquid line for less than 500 fpm velocity
        - Less than 300 fpm velocity if solenoid or other electrically operated valves in the line
        - Less than 100 fpm velocity if line is between condenser and receiver
    - Liquid line risers cause static loss due to head pressure. This provides less sub-cooling, so be sure to check the pressure loss in the liquid line, including the accessories (filter-drier, valves, any other accessory installed in the liquid line that could cause pressure loss). Make sure the system has enough sub-cooling to overcome the frictional & head pressure losses. If it doesn't, add a Heat Exchanger to gain more sub-cooling in the liquid line.

- Suction line down cooling / Discharge line up heating-
    - An inverted trap that rises slightly above the top of the evaporator helps prevent liquid refrigerant from migrating to the compressor during the off cycle. **This practice is not necessary on AAON equipment because AAON designs the evaporator coils with the refrigerant exiting from the top of the evaporator coil. If the evaporator coil is not an AAON coil, use this piping practice.**
    - If a riser is taken directly upward from an evaporator, provide a short horizontal section of tubing and a trap ahead of the riser so the thermostatic expansion valve bulb can be installed in the short horizontal section. The trap serves as a drain area, and helps to prevent the accumulation of liquid under the bulb which could cause erratic expansion valve operation.
    - Trap every 12 feet of vertical rise
    - Do not pitch the lines since refrigerant flows in both directions for heat pumps.

Line Routing & Sizing - AHU above Condensing Unit (Heat Pump) Continued

- Suction line down cooling / Discharge line up heating- continued
  - Heating Mode Conditions - Discharge Line Up
    - Using heating mode conditions, the discharge line must be sized to return oil at minimum loads, but also to minimize pressure loss and sound at full load.
    - Full load discharge velocity < 3500 fpm
      - Digital Compressors - full load velocity > 900 fpm
      - Tandem Digital Compressors - full load velocity > 1800 fpm or use double riser with 900 fpm in each pipe
        - When circuits with digital compressors exceed 40 ft of vertical rise in the discharge line, use controls to raise the system to full capacity for 5 minutes every 24 hours.
    - Minimum load discharge velocity must be greater than the minimum discharge velocity for oil return. **Table 3**
      - On/Off Compressors - minimum load velocity = full load velocity
      - Tandem Compressor - minimum load velocity = 50% of full load velocity
      - 2-step Compressor - minimum load velocity = 67% of full load velocity
      - Typical AC motor VFD Driven Compressor - minimum load velocity = 50% of full load velocity (As compressor technology changes, this turndown can also change. Check the specific compressor capabilities to determine the % of full load velocity).
      - Digital Compressors are only checked at full load velocity. The minimum load velocity table does not apply to Digital Compressors. (see full load velocity limits above)

  - Cooling Mode Conditions - Suction Line Down
    - The suction line uses the same line as the discharge line, so using the discharge line size selected based on the heating mode conditions, check to make sure the suction velocities and pressure loss values are acceptable.
    - Recommended suction line sizes will keep the suction temperature loss less than 2°F to minimize capacity loss.
    - Since suction flow is down, oil return is not as big of a concern as when suction flow is up. Keep suction velocities above the minimum suction velocity for oil return. **Table 3**

Line Routing & Sizing - AHU above Condensing Unit (Heat Pump) Continued

- Hot Gas Reheat up -
  - Purge circuit must be installed for oil return.
  - Pitch horizontal sections 1 inch per 20 feet in the direction of flow (this will be pitched in the opposite direction as the suction line)
  - Keep velocity between 2000 fpm - 3500 fpm
- Hot Gas Bypass up -
  - Purge circuit must be installed for oil return.
  - Pitch horizontal sections 1 inch per 20 feet in the direction of flow (this will be pitched in the opposite direction as the suction line)
  - Keep velocity between 2000 fpm - 3500 fpm

15.6 Line Routing & Sizing- AHU below Condensing Unit (Heat Pump)

- Liquid line down cooling / up in heating-
  - No traps needed
  - No pitch needed
  - Size the liquid line for less than 500 fpm velocity
    - Less than 300 fpm velocity if solenoid or other electrically operated valves in the line
    - Less than 100 fpm velocity if line is between condenser and receiver
  - Liquid line risers cause static loss due to head pressure. This provides less sub-cooling, so be sure to check the pressure loss in the liquid line, including the accessories (filter-drier, valves, any other accessory installed in the liquid line that could cause pressure loss). Make sure the system has enough sub-cooling to overcome the frictional & head pressure losses. If it doesn't, add a Heat Exchanger to gain more sub-cooling in the liquid line.

- Suction line up cooling / Discharge line down heating -
  - P-trap at the bottom of vertical risers for oil return.
  - If a riser is taken directly upward from an evaporator, provide a short horizontal section of tubing and a trap ahead of the riser so the thermostatic expansion valve bulb can be installed in the short horizontal section. The trap serves as a drain area, and helps to prevent the accumulation of liquid under the bulb which could cause erratic expansion valve operation.
  - Trap every 12 feet of vertical rise
  - Do not pitch the lines since refrigerant flows in both directions for heat pumps.

Line Routing & Sizing - AHU below Condensing Unit (Heat Pump) Continued

- Suction line up cooling / Discharge line down heating - continued
  - Cooling Mode Conditions - Suction Line Up
    - Recommended line sizes will keep the temperature loss less than 2°F to minimize capacity loss (ECat allows up to 6°F)
    - Velocity is very important in suction line up systems. The line must be sized to return oil at minimum loads, but also to minimum pressure loss at full load.
    - Full load suction velocity < 4000 fpm
      - Digital Compressors - full load velocity > 1500 fpm
      - Tandem Digital Compressors - full load velocity >3000 fpm or use double suction riser with 1500 fpm in each pipe
        - When circuits with digital compressors exceed 40 ft of vertical rise in the suction line, use controls to raise the system to full capacity for 5 minutes every 24 hours.
    - Minimum load suction velocity must be greater than the minimum suction velocity for oil return. **Table 3**
      - On/Off Compressors - minimum load velocity = full load velocity
      - Tandem Compressor - minimum load velocity = 50% of full load velocity
      - 2-step Compressor - minimum load velocity = 67% of full load velocity
      - Typical AC motor VFD Driven Compressor - minimum load velocity = 50% of full load velocity (As compressor technology changes, this turndown can also change. Check the specific compressor capabilities to determine the % of full load velocity).
      - Digital Compressors are only checked at full load velocity. The minimum load velocity table does not apply to Digital Compressors. (see full load velocity limits above)

  - Heating Mode Conditions - Discharge Line Down
    - The discharge line uses the same line as the suction line, so using the suction line size selected based on the cooling mode conditions, check to make sure the discharge velocities and pressure loss values are acceptable.
    - Since discharge flow is down, oil return is not as big of a concern as when flow is up. Keep velocities above the minimum velocity for oil return. **Table 3**

Line Routing & Sizing - AHU below Condensing Unit (Heat Pump) Continued

- Hot Gas Reheat down -
  - Purge circuit must be installed for oil return.
  - Pitch horizontal sections 1 inch per 20 feet in the direction of flow (this will be pitched in the opposite direction as the suction line)
  - Keep velocity between 2000 fpm - 3500 fpm
- Hot Gas Bypass down -
  - Purge circuit must be installed for oil return.
  - Pitch horizontal sections 1 inch per 20 feet in the direction of flow (this will be pitched in the opposite direction as the suction line)
  - Keep velocity between 2000 fpm - 3500 fpm

15.7 Pitching the Lines Air-Cooled Systems

All gas lines should be pitched down in direction of flow. If lines are not properly sloped, oil will log in the lower portion of the pipe. In long lines, sufficient oil can be stored in the improperly sloped line to cause a loss of lubrication in the compressor causing compressor failure.

- Suction Line - Pitch in the direction of flow (about 1 inch per 20 feet of length) to maintain oil flow towards the compressor.
- Liquid Line - No pitch needed, but can be pitched to match either the suction line or the hot gas line.
- Hot Gas Reheat Line - Pitch horizontal sections 1 inch per 20 feet in the direction of flow (this will be pitched in the opposite direction as the suction line)
- Hot Gas Bypass Line - Pitch horizontal sections 1 inch per 20 feet in the direction of flow (this will be pitched in the opposite direction as the suction line)

15.8 Pitching the Lines Heat Pump Systems

- Suction/Discharge Line - Do not pitch the horizontal lines since they will be flowing in one direction in cooling mode and the opposite direction in heating mode. **The velocities in both the vertical and horizontal sections must be high enough to move oil in each mode of operation**.
- Liquid Line - No pitch needed
- Hot Gas Reheat Line - Pitch horizontal sections 1 inch per 20 feet in the direction of flow
- Hot Gas Bypass Line - Pitch horizontal sections 1 inch per 20 feet in the direction of flow

15.9 Insulating the Lines

- Liquid Line -
  - Cooling Only Systems - When the liquid line is routed through regions where temperature losses are expected, no insulation is required, as this may provide additional sub-cooling to the refrigerant. When routing the liquid line through high temperature areas, insulation of the line is appropriate to avoid loss of sub-cooling through heat gain.
  - Heat Pump Systems - When the liquid line is routed through regions where temperature losses are expected (all lines exposed to outside air conditions), insulate with a minimum 1 inch thick Armaflex insulation, as this will prevent capacity loss during heating mode of operation.
- Suction Line - The entire suction line should be insulated with a minimum 1 inch thick Armaflex insulation. This prevents condensation from forming on the line, and reduces potential loss in capacity associated with heat gain placing additional load on the system. This line should still be insulated in heat pump systems even though it acts as both a discharge and suction line.
- Hot Gas Reheat Line - Insulate the entire length of the hot gas line with a minimum 1 inch thick Armaflex insulation.
- Hot Gas Bypass Line - Insulate the entire length of the hot gas line with a minimum 1 inch thick Armaflex insulation.

15.10 Purge Circuit

The purge circuit is required on hot gas reheat or hot gas bypass lines. The purge circuit needs to be field furnished and installed at the **lowest** point of the line set. With this installation, oil drains into the drain leg of the hot gas reheat line. Oil accumulates until it reaches the level of the 1/8"OD capillary tubing. The combination of capillary action and the pressure difference between the hot gas reheat line (high pressure) and the suction line (low pressure) causes the oil to travel through the capillary tube into the suction line of the first circuit to return the oil to the compressor. The capillary tube connection to the suction line of the first circuit must be a minimum of 5 feet from the inlet to the compressor to allow the oil time to dissipate into the suction vapor and not slug the compressor with liquid oil.

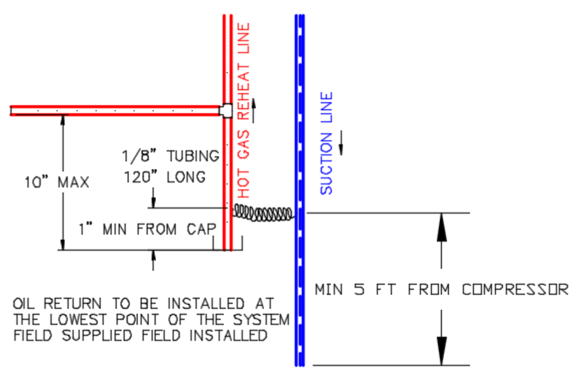


Figure 96 - Hot Gas Purge Circuit

The reason the purge circuit is needed is because both the hot gas reheat and the hot gas bypass are modulating. When only 10% (or less) of the refrigerant flow is going through the hot gas reheat or hot gas bypass lines, the velocities can be too low to keep the oil moving through the vertical lines. The purge circuit is a proven solution to get oil returned when the refrigerant flow is low through the hot gas lines.

15.11 Long Line Strategies

If the refrigerant line length is long, the pressure drop through the lines can get too high and reduce the efficiency of the system. One way to reduce the suction line pressure drop is to size the horizontal lines larger than the vertical lines, making sure to pitch them for oil return to the compressor. The vertical lines can then use a smaller diameter to maintain the velocity necessary to return oil up the riser. Transition to the smaller diameter at the horizontal pipe right before the trap, and then install an expander at the top elbow that transitions to horizontal again.

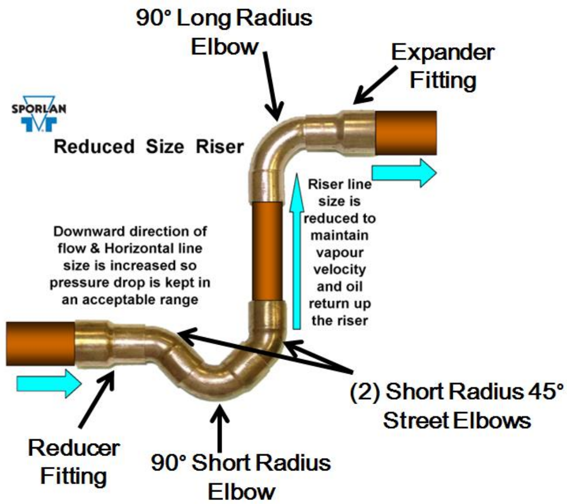
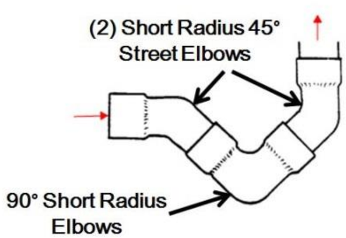


Figure 97 - Reduced Vertical Riser

Courtesy of Sporlan Division – Parker Hannifin Corporation

15.12 Trap Construction



The bottom traps can be constructed of two street elbows and one regular elbow to minimize the amount of oil that can get trapped in them. The mid-way oil traps can be constructed of four 90° long radius elbows.

16 Piping Diagrams

See the CF Installation and Operation Manual for the most up to date split system piping diagrams.

17 Addition of Oil

Excessive oil in the system reduces heat transfer and therefore capacity. Refrigerant is attracted to oil so it can migrate into the compressor and condense into a liquid.

It is critical that the refrigerant line piping is designed to maintain proper oil return to the compressors. Some systems may require oil to be added in addition to what is provided in the compressors.

Proper oil level should be observed under minimum load conditions. On units equipped with uneven tandem compressors, all oil is returned to the lead compressor in each tandem pair. When only the lead compressor is running, the oil level should be a minimum of ⅜ from the bottom of the sight glass. With both compressors running, the level in the lead compressor should drop to the bottom of the sight glass and the level in the second compressor should be a minimum of ⅜, from the bottom of its sight glass.

Systems with long line set or extra refrigerant charge may need oil added. And estimation of the amount of additional oil to add when the circuit charge exceeds 20 lbs of refrigerant is as follows:

- For compressors sized 24 MBtu/hr through 91 MBtu/hr (Copeland AE4-1331 R7 & AE4-1365 R4)
  - Add one fluid ounce of oil for every 5 lbs of refrigerant charge over the initial 20 lbs. Example 35 lbs of charge would require (35lb - 20lb) = 15lb / 5lb = 3 oz of oil added
  - If the system includes an accumulator, the manufacturer of the accumulator should be consulted for a pre-charge recommendation.
- For compressors sized 103 MBtu/hr through 485 MBtu/hr (Copeland AE4-1303 R15 & AE4-1388 R3)
  - Single compressor application - add 0.5 fluid ounce of oil per pound of refrigerant over initial 20 pounds. Example 35lbs of charge would require 15*0.5 = 7.5 oz of oil added
  - Tandem compressor application - add 0.7 fluid ounce of oil per pound of refrigerant over initial 20 pounds. Example 35lbs of charge would require 15*0.7 = 10.5 oz of oil added
  - If the compressor has an oil sight glass, the oil level should be checked during field commissioning and field servicing. The compressor oil level should be checked with the compressor off. The compressor oil level should not fall more than 1.5” below the center of the sight glass; however, the oil level should not rise above the 3/4 full level.
  - Tandem compressors should have their oil checked after 20 to 30 seconds of off time.
  - If the system includes an accumulator, the manufacturer of the accumulator should be consulted for a pre-charge recommendation.

18 Charging

Adjusting the charge of a system in the field must be based on determination of liquid sub-cooling and evaporator superheat. On a system with a thermostatic expansion valve, liquid sub-cooling is more representative of the charge than evaporator superheat but both measurements must be taken.

Unit being charged must be at or near full load conditions before adjusting the charge.

After adding or removing charge the system must be allowed to stabilize, typically 10-15 minutes, before making any other adjustments.

The type of unit and options determine the ranges for liquid sub-cooling and evaporator superheat. Refer to **Table 4** when determining the proper sub-cooling.

- Hot Gas Bypass Units - must have the hot gas bypass valve closed to get the proper charge.
- Hot Gas Reheat Units - must be charged with the hot gas valve closed while the unit is in cooling mode. After charging, unit should be operated in reheat (dehumidification) mode to check for correct operation.
- Heat Pump Units - should be charged in cooling mode with sub-cooling values from **Table 4**. After charging, unit should be operated in heating mode to check for correct operation. Charge may need to be adjusted for heating mode. If adjustments are made in the heating mode, cooling mode must be rerun to verify proper operation.
- Low Ambient Control (LAC) Flooded Condenser Units being charged when the ambient temperature is warm:
    - Once enough charge has been added to get the evaporator superheat and sub-cooling values to the correct setting, more charge must be added. Add approximately 80% of the receiver tank volume to the charge to help fill the receiver tank. The additional charge is required for the system when running in cold ambient conditions.
- Low Ambient Control (LAC) Flooded Condenser Units being charged when the ambient temperature is cold:
    - Once enough charge has been added to get the evaporator superheat and sub-cooling values to the correct setting, more charge may need to be added. If the ambient temperature is 0°F no more charge is required. If the ambient temperature is around 40°F add approximately 40% of the receiver tank volume.
    - The unit will have to be checked for proper operation once the ambient temperature is above 80°F.

18.1 Checking Liquid Sub-cooling

Measure the temperature of the liquid line as it leaves the condenser coil.

Read the gauge pressure at the liquid line close to the point where the temperature was taken. Use liquid line pressure as it will vary from discharge pressure due to condenser coil pressure drop.

Convert the pressure obtained to a saturated temperature using the appropriate refrigerant temperature-pressure chart (**Table 13** See table footnotes and correct for non-sea level altitudes).

Subtract the measured liquid line temperature from the saturated temperature (from **Table 13**) to determine the liquid sub-cooling.

Compare calculated sub-cooling to **Table 4** for the appropriate unit type and options.

18.2 Checking Evaporator Superheat

Measure the temperature of the suction line close to the compressor.

Read gauge pressure at the suction line close to the compressor.

Convert the pressure obtained to a saturated temperature using the appropriate refrigerant temperature-pressure chart (**Table 13** See table footnotes and correct for non-sea level altitudes).

Subtract the saturated temperature (from **Table 13**) from the measured suction line temperature to determine the evaporator superheat.

For refrigeration systems with tandem compressors, it is **critical** that the suction superheat set point on the TXV is set with one compressor running. The suction superheat should be 10-13°F with one compressor running. The suction superheat will increase with both compressors in a tandem running. Inadequate suction superheat can allow liquid refrigerant to return to the compressors which will wash the oil out of the compressor. Lack of oil lubrication will destroy a compressor. Liquid sub-cooling should be measured with both compressors in a refrigeration system running.

Compare calculated superheat to **Table 4** for the appropriate unit type and options.

Table 4 - Acceptable Liquid Sub-Cooling Values for Fin & Tube Condenser Coil

	Cooling Mode Liquid Sub-Cooling Values
Cooling Only Unit[4]	8-15°F
Cooling Only Unit with Hot Gas Reheat[1,4]	5-15°F
Heat Pump Unit[2,4]	2-4°F
Heat Pump Unit with Hot Gas Reheat[3,4]	2-6°F
Cooling Only Unit with LAC[4]	8-15°F
Cooling Only Unit with Hot Gas Reheat & LAC[4]	8-15°F

Notes:

1. Must be charged with the hot gas valve closed. After charging, unit should be operated in reheat (dehumidification) mode to check for correct operation.
2. The sub-cooling value in this table is for the unit running in cooling mode of operation. After charging, unit should be operated in heating mode to check for correct operation.
3. The sub-cooling value in this table is for the unit running in cooling mode of operation and the hot gas valve closed. After charging, unit should be operated in reheat (dehumidification) mode to check for correct operation and then in heating mode to check for correct operation.
4. Sub-cooling must be increased by 1°F per 10 feet of **vertical liquid line** rise for R-410A (AHU above CU). For example, a cooling only unit with hot gas reheat and a vertical liquid drop can charge to a sub-cooling value of 5-15°F, but a cooling only unit with hot gas reheat and a vertical liquid rise of 30 ft must charge to a sub-cooling value of at least 8-15°F.

Table 5 - Acceptable Liquid Sub-Cooling Values for Microchannel Condenser Coil

	Cooling Mode Liquid Sub-Cooling Values(°F)				
Ambient (°F)	Evaporator Coil Saturation Temperature (°F)				
	40	45	48	50	55
67	9 - 14	8 - 13	8 - 13	7 - 12	5 - 10
72	10 - 15	9 - 14	9 - 14	8 - 13	7 - 12
82	10 - 15	10 - 15	10 - 15	9 - 14	7 - 12
95	10 - 15	10 - 15	10 - 15	9 - 14	8 - 13
105	11 - 16	11 - 16	10 - 15	10 - 15	8 - 13
115	10 - 15	11 - 16	11 - 16	11 - 16	9 - 14

Notes:

1. Microchannel condenser coils are more sensitive to charge. The system must be running in cooling mode with compressor, supply airflow & condenser fan speed at full load. The sub-cooling value changes depending on the ambient temperature reading and the microchannel evaporator coil saturation temperature. To find the correct sub-cooling value, find the ambient temperature on the first column and follow that across to the SST (40-55°F).

Table 6 - Acceptable Suction Superheat Values

	Cooling Mode Superheat Values
Fin & Tube Condenser Coil	8-15°F
Microchannel Condenser Coil	9-15°F

Notes:

1. Superheat will increase with long suction line runs

18.3 Adjusting Sub-cooling and Superheat Temperatures

Overcharge - high sub-cooling; Evaporator is flooded - low superheat

Undercharge - low sub-cooling; Evaporator is starving - high superheat

The system is overcharged if the sub-cooling temperature is too high and the evaporator is fully loaded (low loads on the evaporator result in increased sub-cooling) and the evaporator superheat is within the temperature range as shown in **Table 4** (high superheat results in increased sub-cooling)

- Correct an overcharged system by reducing the amount of refrigerant in the system to lower the sub-cooling.

The system is undercharged if the superheat is too high and the sub-cooling is too low.

- Correct an undercharged system by adding refrigerant to the system to reduce superheat and raise sub-cooling.

If the sub-cooling is correct and the superheat is too high, the TXV may need adjustment to correct the superheat.

Flooded condenser charging - Sporlan 90-30-1

19 Liquid Line Receiver

The purpose of the liquid line receiver is to hold excess refrigerant volume that is not being circulated through the air conditioning system during lower capacity loads.

AAON units with Hot Gas Reheat, Heat Pump, and Low Ambient Flooded Condenser Controls include a receiver factory installed. Exception is the CB unit which includes a factory provided, but field installed receiver.

Receiver pump-down capacities are calculated at 90% of the receiver volume and 90°F for R-410A.

Receivers can be horizontal or vertical. The horizontal receiver must be installed with the connections on the top of the receiver (See **Figure 100**). If it is installed with the connections out to the side, it will not work correctly.

Figure 99 - Horizontal Receiver

Figure 98 - Horizontal Receiver Internals

Location: in the liquid line, after the condenser coil

Figure 100 - Horizontal Receiver Installed in Condensing Unit

Figure 101 - Vertical Receiver

When the evaporator and TXV are located above the receiver (AHU above CU), extra care must be given to make sure flash gas doesn't form in the liquid line preceding the TXV. If the vertical lift is excessive, it may be necessary to install a suction-liquid Heat Exchanger to ensure a solid liquid column entering the TXV.

How can a sub-cooled liquid come out of the receiver that has vapor and liquid stages? The answer is that the vapor and liquid (saturated state) is only at the liquid surface, but the refrigerant below the surface can exist at a sub-cooled state. The refrigerant is picked up at the bottom of the receiver where the refrigerant liquid is sub-cooled.

20 Suction Line Accumulator

Figure 102 - Suction Line Accumulator

Figure 103 - Suction Line Accumulator Internal

The function of the suction line accumulator is to keep refrigerant liquid from entering the compressor. It should be located in the suction line near the compressor. All AAON condensing units with heat pump will include a factory installed suction line accumulator.

A suction line accumulator is recommended on systems with a large refrigerant charge, or on any system where liquid floodback is likely to occur.

Figure 104 - Suction Line Accumulator Oil Return Port

Location - in the suction line, near the compressor

Figure 105 - Suction Line Accumulators & Liquid Line Receivers Installed in a Heat Pump Condensing Unit

What should the AAON stance be on when to use suction line accumulators in field installation for AC systems?

Question: When do I need a suction line accumulator in my split system piping?

Answer: There is no absolute rule for when to use a suction line accumulator; following are application guidelines. However, the application guidelines below are AAON's position.

Suction line accumulators are safety devices used to prevent liquid refrigerant from returning to the compressor. Some "sub-cooling" or "heat exchanger" type accumulators have a section of the liquid line run through the accumulator, which helps evaporate any liquid refrigerant in the suction line section and also provides additional sub-cooling to the liquid line. The suction line accumulator also serves as a temporary liquid refrigerant storage chamber for heat pump systems. It protects the compressors when the system switches between heating mode and cooling modes of operation.

The accumulator must be sized to handle enough volume to hold the maximum expected liquid in the suction line and a large enough diameter to separate liquid from suction gas. For systems that operate with a thermostatic expansion valves this is approximately 50% of the circuit charge. For systems that use a fixed orifice this is approximately 100% of the circuit charge.

Systems that may require a suction accumulator include:

1. Systems with wide load variations or rapid load changes.
2. Systems with many steps of capacity control or a complicated capacity-control system.
3. Heat pump systems, where the suction line accumulator also serves as a temporary liquid refrigerant storage chamber. Suction line accumulators are factory provided with AAON heat pump split systems
4. Systems with hot gas bypass or hot gas reheat.
5. Systems which have required a compressor replacement because of liquid slugging.
6. Split systems where the sub-cooling required exceeds 8°F may need a heat exchanger type suction line accumulator. This may occur when the liquid line has a vertical rise of 25ft or higher (condensing unit below air handling unit). The ECat32 split system selection software and Engineering Toolkit line sizing program will include the sub-cooling required.

20.1 Suction Line Accumulator - Heat Exchanger Type

When the liquid line needs additional sub-cooling a heat exchanger type suction line accumulator can be used. Notice the liquid line refrigerant flow is routed through the accumulator. The suction temperatures are much lower (ie 45°F) than the liquid line temperature (ie 110°F), so the refrigerant in the liquid line can be sub-cooled by the suction gas in the accumulator.

When the air handling unit is located above the condensing unit, especially if there is a receiver in the condensing unit, it may be necessary to use a heat exchanger to ensure subcooling in the liquid line entering the TXV.

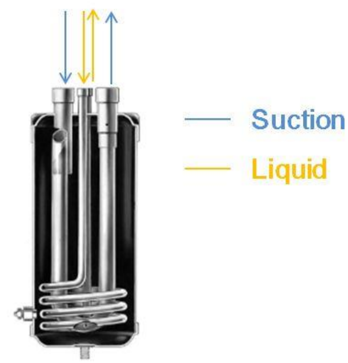


Figure 106 - Heat Exchanger Type Suction Line Accumulator

See Refrigeration Research website or Westermeyer Industries website for more information.

21 Heat Exchanger

The heat exchanger does the same thing as the heat exchanger inside a suction line accumulator. So if it is a heat pump unit with a standard suction line accumulator and the AHU is above the CU, a heat exchanger could be used to gain sub-cooling in the liquid line to avoid flash gas entering the TXV.

Figure 107 - Heat Exchanger

On systems with excessive vertical lift, install the heat exchanger near the receiver before the vertical lift occurs. Vapor in the liquid line considerably increases the pressure drop through the liquid line so locating the heat exchanger close to the receiver can minimize pressure drop.

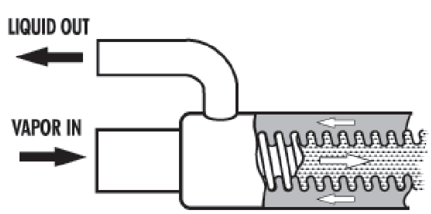


Figure 108 - Heat Exchanger Schematic

The benefits of the heat exchanger are that it sub-cools liquid refrigerant, improves refrigerating capacity, evaporates any liquid in the suction line before it enter the compressor.

See Packless website for more information.

22 Sight Glass

A moisture and liquid indicating sight glass may be field installed anywhere in the liquid line to indicate the occurrence of premature flashing and moisture in the line. Bubbles in the glass view port indicate flash gas in the liquid refrigerant. The green dot in the center is a moisture indicating chemical that changes colors based on ppm of moisture it reads.

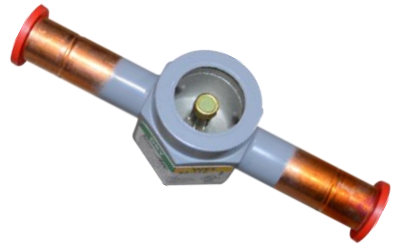


Figure 109 - Sight Glass

yellow = wet

green = dry

Figure 110 - Sight Glass with Cap

When it is first installed, allow the system to operate for 12 hours to allow the system to reach equilibrium before deciding if the filter-drier needs to be replaced. If the indicator turns yellow, a new filter-drier should be installed. The indicator element does not need to be changed, as it will turn back to green when the system is dry. The only exception is if the indicator is in contact with a lot of water such as a broken tube in a water cooled condenser; if this happens, the indicator paper will need to be replaced. If the system is circulating an excessive amount of oil, the indicator paper may appear brown, but once the oil gets mixed back into the refrigerant, the paper will return to its proper color.

The sight glass should not be used to determine if the system is properly charged. **Use temperature and pressure measurements to determine liquid sub-cooling, not the sight glass.**

See Sporlan Bulletins 70-10 & SD-21 for more information.

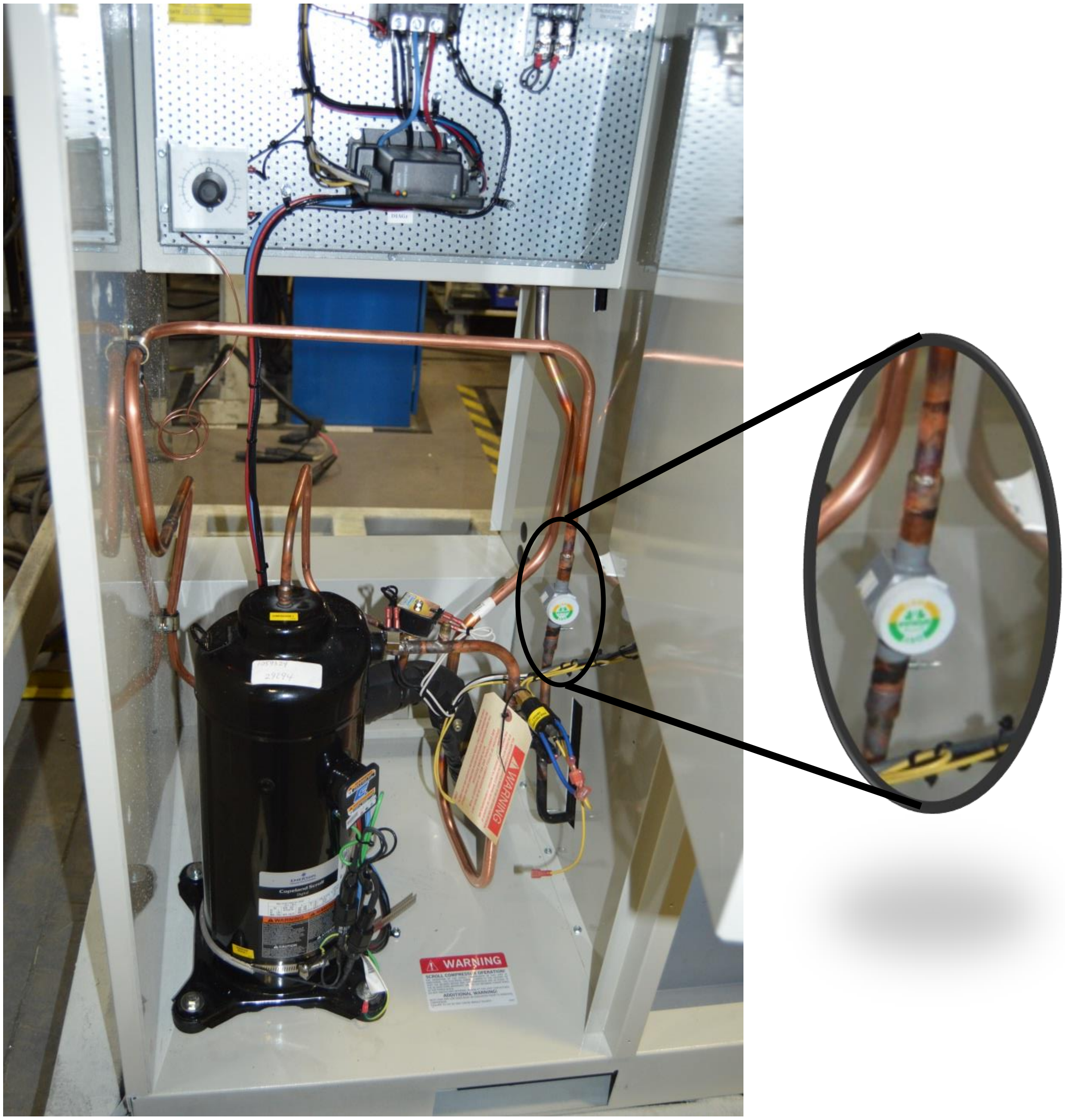


Figure 111 - Sight Glass Installed

23 Filter-Drier

Liquid line filter-driers are installed on all AAON refrigerant circuits. Liquid line filter-driers remove moisture, dirt, scale, and metallic particles from the refrigerant. Moisture can interact with POE lubricant or refrigerant and filter-drierform acid. The acid could cause copper plating. The moisture could also form into ice crystals causing several possible issues.

A temperature differential greater than 3°F across the filter-drier indicates that it could be clogged. Refrigerant flow capacity in Sporlan's catalog is rated at a pressure drop of 1 psi.

<u>Location - in the liquid line, after the receiver and before any valves</u>

23.1 Liquid Line Filter-Drier - Directional

Notice the arrow on the filter-drier below is only in one direction. The second picture below is a liquid line filter-drier that was cut in half to show the inside parts. The desiccant core is normally a single piece.

Figure 113 - Filter-Drier

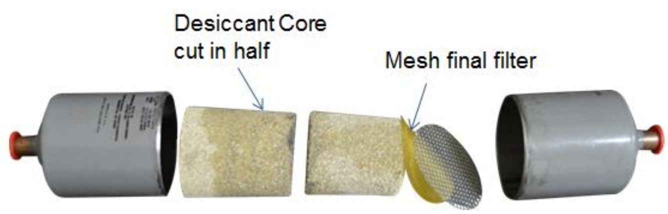


Figure 112 - Filter-Drier Internals

See [Sporlan Bulletin 40-10]() for more information.

23.2 Liquid Line Filter-Drier - Reversible Heat Pump

Notice the arrows on the filter-drier below go both directions, which can be used in a heat pump system where the liquid line changes directions depending on which mode is in operation; heating mode or cooling mode.

Figure 114 - Filter Drier Bi-Directional

Notice there are two of the metal pieces with a large hole in the middle and several small holes on the outside. The little metal flaps are similar to check valves, and they work so that when the flow is in one direction, they are forced open and when the flow is the opposite direction, they are forced closed. When the refrigerant flows in one direction, it comes in through the small outside holes and exits the other side through the large hole in the middle. When the refrigerant flows the opposite direction, it comes in through the large hole in the middle and exits through the small holes on the outside. See the HPC-100 Series schematic for the two different flow paths.

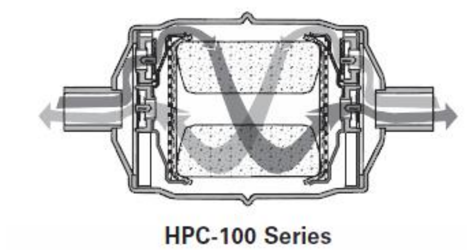


Figure 115 - Sporlan HPC-100 Series Schematic

Courtesy of Sporlan Division – Parker Hannifin Corporation

Figure 116 - Filter Drier Bi-Directional Internals

The picture above is a liquid line filter-drier that was cut in half to show the inside parts. The desiccant core is normally a single piece.

See Sporlan Bulletin SD-111 for more information.

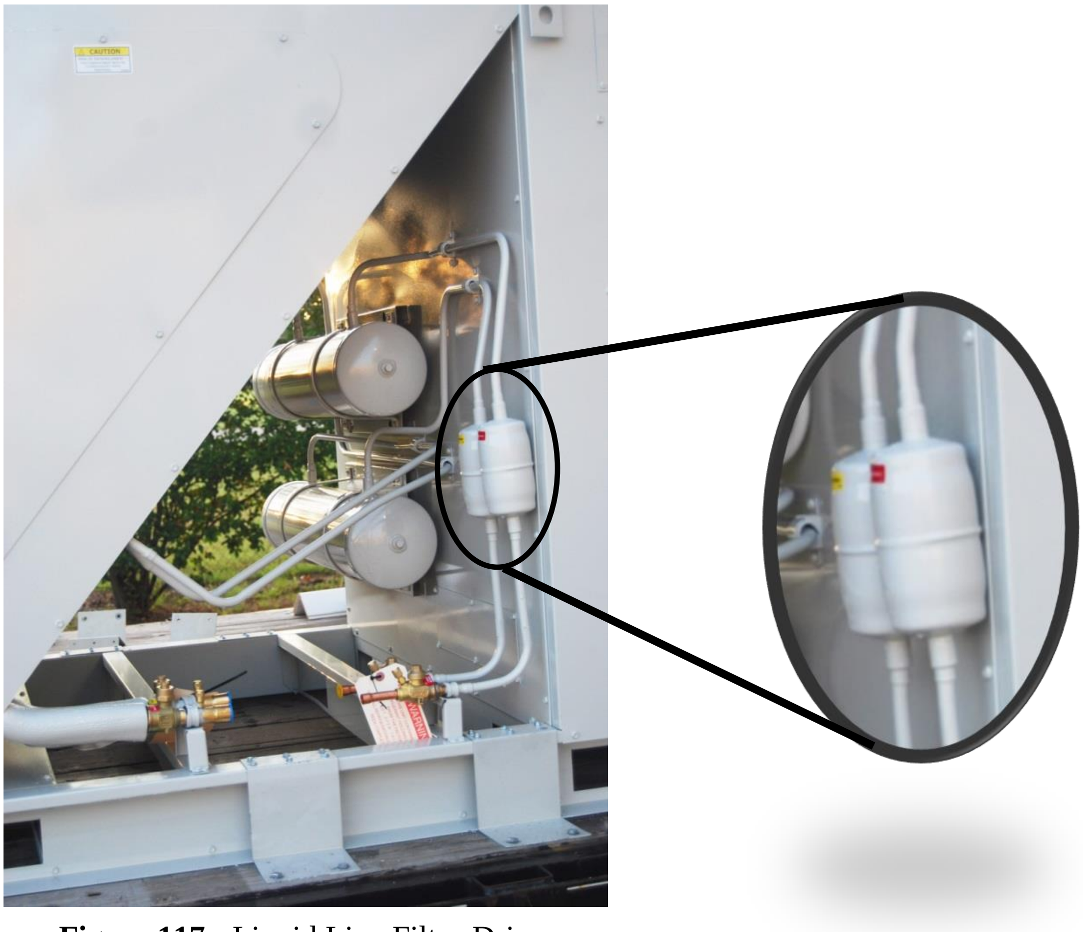


Figure 117 - Liquid Line Filter-Driers
Installed in Condensing Unit

23.3 Liquid Line Filter-Drier - Replaceable Core

Replaceable core liquid line filter-driers are a factory installed on some AAON equipment. They are used in place of the standard filter-drier so that changing the filter-drier does not require refrigerant evacuation and brazing. This design is especially advantageous on larger systems with more refrigerant charge.

See [Sporlan SD-214] for installation and servicing instructions.

Figure 118 - Filter Drier Replaceable Core

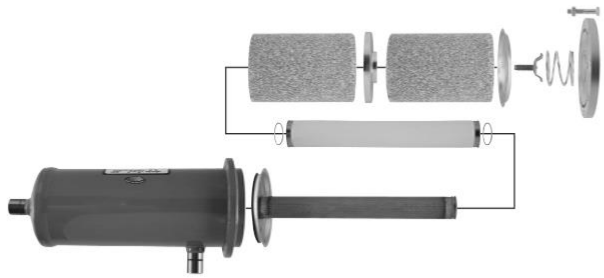


Figure 119 - Filter Drier Replaceable Core Internals

Courtesy of Sporlan Division – Parker Hannifin Corporation

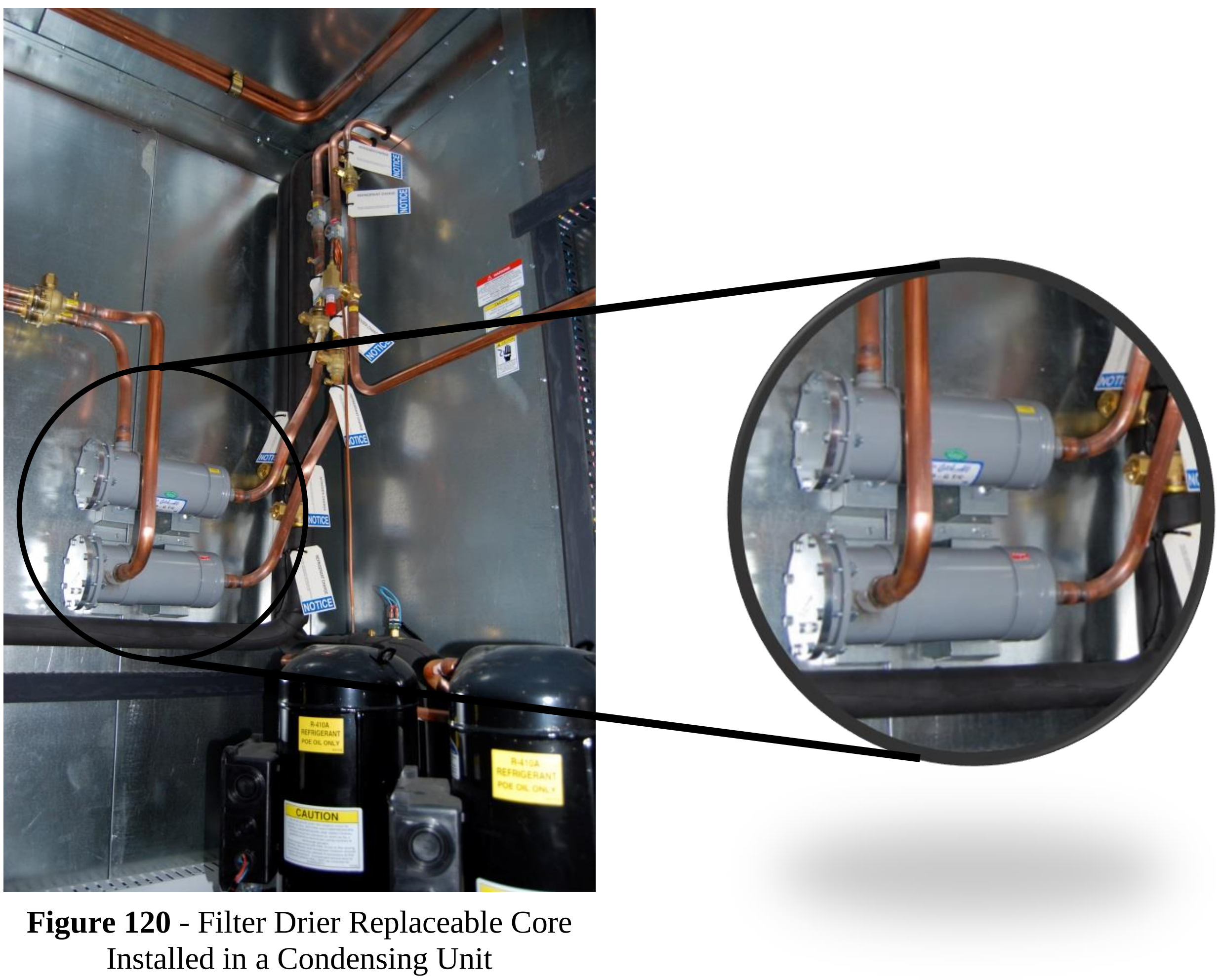


Figure 120 - Filter Drier Replaceable Core Installed in a Condensing Unit

23.4 Liquid Line Filter-Drier - HH Style for Wax Removal

Small amounts of wax are often a problem on low temperature systems, so Sporlan has an HH style filter-drier that includes activated charcoal. This would be a very specific field installed application if needed.

See Sporlan Form 40-109 for more information.

23.5 Suction Line Filter-Drier

A suction line filter-drier is typically used for clean-up after a compressor burn-out.

Many suggest that the suction line filter-drier be removed after 3 days. This is to eliminate the additional pressure drop caused by the suction line filter-drier. Sporlan's clean up procedure allows the suction line filter-drier to be left in the system to eliminate additional service calls.

See Sporlan Bulletin 40-10 for a step by step clean-up procedure after a compressor burn-out.

Location - in the suction line, before the compressor

23.6 Suction Line Filter

AAON factory installs pleated replaceable core suction line filters on some equipment to be removed one month after startup. The suction filter-drier helps to protect the compressors from contaminants during testing and startup. Removing the suction line filter improves the efficiency and capacity of the unit.

Figure 121 - Suction Line Filter

24 Oil Separator

Oil separators are typically utilized on low or ultra-low temperature refrigeration systems and on large air conditioning applications with long refrigerant lines. Oil separators remove oil from the compressor discharge gas, temporarily store the oil, and then return it to the compressor. An oil separator gives more time before the compressor runs out of oil, but if there are regular intervals of 100% load, an oil separator can help to bridge long operating periods of low load.

Location - in the discharge line, close to the compressor's crankcase

25 Automatic Pump-down System

An automatic pump-down system can be field installed to help protect the system against refrigerant migration problems. In winter conditions when the compressors are not running, refrigerant migrates to the coldest part of the refrigerant system. The coldest part is typically the condenser coil and compressors since they are outside.

Since the compressors are not meant to handle a liquid refrigerant, refrigerant migration is a big problem for the compressors. Automatic pump-down systems pump most of the refrigerant into the condenser coil and receiver before shutting down the compressors. A normally closed (N/C) solenoid valve is field installed in the liquid line and is wired to the room thermostat. When the call for cooling is met, the solenoid valve de-energizes and closes while the compressors are still running. The compressors will run until the suction pressure reaches the low pressure safety switch. This pumps most of the refrigerant through the TXV, evaporator coil, and compressor to be stored in the condenser coil and receiver. When the thermostat calls for cooling again, the solenoid valve will energize (open) and allow the liquid refrigerant back into the TXV to be evaporated in the evaporator coil before entering the compressor.

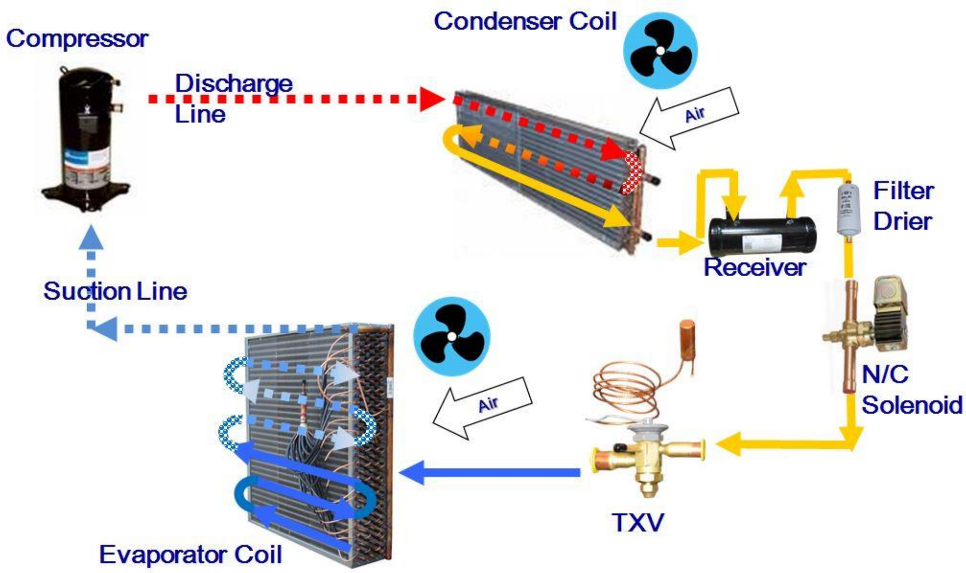


Figure 122 - Pump down System

Pump-down cycle will not protect against flooding during operation. It also adds a risk of short cycling unless a separate discharge line check valve is installed. A simple and cost effective solution instead of pump-down cycle is simply closing a liquid line solenoid valve when the compressor is off. This method is suggested by Copeland in AE4-1388 R3.

Solenoid valves are uni−directional devices. Never install a pump-down cycle on a heat pump system. Copeland does not recommend pump-down cycle for scroll compressors sizes 1.5-15 ton.
(See AE4-1331 R7, AE4-1303 R15, AE4-1365 R4)

26 Crankcase Heater

Many AAON compressors are provided with factory installed crankcase heaters to prevent refrigerant from condensing in the compressor and causing damage or wear. The unit must have continuous power 24 hours prior to startup. This will give the crankcase heater time to clear any liquid accumulation out of the compressor before it is required to run. Crankcase heaters keep the oil in the compressor warmer than the coldest part of the system to slow down refrigerant migration. There is a risk of overheating the oil, so the correct sized crankcase heater must be used. It is possible at temperatures lower than 0°F, that the crankcase heater may not be able to prevent all migration. Crankcase heaters also do not protect against liquid floodback. Some AAON packaged units do not require crankcase heaters on the compressors since they have a small refrigerant charge.

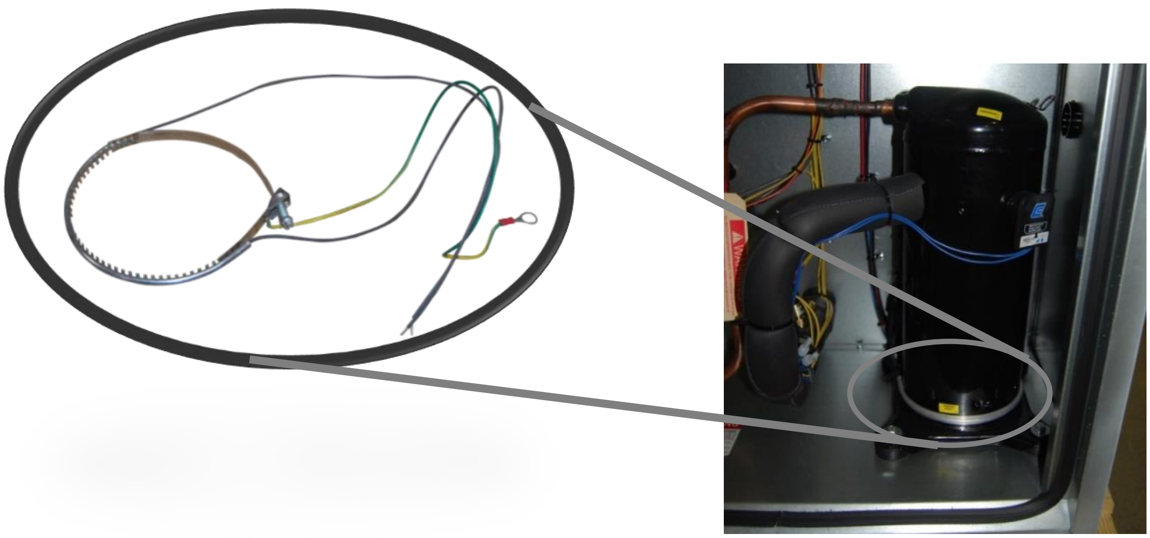


Figure 123 - Crankcase Heater Installed on Compressor

27 Valves

27.1 Liquid Line Solenoid Valve

Solenoid valves can be normally-closed (N/C) type, which means the valve opens when the coil is energized, or normally-open (N/O) type, which means the valve closes when the coil is energized. They either allow a fluid to flow through or they stop a fluid from flowing (on/off type control). In very basic terms, when an electrical signal is sent to a N/C solenoid valve, the valve will open and allow fluid to flow through, and when the electric signal stops, the N/C solenoid valve will close.

The solenoid coil can have several different voltage and cycle ratings (24V, 120V, 208-240V / 50-60 hz) with either AC or DC.

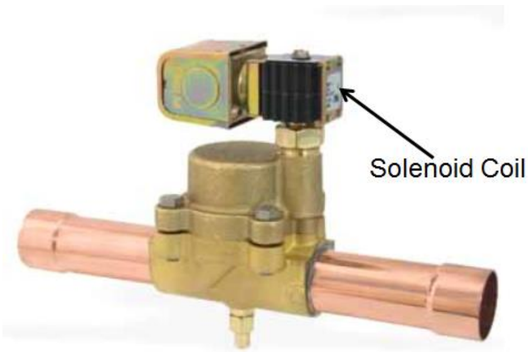


Figure 124 - Liquid Line Solenoid Valve

When to use?

Liquid line solenoid valves are used in Automatic Pump-down Systems. A simple and cost effective solution instead of pump-down cycle is simply closing a liquid line solenoid valve when the compressor is off.

Do not use a liquid line solenoid valve on a heat pump system, the accumulator is already in place to protect the compressor and the solenoid valve is uni-directional so it will not work in a bi-directional system.

27.2 Check Valve

A check valve will only allow flow in one direction. If flow comes from the opposite direction, it gets blocked by the internal workings of the check valve. They are used to prevent refrigerant from flowing in the wrong direction.

Figure 125 - Check Valve

Figure 126 - Check Valve Internals

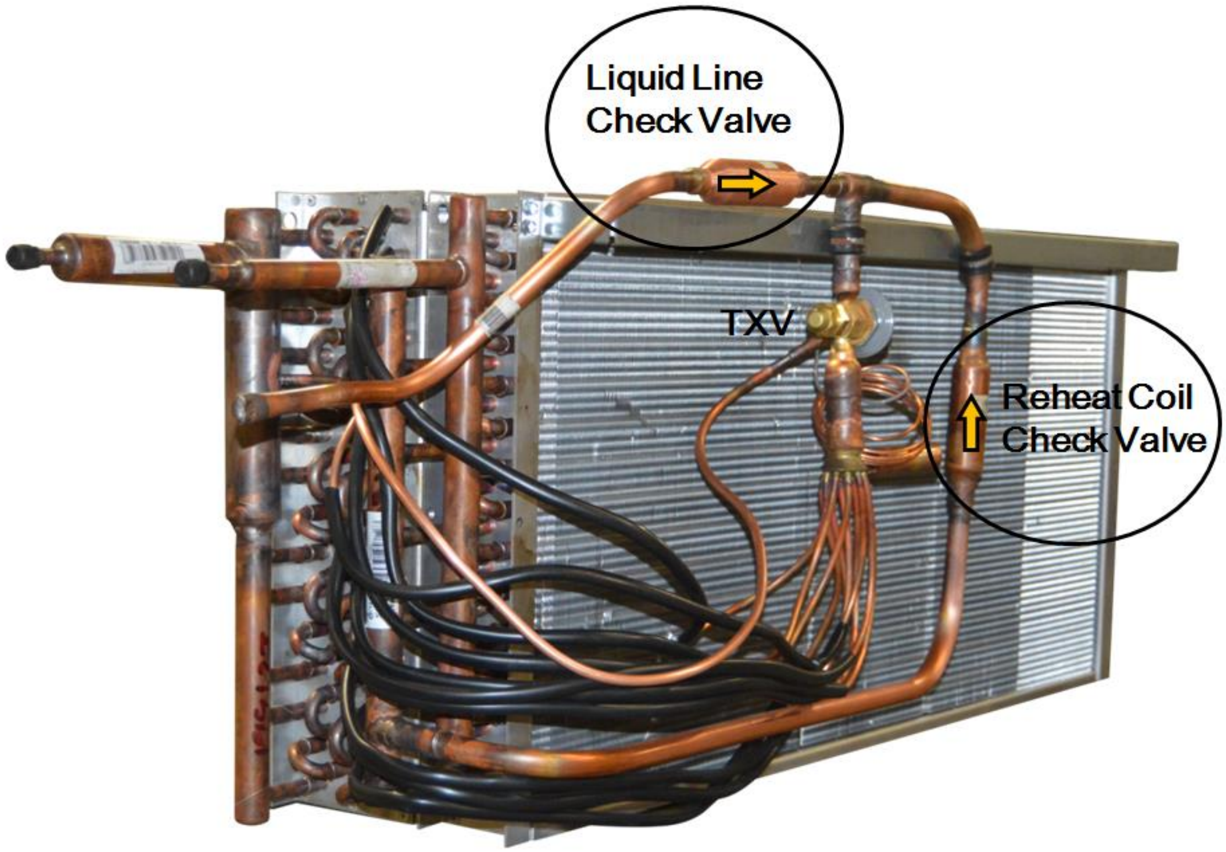


Figure 127 - Check Valves Installed for Modulating Hot Gas Reheat

27.3 Ball Valves

Ball valves are either fully open or fully closed. In the picture below they are used as shut-off valves to the liquid and suction line connections. The cap on the top can be twisted off and then the valve can be manually turned using the stem on the top.

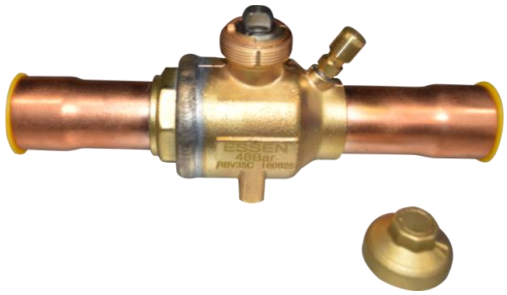


Figure 128 - Ball Valve

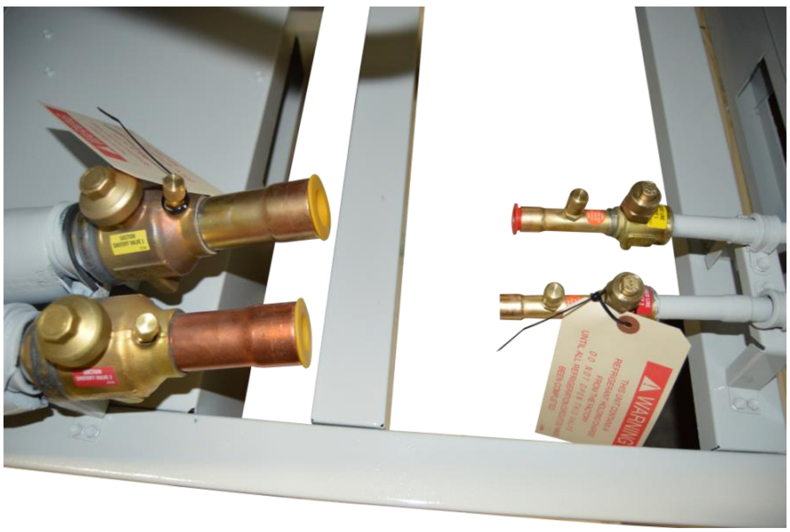


Figure 129 - Ball Valves Installed on a Condensing Unit as Shut-off Valves

27.4 Isolation Valves

Another application of the ball valve is compressor or filter-drier isolation valves. Ball valves installed around a component such as a compressor allows a service person to shut both valves and minimize the refrigerant evacuated in order to remove that component.

Figure 131 - Compressor Isolation Valve

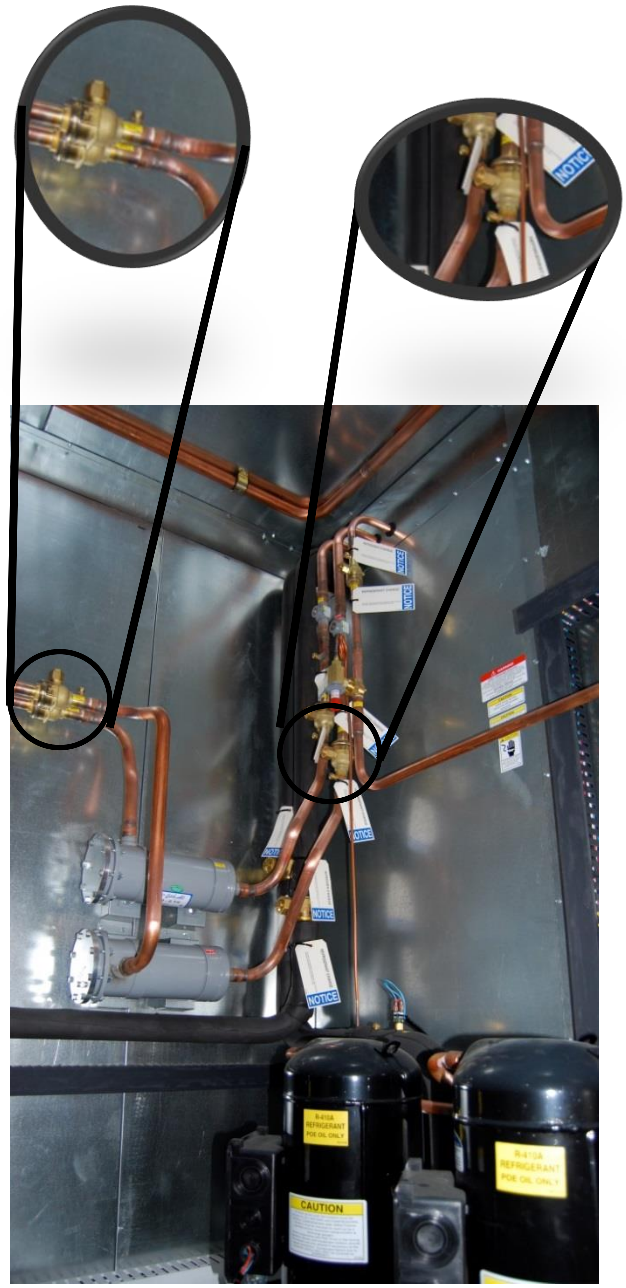


Figure 130 - Filter Drier Isolation Valve

27.5 Backseat (King) Valves

Backseat valves are also known as King valves. They are also either fully opened or fully closed valves. They include a service port for charging or reading a pressure.

Figure 132 - Backseat Valve

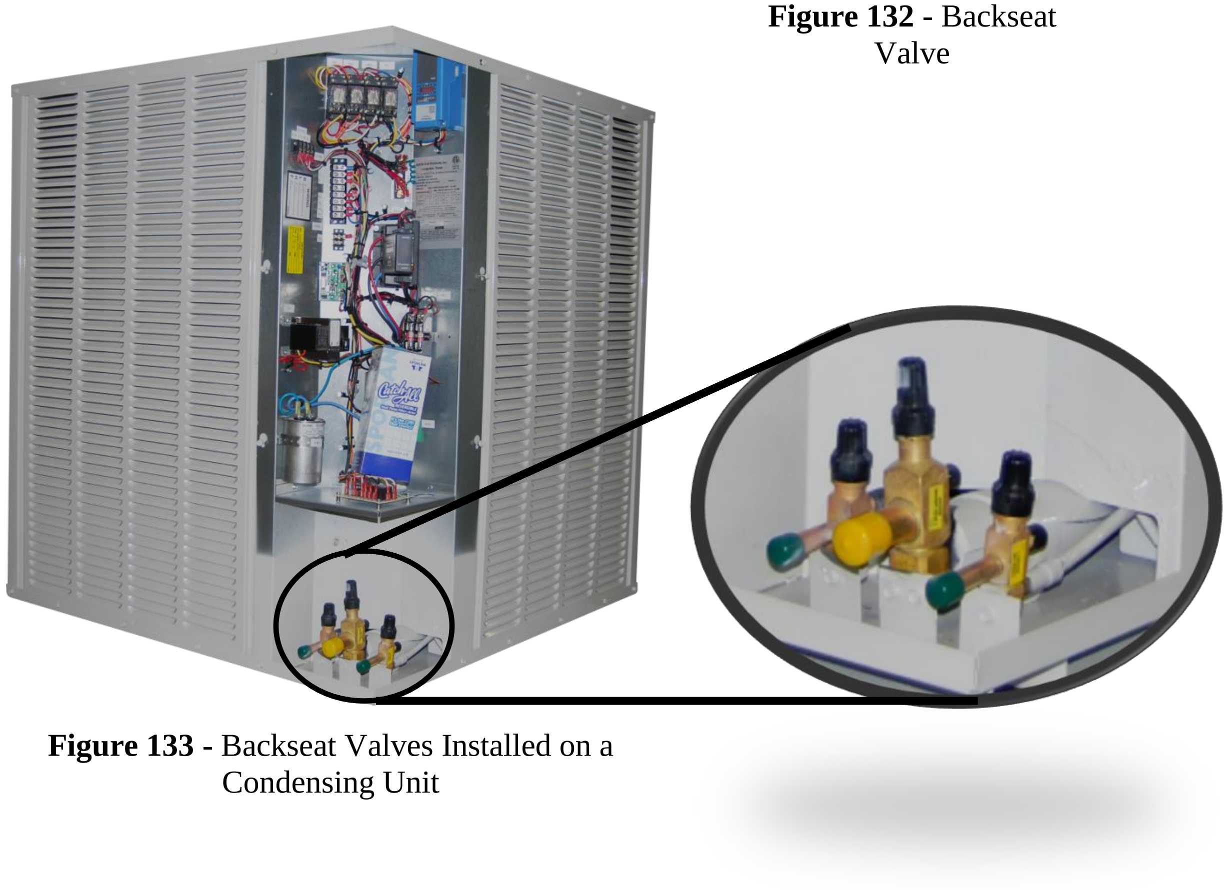


Figure 133 - Backseat Valves Installed on a Condensing Unit

27.6 Schrader Valve

Schrader valves can be used for charging the refrigerant system or for checking refrigerant pressures. They are brazed into the refrigerant lines. They are also installed and then high and low pressure switches or pressure transducers can be screwed onto the Schrader valve.

Figure 134 - Schrader Valve

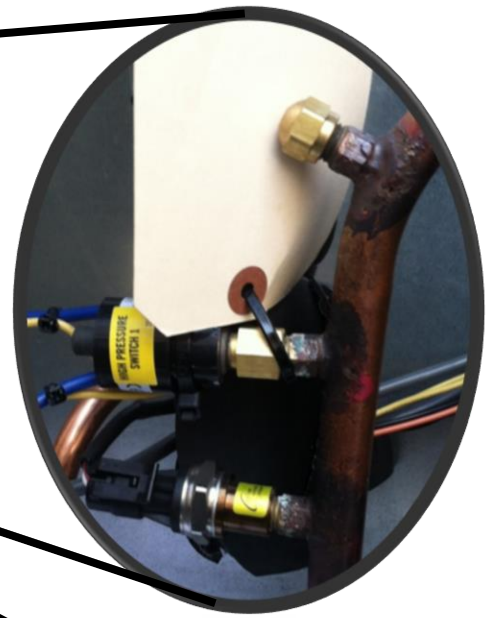


Figure 135 - Schrader Valves Installed on Discharge and Suction Lines

28 Low Ambient Head Pressure Control

If cooling operation is necessary at outside temperatures under 55°F, head pressure control must be included in the system. The methods of low ambient head pressure control are adjustable fan cycling, VFD controlled condenser fan, ECM controlled condenser fan, and flooded condenser. The first three methods allow cooling operation down to 35°F ambient, while the flooded condenser allows cooling operation down to 0°F ambient.

28.1 Adjustable Fan Cycling Head Pressure Control (Down to 35°F Ambient)

Adjustable fan cycling control is the simplest and least efficient method of head pressure control. Basically this switch simply turns the condenser fan on and off based on the discharge (head) pressure. The head pressure control setpoint (100-470 psi) and pressure differential (35-200 psi) is field adjustable. The recommended cut-in pressure is 425psi, and the recommended differential is 155psi, making the cut-out pressure 270psi. Using these pressure values, the fan cycling switch will turn the fans off when the pressure drops below 270psi. This allows the compressors to continue operating and cooling the space without the unit tripping out on low discharge pressure. While the condenser fan is off, the discharge pressure builds up and when it reaches 425psi, the fan cycling switch will turn the fans on. Minimum allowable ambient temperature for cooling operation is 35°F.

Figure 136 - Adjustable Fan Cycling Switch

Figure 137 - Adjustable Fan Cycling Switch - Split Systems

Figure 138 - Adjustable Fan Cycling Installed in a Split System Condensing Unit

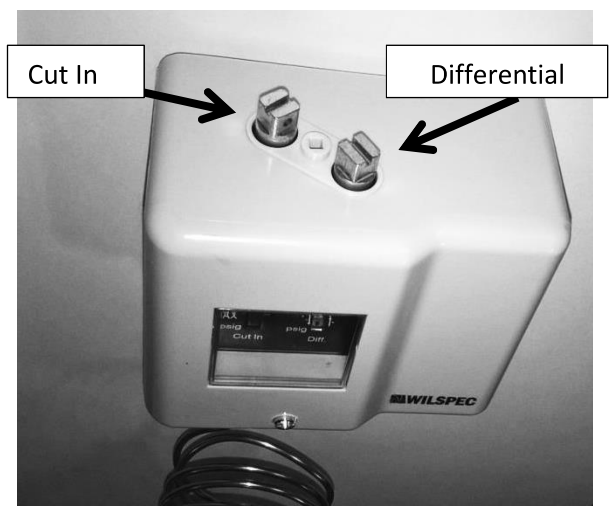
Rooftop units use a different brand of fan cycling switch, but both switches operate in the same way. The procedure for setting the rooftop fan cycling switch is on the next page. In general, this procedure can be used on both switches, but the location of the cut-in and differential knobs are different.

To adjust the fan cycle switch use a flathead screwdriver.

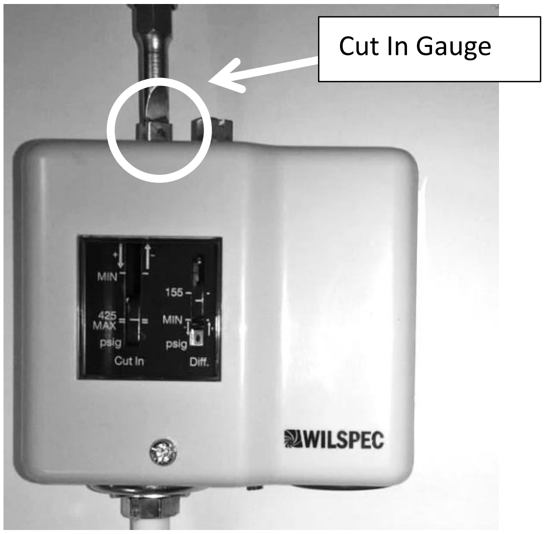
Recommended Settings

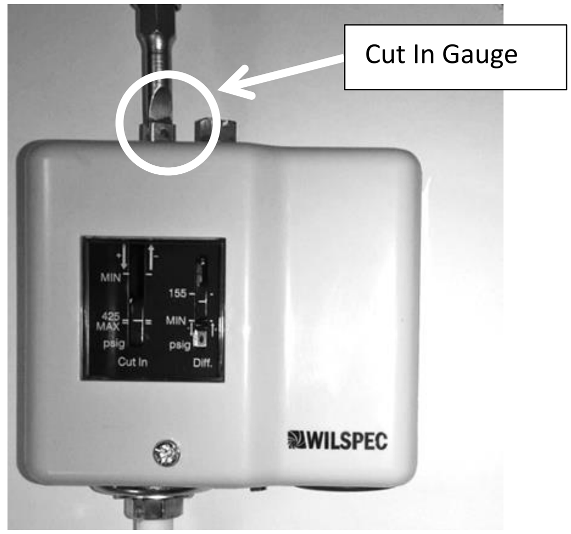
The switch will come factory set to cut-in at 425psi (+/– 5psi) and a differential of 155psi (or open at 270psi (+/– 5psi)).

Settings for CUT IN and DIFFERENTIAL PRESSURE are indicated with two slider gauges.



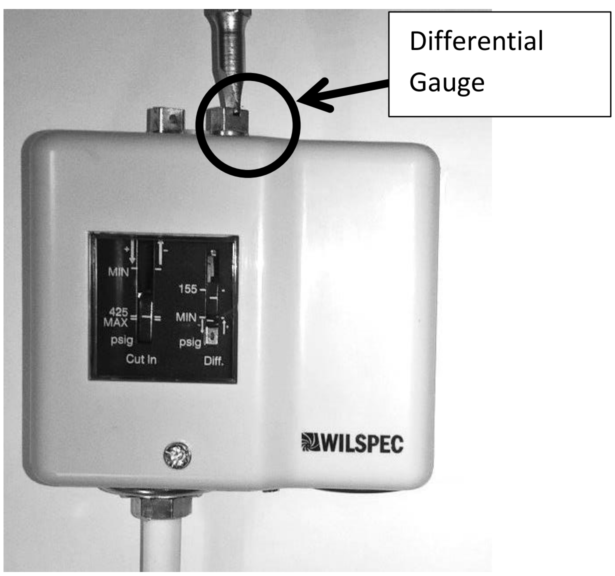
Each adjustment screw sits above the setting that it controls.

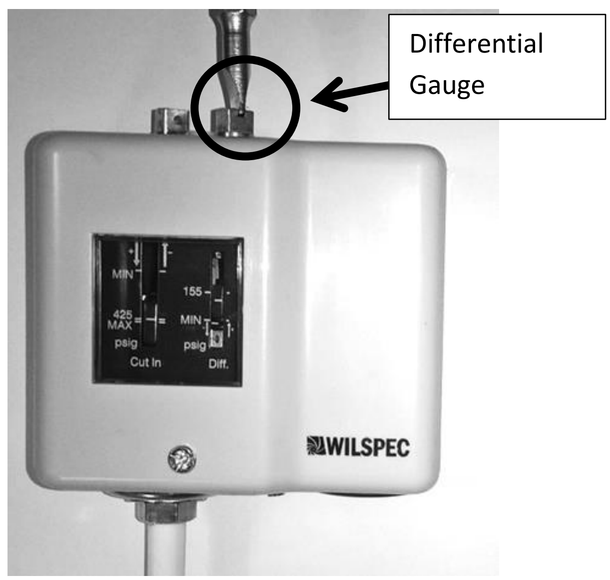




To lower the pressure set point for the **CUT IN** gauge, turn the adjustable screw clockwise.

To raise the pressure set point for the **CUT IN** gauge, turn the adjustable screw counter clockwise.





To raise the pressure set point for the **DIFFERENTIAL** Gauge, turn the adjustable screw clockwise.

To lower the pressure set point for the **DIFFERENTIAL** Gauge, turn the adjustable screw counter clockwise.

NOTE: The pressure values on the gauge should be verified with gauges on the refrigerant line. The gauge scale is for illustration purposes only.

28.2 VFD Controlled Condenser Fan Head Pressure Control (Down to 35°F Ambient)

VFD controlled condenser fan head pressure control is more efficient than adjustable fan cycling head pressure control. Instead of turning the fans on and off to control the discharge (head) pressure, the VFDs modulate the fan speed to maintain a discharge pressure. Factory provided and programmed VFDs receive inputs from the discharge pressure transducers on each refrigerant circuit and vary the fan speed to maintain the desired discharge pressure. Standard discharge pressure setpoint is 340 psi for standard air-cooled systems and 400 psi for modulating hot gas reheat air-cooled systems. Minimum allowable ambient temperature for cooling operation is 35°F.

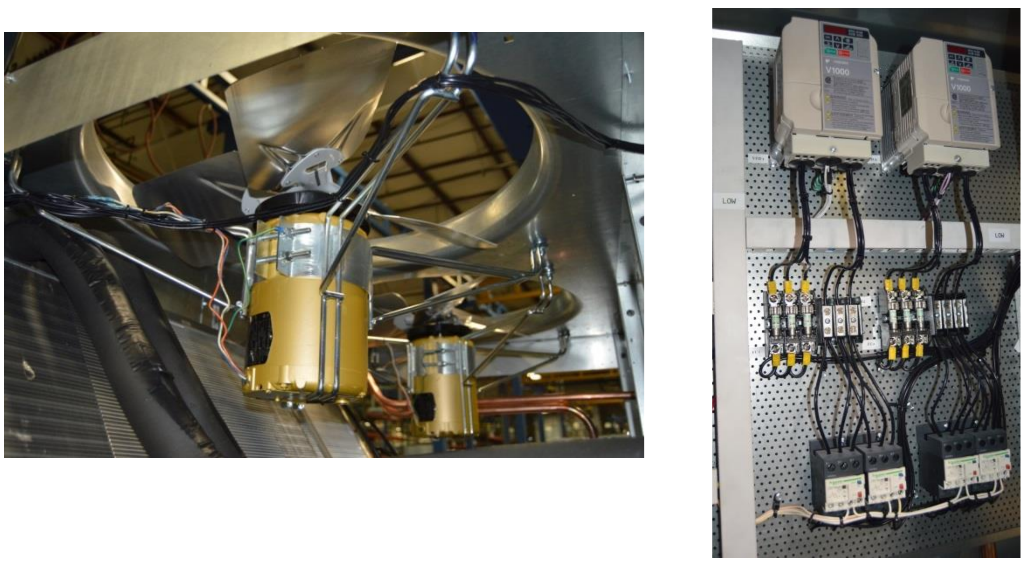


Figure 139 - VFD Controlled Condenser Fan Head Pressure Control

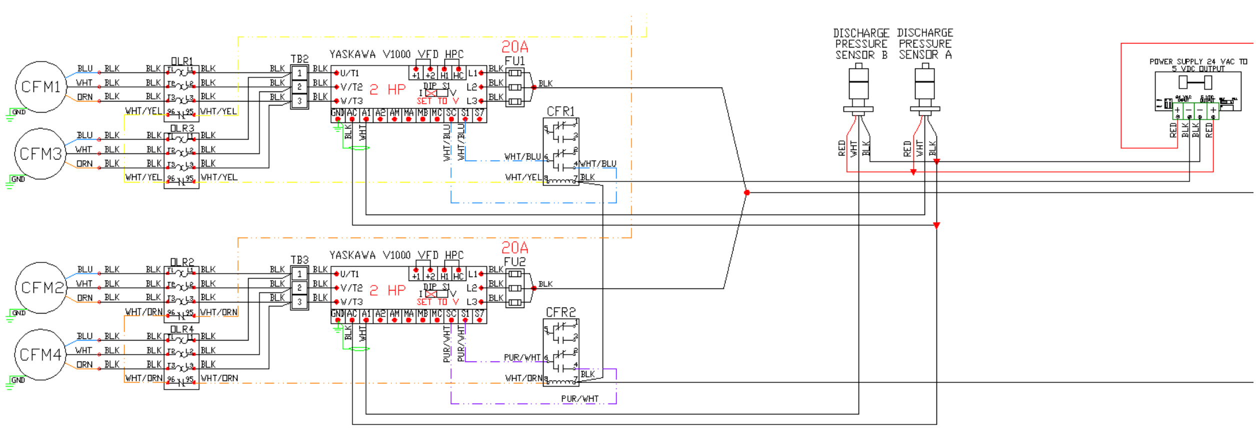


Figure 140 - VFD controlled condenser fans head pressure with terminal block control wiring

VFD controlled condenser fans sometimes include a power supply output board. The discharge pressure sensor sends a 0.5 -4.50 VDC signal, corresponding to the discharge pressure in the line, to the VFD. The VFD uses the value to determine whether to speed up or slow down the fan.

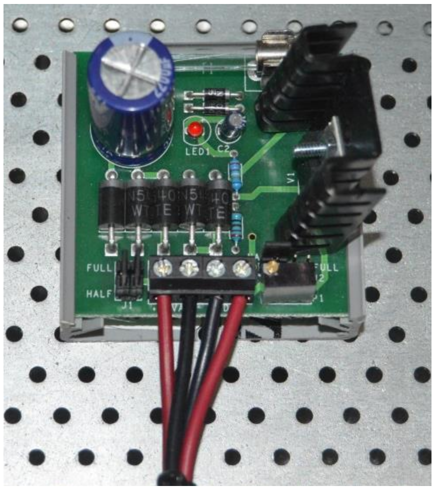


Figure 141 - Power Supply 24VAC to 5VDC

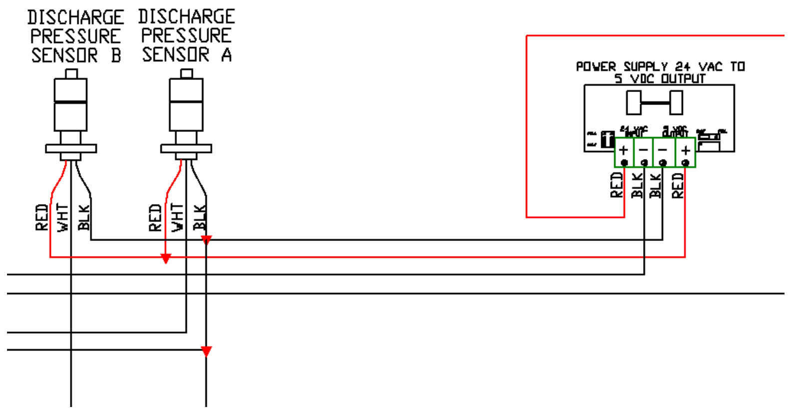


Figure 142 - Power Supply Board Wiring

Table 7 - Discharge Pressure Transducer PSI to VDC

Courtesy of Micro Control Systems

PSI to VDC

PSI	SI (vdc)	PSI	SI (vdc)	PSI	SI (vdc)	PSI	SI (vdc)	PSI	SI (vdc)	PSI	SI (vdc)	PSI	SI (vdc)
0	0.50	100	1.10	200	1.70	300	2.30	400	2.90	500	3.50	600	4.10
10	0.56	110	1.16	210	1.76	310	2.36	410	2.96	510	3.56	610	4.16
20	0.62	120	1.22	220	1.82	320	2.42	420	3.02	520	3.62	620	4.22
30	0.68	130	1.28	230	1.88	330	2.48	430	3.08	530	3.68	630	4.28
40	0.74	140	1.34	240	1.94	340	2.54	440	3.14	540	3.74	640	4.34
50	0.80	150	1.40	250	2.00	350	2.60	450	3.20	550	3.80	650	4.40
60	0.86	160	1.46	260	2.06	360	2.66	460	3.26	560	3.86	660	4.46
70	0.92	170	1.52	270	2.12	370	2.72	470	3.32	570	3.92	667	4.50
80	0.98	180	1.58	280	2.18	380	2.78	480	3.38	580	3.98		
90	1.04	190	1.64	290	2.24	390	2.84	490	3.44	590	4.04		

28.3 ECM Condenser Fan Head Pressure Control (Down to 35°F Ambient)

Electrically Commutated Motor (ECM) condenser fan head pressure control operates in a similar manner to the VFD controlled condenser fan head pressure control. Both are more efficient than adjustable fan cycling head pressure control. The EC motors either speed up or slow down to adjust air flow in order to maintain the head pressure setpoint. Standard discharge pressure setpoint is 340 psi for standard air-cooled systems and 400 psi for modulating hot gas reheat air-cooled systems. Option includes ECMs, condenser head pressure controller and discharge pressure transducers. Minimum allowable ambient temperature for cooling operation is 35°F.

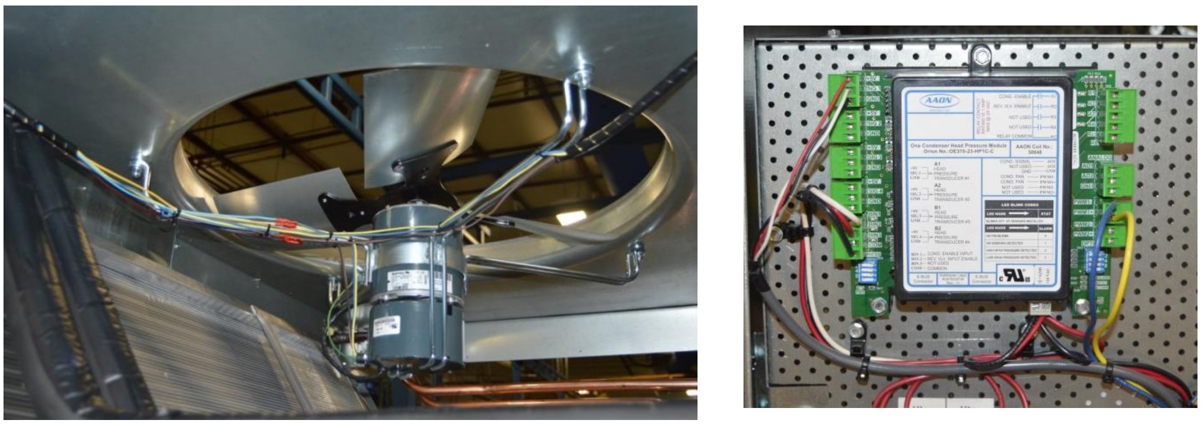


Figure 143 - ECM controlled condenser fan head pressure control

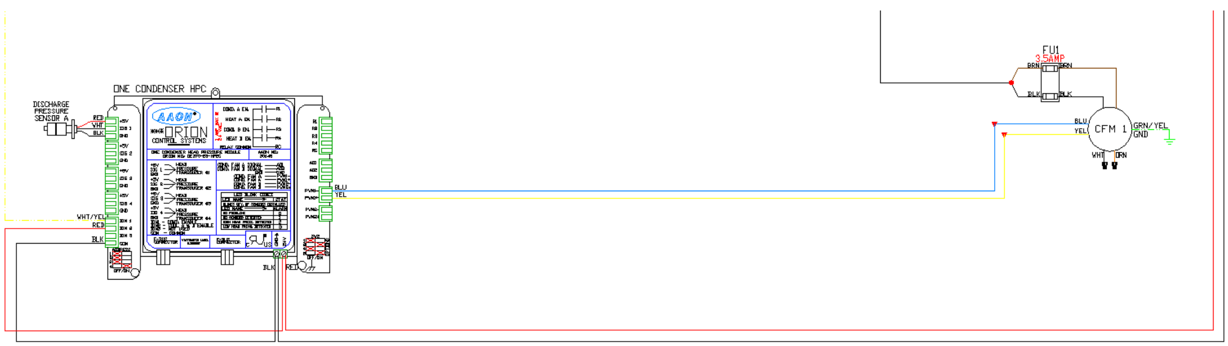


Figure 144 - ECM controlled condenser fan head pressure control wiring with terminal block

Sometimes an ECM motor controller must be used to convert a 0-10 VDC signal to a PWM signal when a PWM (Pulse Width Modulating) motor is being used. See **Table 8** for the input & output signals.

Table 8 - ECM Motor Controller Inputs & Outputs

Speed	Input	Output
%	VDC	VDC
0	0.0	2.0
20	2.0	6.5
50	5.0	13.0
75	7.5	18.0
100	10.0	23.0

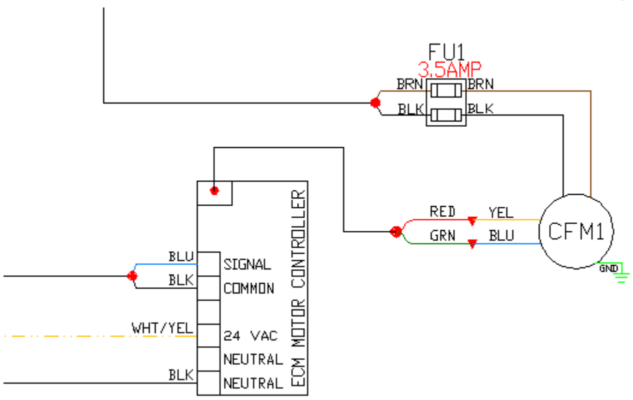


Figure 145 - ECM Motor Controller Wiring

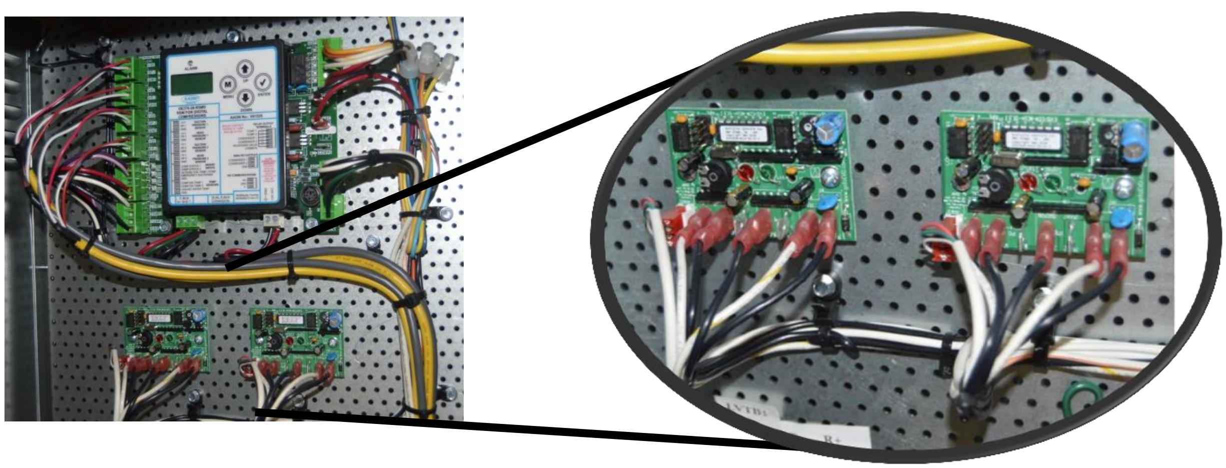


Figure 146 - ECM Motor Controllers

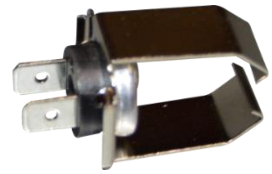
28.4 Flooded Condenser (Down to 0°F Ambient)

This low ambient head pressure control method allows cooling operation down to 0°F ambient temperature. This method is described in detail in section Split System -Flooded Condenser Low Ambient Controls (LAC).

29 Switches & Safeties

29.1 Discharge Line Safeties & Switches

Figure 148 - Digital Discharge Thermistor



Discharge Thermistor is wired to the Copeland Scroll Digital Compressor Diagnostics. A discharge temperature above setpoint will cause a trip. Five high discharge temperature trips within 4 hours will cause a compressor lockout. (Flash Code 2)

Figure 147 - Digital Compressor Circuit

High Pressure Switch is a safety feature on all compressor circuits. If the discharge pressure exceeds the setpoint, the switch will shut the compressor off. The high pressure cut-out switch is set to 650 psig for R-410A.

Figure 149 - High Pressure Switch

Figure 150 - Discharge Pressure Transducer

Discharge Pressure Transducers are included on each compressor circuit that includes head pressure control.

29.2 Suction Line Safeties & Switches

Figure 151 - Digital Compressor Circuit

Low Pressure Switch is a safety feature on all compressor circuits. If the suction pressure drops below the setpoint, the switch will shut the compressor off. The low pressure cut-out switch is set to 60 psig for R-410A straight AC units and 30 psig for R-410A heat pump units.

Figure 152 - Low Pressure Switch

Suction Pressure Transducer is used by the solenoid valve bypass to modulate compressor capacity during dehumidification. During cooling operation, the transducer can serve as an electronic safety and controller output.

Figure 153 - Suction Pressure Transducer

29.3 Other Safeties & Switches

Figure 154 - Freeze-Stat

Freeze-Stat is used for defrost control of the coil in heat pump mode. It is mounted on the return bend of the coil and if it signals a temperature below setpoint, the unit will switch from heat pump mode to cooling mode to defrost the coil.

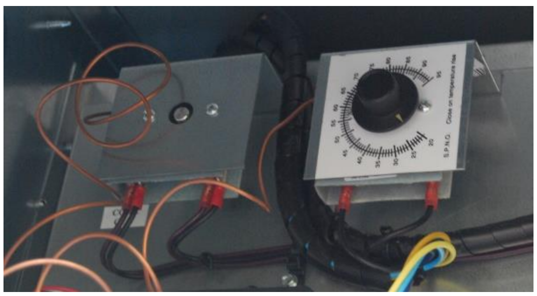


Figure 156 - Compressor Lockouts

Figure 156 - Compressor Lockout Ambient Sensors

Compressor Lockouts are included with heat pump units without factory provided controls. The cooling mode uses a non-adjustable compressor lockout set to 55°F and the heating mode uses an adjustable compressor lockout with a range from 20°F to 95°F. The ambient temperature sensors are installed on the outside of the unit as shown in the picture on the right above. On units with factory provided controls, the lockouts can be adjusted in the controls configuration.

Figure 157 - Guaranteed Off Timer

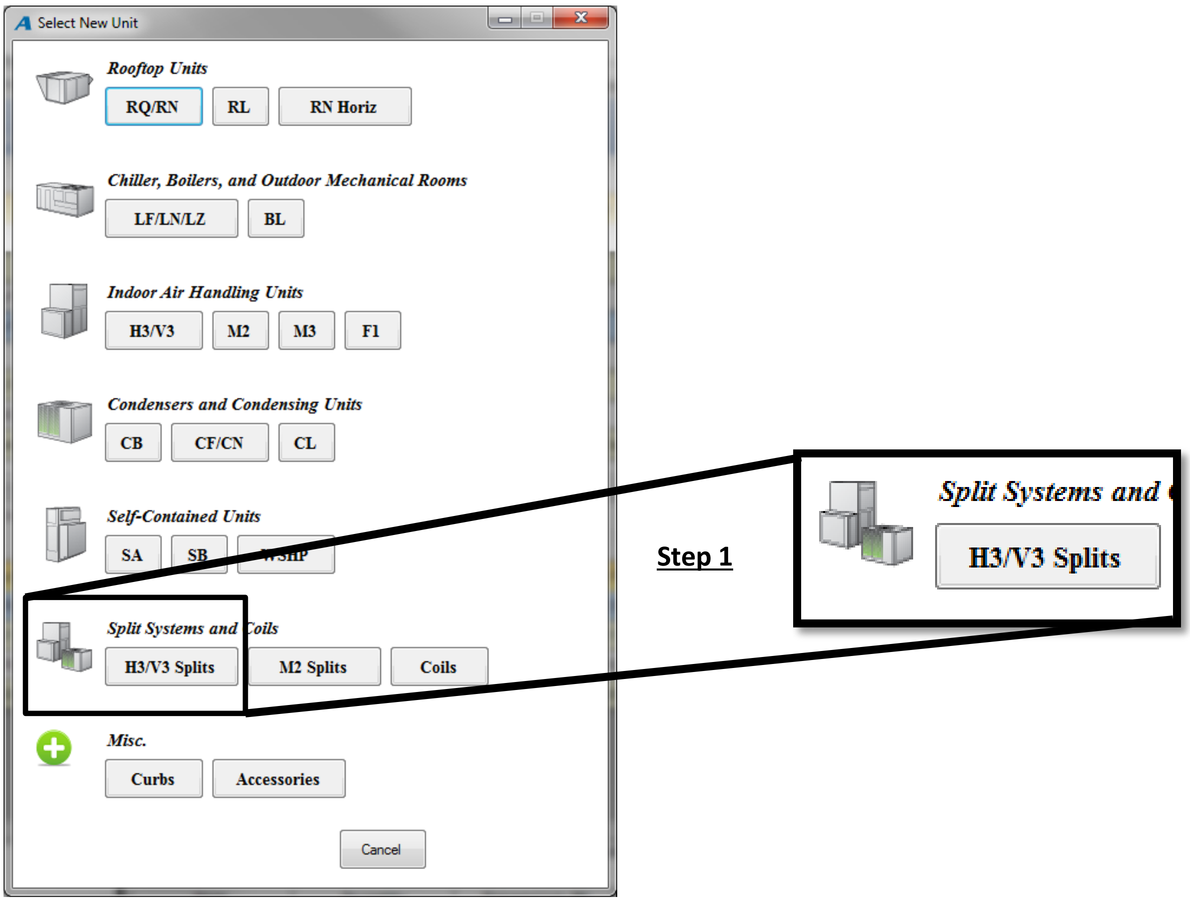
5 Minute Off Compressor Time Delay Relay protects the compressor by guaranteeing a 5 minute compressor off time to avoid compressor short cycling.

30 ECat Split System Line Sizing

AAON split system selection software can make line sizing selections easy for most jobs. Here are the steps to follow:

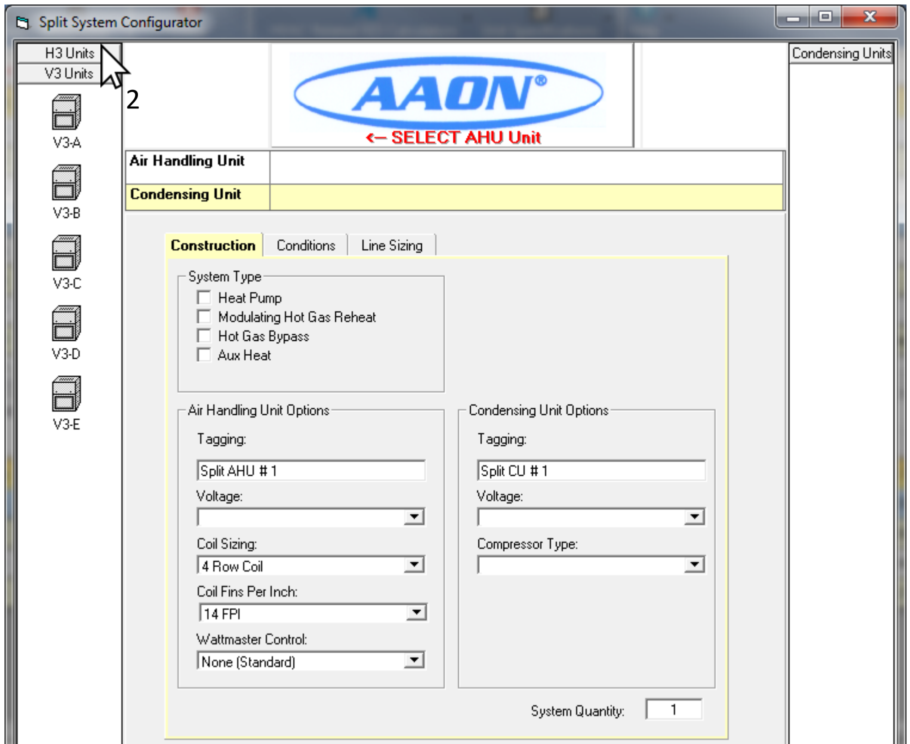
<u>Step 1:</u> After selecting 

from the ECat screen, make sure to select either an H3/V3 Split or M2 Split.

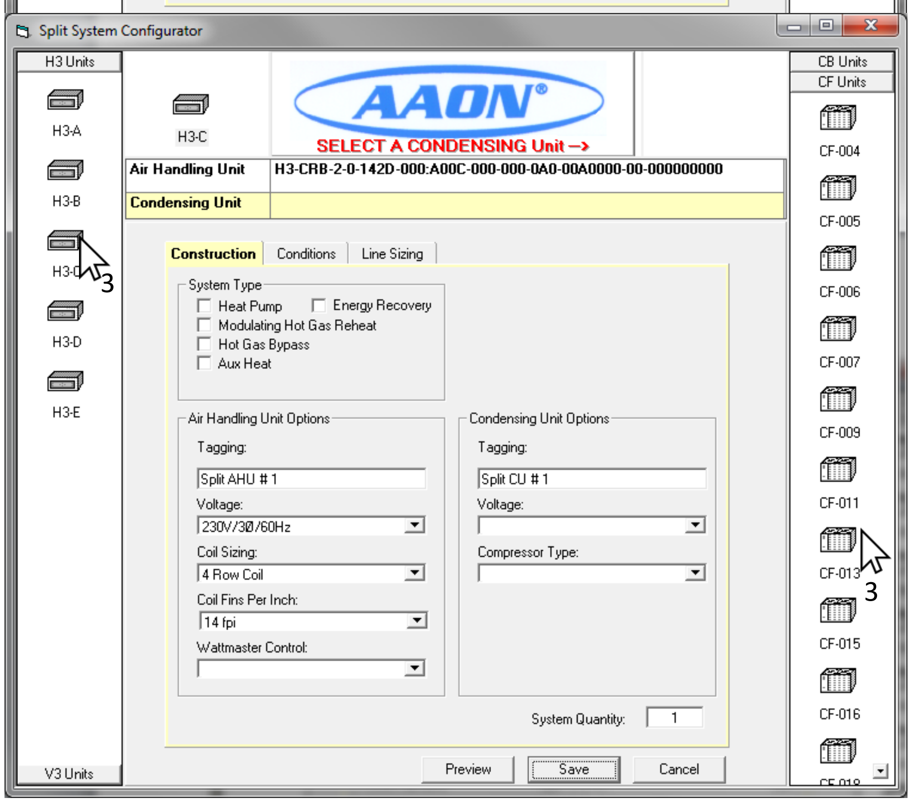


Note: the split system software currently includes H3/V3/M2/CF/CB units. If the selection includes a unit type other than these, go to the <u>EES Toolkit Line Sizing</u> section of this document.

Step 2: This page will be the next page. On the left side of the screen, click on the air handling unit size. Click on the H3 Units tab to see the H3 Unit options.

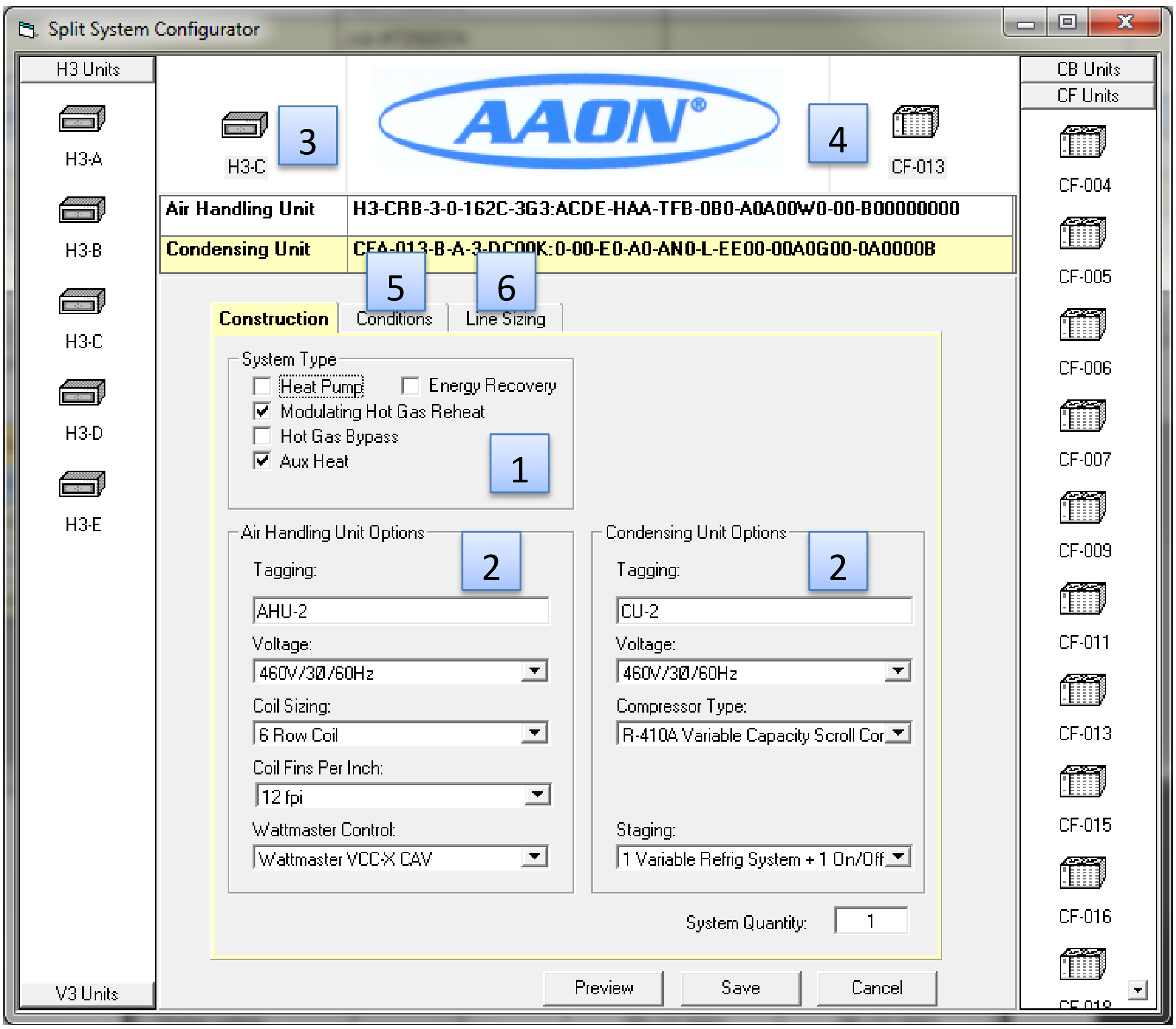


Step 3: After the AHU has been selected, the options for condensing units become available on the right side of the screen. Select the tonnage condensing unit. Click on the CB Units tab to see the CB Unit options.

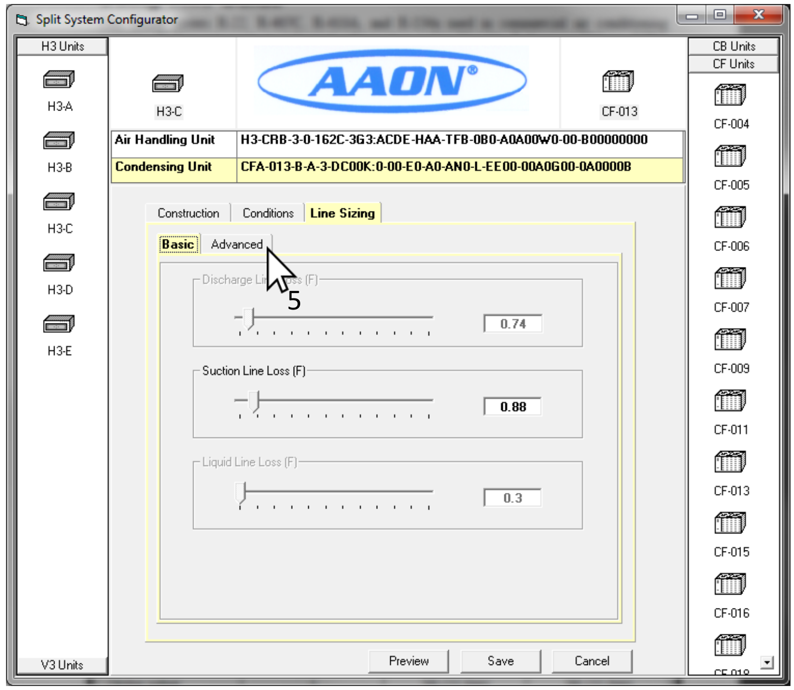


Step 4: After both the AHU & CU have been selected

1. Check the System Type boxes that apply to the split system (heat pump, modulating hot gas reheat, hot gas bypass, auxilary heat, energy recovery).
2. Select the drop-down options for the AHU & CU
3. Double-click the H3-C icon or the configuration string to select additional featuresfor the AHU.
4. Double-click the CF-013 icon or the configuration string to select additional feature for the CU.
5. Click on the conditions tab and update the conditions to the schedule.
6. Click on the line sizing tab



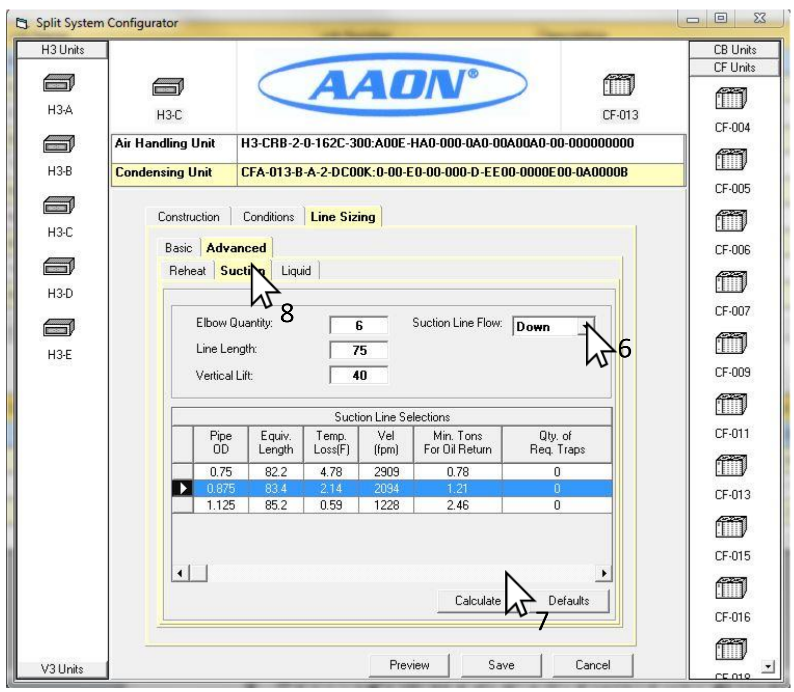
Step 5: The first screen that comes up in the line sizing tab is the basic line sizing tool. Skip past this page and click on the Advanced Tab.



The basic tab does not help size lines. It simply shows the de-rate in capacity for a specific suction line loss. The input is suction line loss and it can be changed by moving the slider.

Step 6: The advanced tab allows inputs for the line length, elbow quantity, vertical lift, and the direction of suction flow. All inputs can be entered in the suction line tab.

Suction Line Flow drop down arrow is very important.

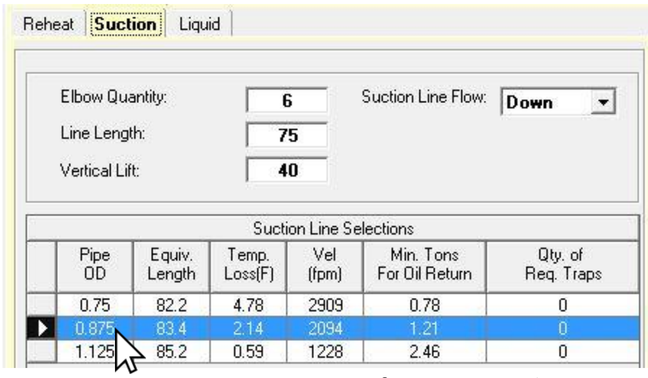


Suction Line Flow = Up when the condensing unit is physically above the air handling unit. The suction flow will be up and the liquid, hot gas reheat, and hot gas bypass flow will be down.

Suction Line Flow = Down when the condensing unit is physically below the air handling unit. The suction flow will be down and the liquid, hot gas reheat, and hot gas bypass flow will be up.

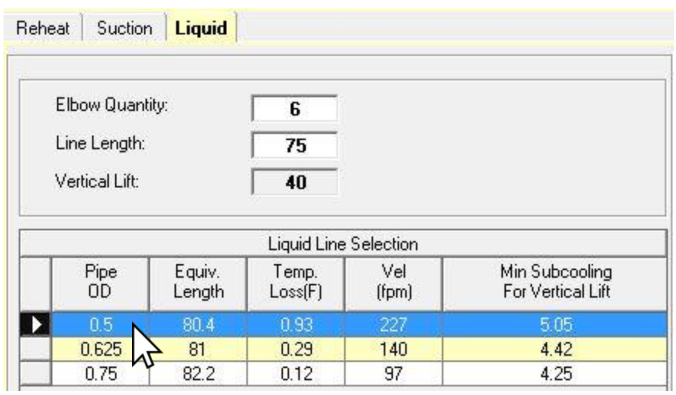
Step 7: Click on Calculate. This shows the line sizes that are within the velocity and temperature loss limits.

Step 8: Click on the different tabs (suction, liquid, reheat, bypass) to select each line individually.



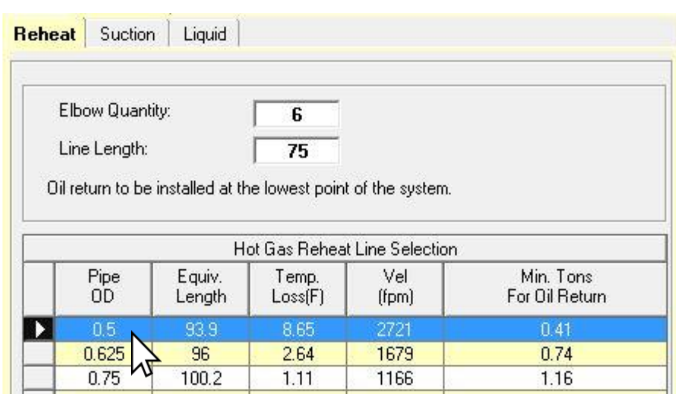
Step 9: Suction Line - Click on the desired Pipe OD to make a selection

Refrigerant line sizing is a decision based on weighing the pros and cons of a line size. Choose the 0.875" OD suction line for this split system. The higher suction temperature line loss of 4.78°F on the 0.75"OD option gave a gross capacity of 128MBH and the 0.875"OD suction line with a temp line loss of 2.14°F gave a gross capacity of 133MBH (Notice ECat accounts for the line loss in the capacity calculations). Notice the quantity of required traps is zero in this example since the suction line flow is down. The Qty. of Req. Traps includes the bottom trap in the number.



Step 10: Liquid Line - Click on the desired Pipe OD to make a selection

Choose the 0.5"OD liquid line for this split system. With the liquid line, it is best to try to pick the smallest line that will work (to minimize the refrigerant charge). ASHRAE mentions that the velocity should be kept below 300 fpm if solenoids are in the line to prevent liquid hammer. In this example, the liquid line flows up, which means the system will lose sub-cooling if it has a vertical section. Make sure the available sub-cooling exceeds the minimum sub-cooling for vertical lift. If the liquid line were flowing down, the Min Sub-cooling for Vertical Lift values would be negative which represents a gain of sub-cooling.



Step 11: Reheat Line - Click on the desired Pipe OD to make a selection

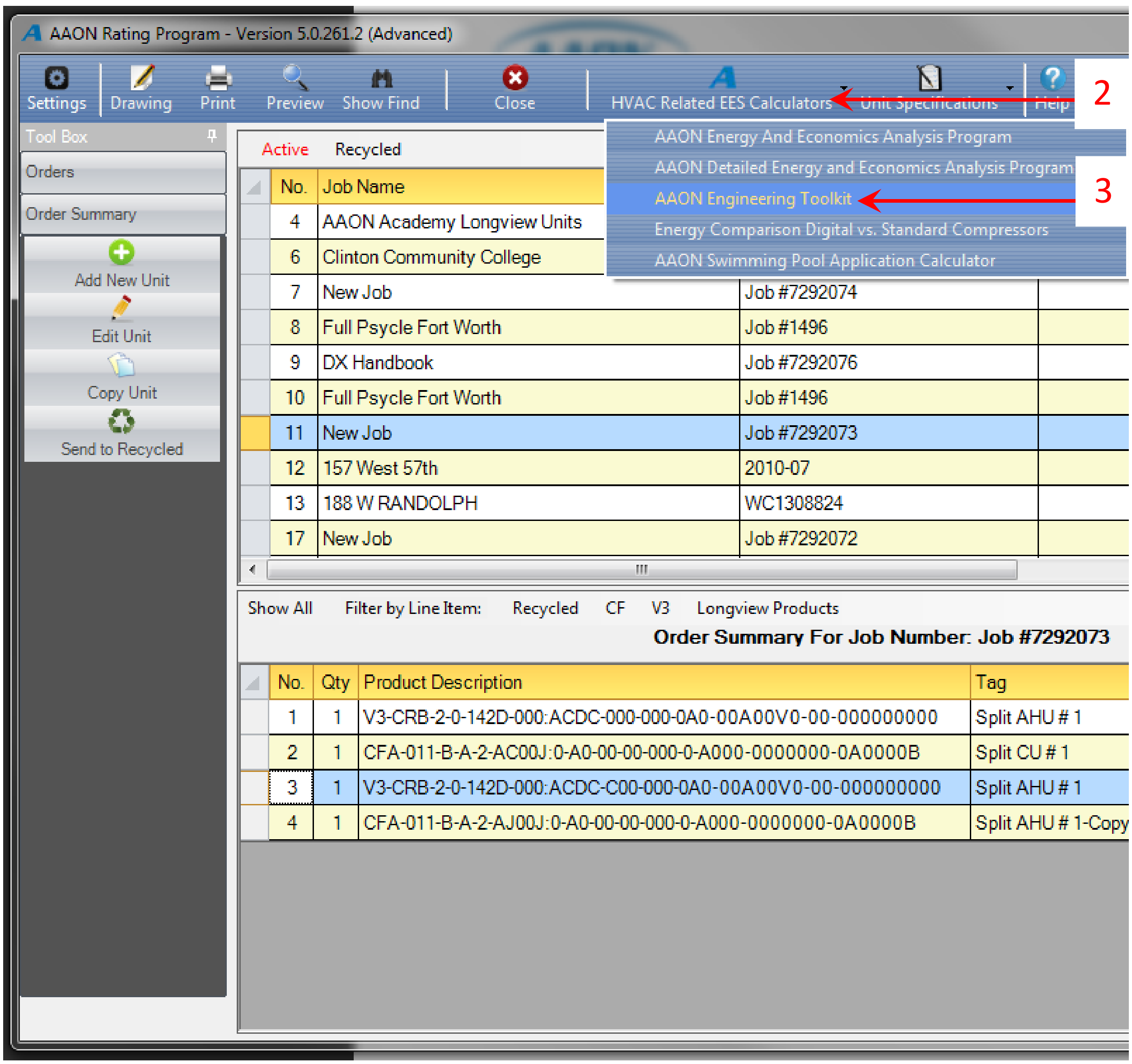
Choose the 0.5"OD reheat line for this split system. Reheat flow is modulating so pick the smallest line that is within the range of 2000 fpm - 3500 fpm.

31 EES Toolkit Line Sizing

Sometimes either it is not possible to choose a split system in ECat, or the ECat Line Sizing does not provide any lines. When ECat Line Sizing does not produce an answer, another tool available is the EES Toolkit Line Sizer.

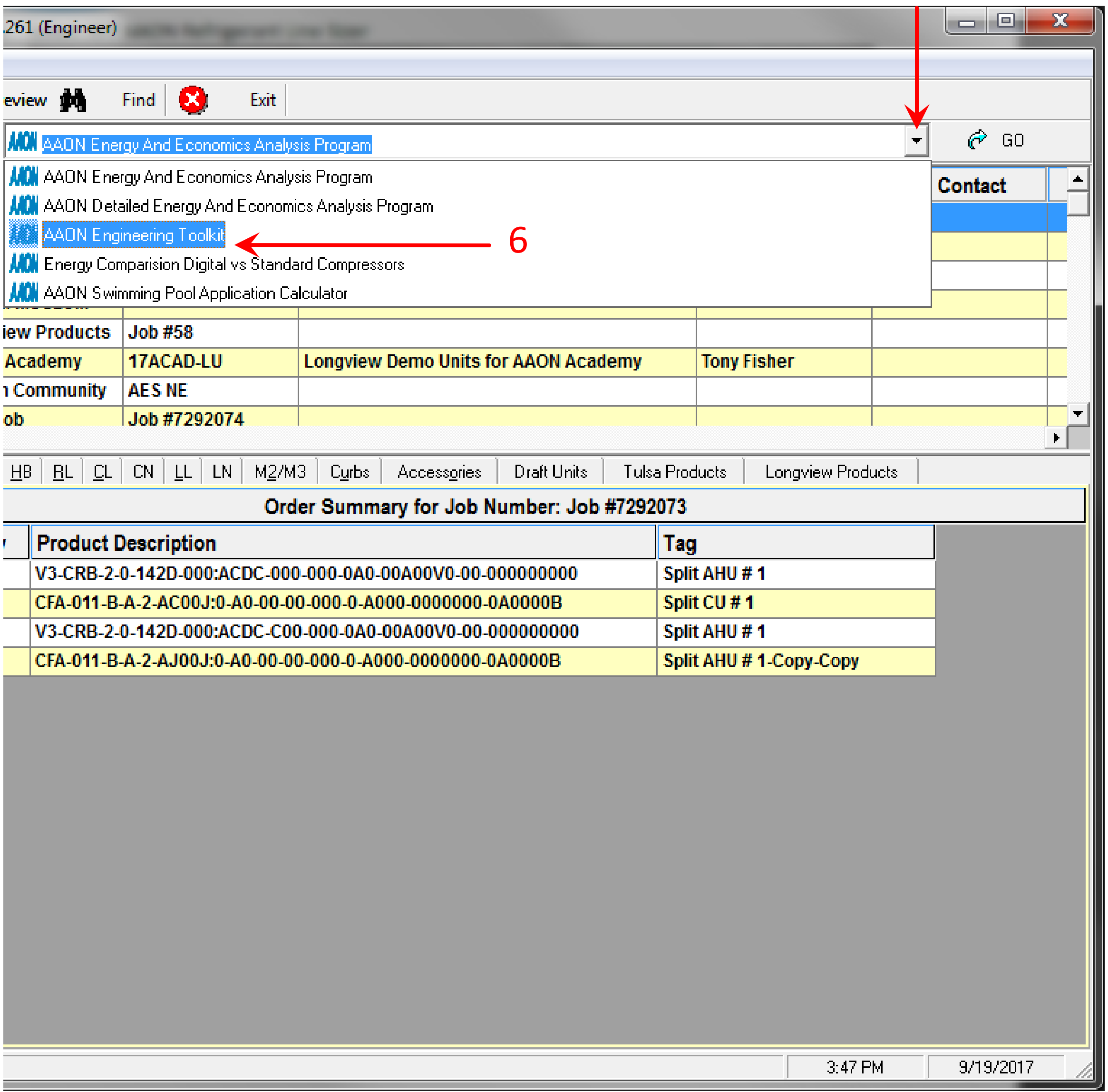
To find it in ECat Version 5:

1. Open ECat Version 5
2. Click the HVAC Related EES Calculators
3. Click AAON Engineering Toolkit



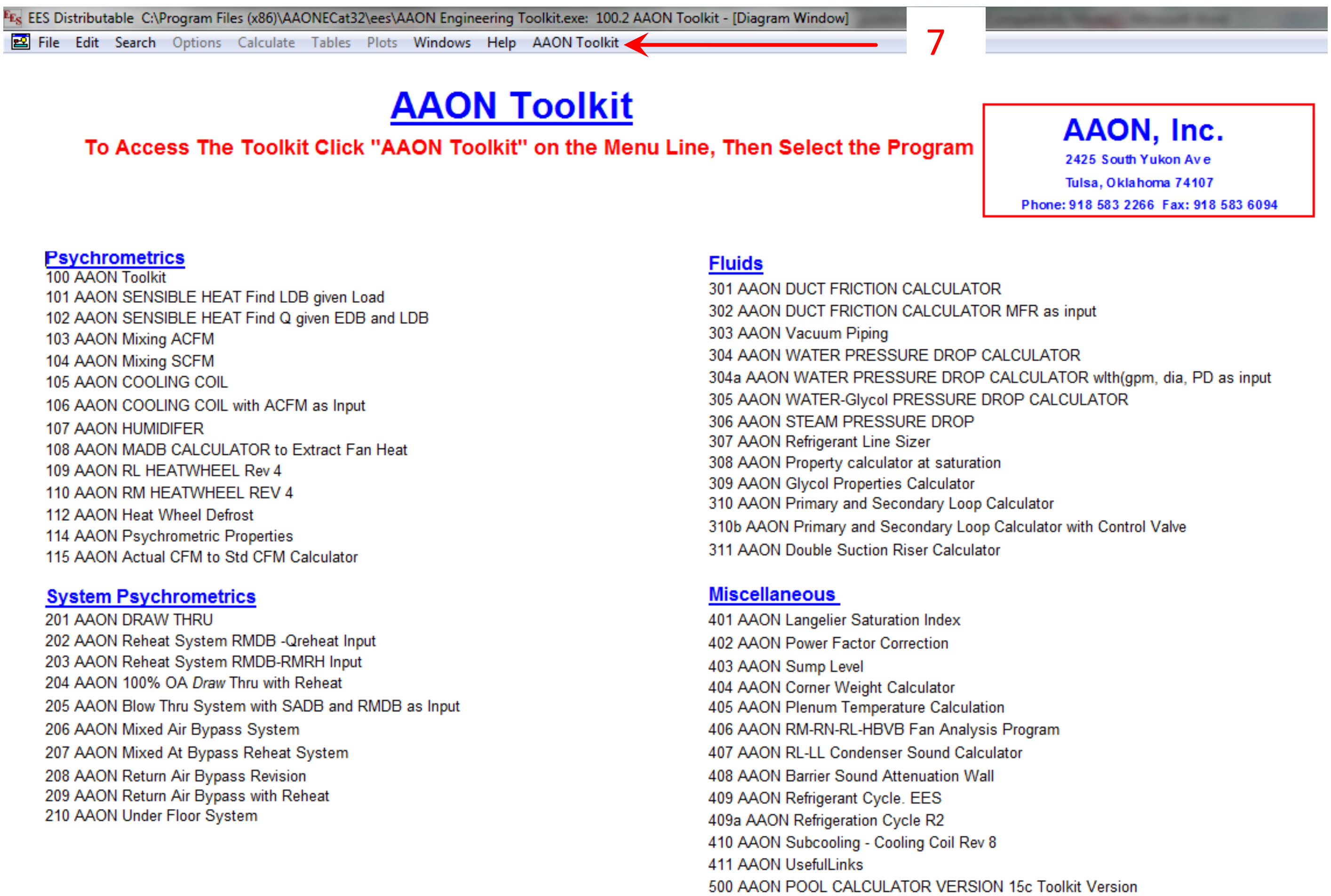
To find it in ECat Version 4:

4. Open ECat Version 4
5. Click the down arrow from the top of the screen
6. Select AAON Engineering Toolkit

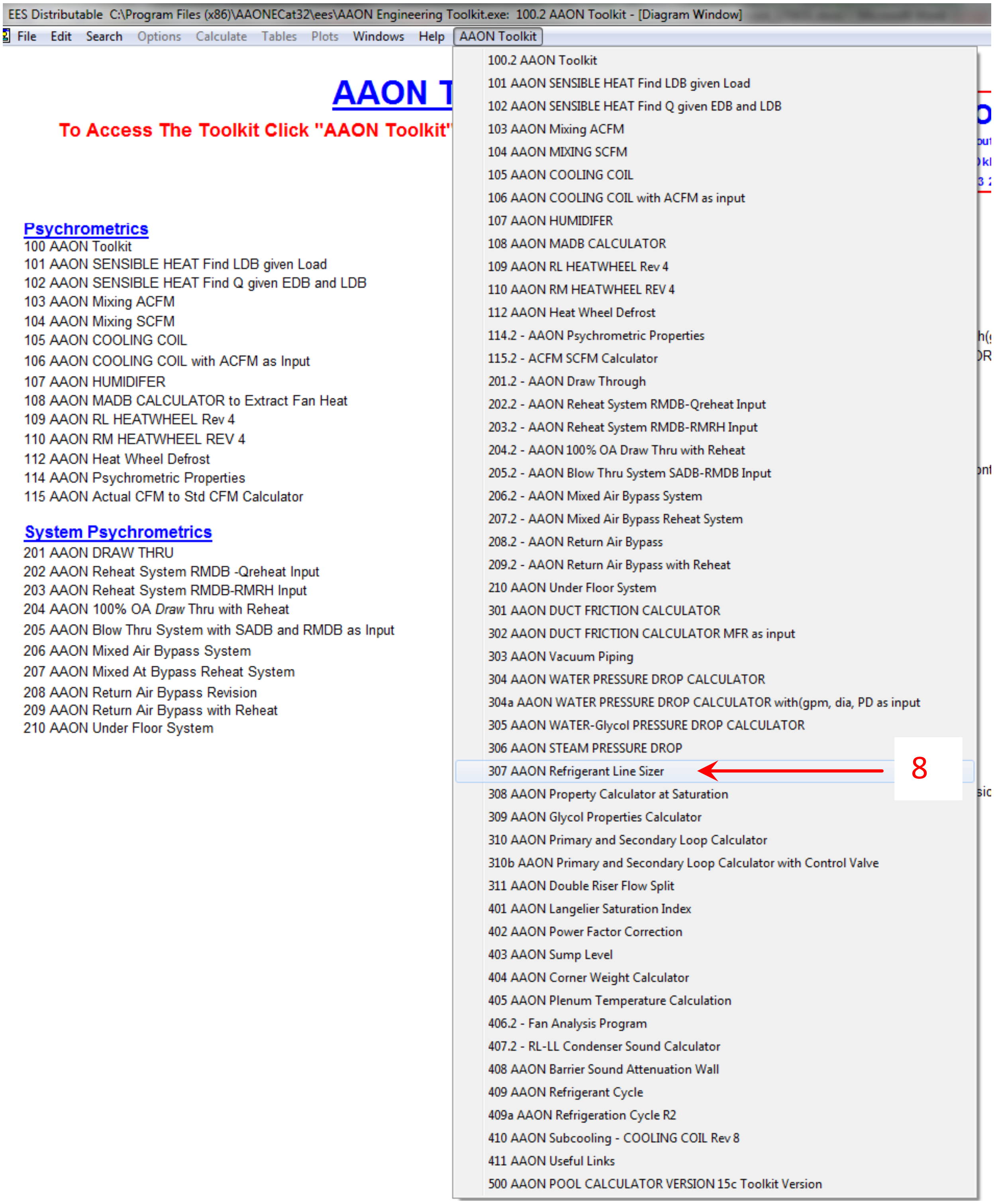


From this point it will be the same steps no matter which ECat Version (Version 5 or Version 4).

7. Click AAON Toolkit button on the top right of the screen



8. Select 307 AAON Refrigerant Line Sizer



This opens the following page. The numbered values are all user inputs. They are explained in the following section.

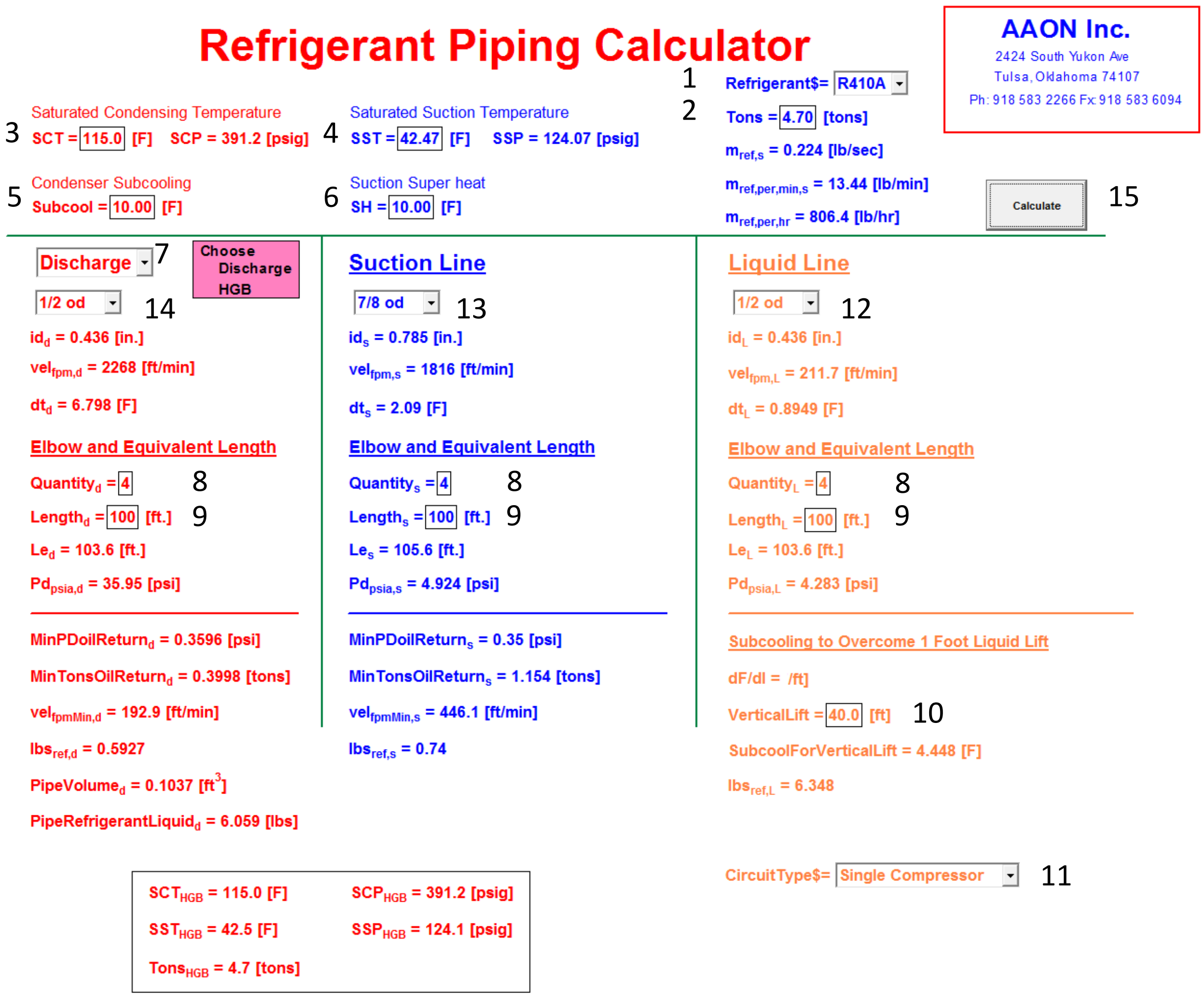


Figure 158 - EES Refrigerant Piping Calculator Inputs

31.1 EES Refrigerant Piping Calculator Inputs

The previous page has numbers for all the inputs. Each Line type (Liquid, Suction, Discharge, Hot Gas Bypass) will be broken down further and the outputs will also be explained. This page simply shows the inputs required by the user.

1. The refrigerant for CB, CF, CN, and CL Series condensing units is R-410A.
2. Change the tons to the correct tons per circuit in the condensing unit. See **Table 9 - Table 12** for the possibilities and to determine how the condensing unit is circuited. A less accurate (and not recommended) way to find tons is to divide the nominal tonnage of the condensing unit by the # of circuits to get the full load tonnage.

   The recommended and more accurate tonnage value comes from the Total Cooling Capacity from the Split System ECat rating page or the Circuit Capacities from the Condensing Unit only rating page. The Total Capacity under the Cooling Section of the rating page is in MBH, so use the total capacity and divide by 12 MBH to get tonnage and then divide by the number of circuits in the condensing unit.
   Using the capacity below for a CF-011 (2 circuits)
   = 112 MBH/12MBH per ton = 9.3 tons / 2 circuits = 4.7 tons per circuit

   <u>Split System Rating Total Capacity</u>

   Cooling Section

	Gross	Net
Total Capacity:	*111.98*	*109.30 MBH*
Sensible Capacity:	*76.90*	*74.22 MBH*
Latent Capacity:	*35.08 MBH*	
Mixed Air Temp:	*80.00 °F DB*	*67.00 °F WB*
Entering Air Temp:	*80.00 °F DB*	*67.00 °F WB*
Lv Air Temp (Coil):	*54.04 °F DB*	*53.28 °F WB*
Lv Air Temp (Unit)	*54.90 °F DB*	*53.64 °F WB*
Evap Suction Temp:	*42.47 °F*	

   If using the Condensing Unit Circuit Capacities, make sure to run a cross plot with the air handling unit to determine which suction temperature to use to get the capacity. The Condensing Unit Circuit Capacities already account for the number of circuits, so divide the circuit 1 capacity by 12MBH to get tonnage

   =59.9 MBH/12 MBH = 5.0 tons per circuit

   <u>Condensing Unit Circuit Capacities</u>

	Capacity (MBH)		
Suction Temp	Total Unit	Circuit 1	Circuit 2
Design (50°)	130.1	65.0	65.2
35°	100.5	50.5	50.0
40°	109.8	55.0	54.8
45°	119.7	59.9	59.8
50°	130.1	65.0	65.2

EES Refrigerant Piping Calculator Inputs Step 2 Continued:

Some compressor types require evaluation at two different tonnages. Again, the most accurate full load tonnage can be found as discussed above, and calculate the part loads by multiplying the full load tonnage by the percentages given below:

- Two-Step Compressors must be checked at full load tonnage and at 67% of full load tonnage.
- Tandem Compressors must be checked at full load tonnage and at 50% of full load tonnage.
- Typical AC motor VFD Speed Controlled Compressors must be checked at full load tonnage and at 50% of full load tonnage. (As compressor technology changes, this turndown can also change. Check the specific compressor capabilities to determine the % of full load velocity).
- Digital Compressors can be checked at only full load tonnage, but they have different velocity requirements than the other compressors.

  Tandem Digital Compressors still must be checked at full load tonnage and at 50% of full load tonnage using the velocity requirements for the digital compressors at both tonnages.

Table 9 - CB Series Circuits & Compressors

CB Series	Tons	# of Circuits	Compressor Options
024	2	1	Two-Step Scroll
036	3		Two-Step Scroll Digital Scroll
048	4		
060	5		

Table 10 - CF Series Circuits & Compressors

CF Series Size in Tonnage	# of Circuits	Compressor Options
002	1	Scroll Two-Step Scroll
003		Scroll Two-Step Scroll Digital Scroll
004		
005		
006		Scroll Digital Scroll
007		
009	2	
011		
013		
015		
016		
018		
020		
025		
030		
026	4 OR 2	(4) Scroll (4) Digital Scroll OR (2) Tandem Scroll (2) Tandem Digital Scroll
031		
040		
050		
060		
070		

Table 11 - CN Series Circuits & Compressors

CN Series Size in Tonnage	# of Circuits	Compressor Options
055	2	AC Motor VFD Speed Controlled Scroll Compressor
065		
075		
090		
105		
120	4	
130		
140		

Table 12 - CL Series Circuits & Compressors

CL Series Model	*Tonnage for Air-Cooled	*Tonnage for Evaporative-Cooled	# of Circuits	Compressor Options
045	40	45	4 OR 2	(4) Scroll Compressors OR (2) Tandem Scroll Compressors
060	51	58		
070	63	68		
075	69	77		
095	91	97		
100	100	106	3	Tandem Scroll Compressors
110	102	114		
125	129	134		
134	129	133		
135	136	144		
155	147	155		
170	159	174		
190	181	193	4	
210	194	212		
230	207	232		

*Note: The tonnages reflect the use of R-410A refrigerant at ambient temperature of 95°F DB/75°F WB and 50°F Saturated Suction Temperature.

EES Refrigerant Piping Calculator Inputs Continued:

3. Leave the Saturated Condensing Temperature at the default unless the value is known.
4. The Saturated Suction Temperature (SST) should be between 35°F-55°F. Use the Evap Suction Temp: from the ECat rating page if it is a split system. If a cross plot was run, use the calculated X value from the cross plot equations.

Cooling Section

	Gross	Net
Total Capacity:	*111.98*	*109.30 MBH*
Sensible Capacity:	*76.90*	*74.22 MBH*
Latent Capacity:	*35.08 MBH*	
Mixed Air Temp:	*80.00 °F DB*	*67.00 °F WB*
Entering Air Temp:	*80.00 °F DB*	*67.00 °F WB*
Lv Air Temp (Coil):	*54.04 °F DB*	*53.28 °F WB*
Lv Air Temp (Unit)	*54.90 °F DB*	*53.64 °F WB*
Evap Suction Temp:	*42.47 °F*	
Supply Air Fan:	*1 x 450AQ @ 0.90 BHP*	
SA Fan RPM / Width:	*1258 / 8.030"*	

5. Sub-cooling should be in the range of 8-15°F for straight cooling units and in the range of 2-4°F for heat pump units. See **Table 4** and **Table 5**
6. Suction Superheat should be in the range of 8-15°F.
7. This drop down menu allows either Discharge or HGB. Choose HGB when sizing the hot gas bypass line. Choose Discharge when sizing the hot gas reheat line, when determining the discharge line velocities for the heat pump mode, or when sizing discharge lines for remote condenser. For cooling only systems, this section is not used to size lines.

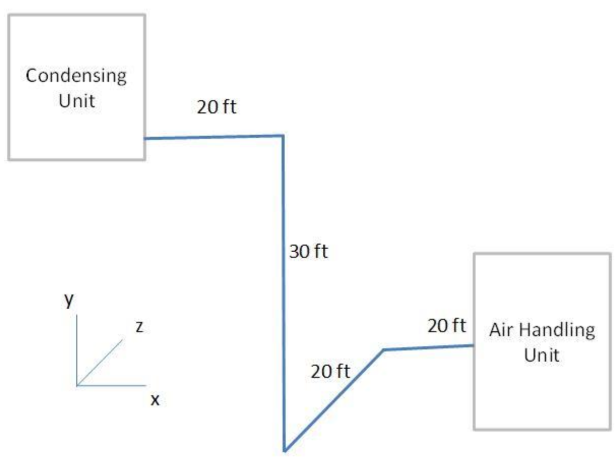


Figure 159 - Isometric Suction Line Sketch

EES Refrigerant Piping Calculator Inputs Continued:

8. Enter the total quantity of 90 degree elbows. The EES Toolkit calculator uses the 90° long radius values from the Crane Handbook (see **Table 14 -** Equivalent Length Table) to calculate the equivalent length of the bends. Typically all lines run the same route so the quantity of elbows would be the same for all lines. The exception is when the suction line has oil traps every 20 feet when the suction flow is up, or on heat pump units when the discharge line has oil traps every 12 feet when the discharge flow is up. Each oil trap will include four 90° long radius bends. From **Figure 159**, if suction is up, enter Quantity = 7
9. Enter the actual length of lines = horizontal + vertical. The equivalent length is calculated using the actual length plus the equivalent length of elbows and is the Le_L value right under the entered length.
10. Enter the vertical lift. This is the distance that the air handler is above the condensing unit. If the condensing unit is above the air handler, enter zero for the vertical lift because that means the liquid is traveling down instead of up. This value is NOT used in the calculation of equivalent length, so make sure the vertical lines are included in the calculation of the actual length input.
11. The circuit type only affects the sizing of the Hot Gas Bypass lines. If there is no hot gas bypass line to size, ignore this input.
12. This drop down menu changes the liquid line OD. The liquid line selection process will be discussed more in the Liquid Line Section.
13. This drop down menu changes the suction line OD. The suction line selection process will be discussed more in the Suction Line Selection
14. This drop down menu changes the discharge OD (as hot gas reheat or heat pump mode discharge) or hot gas bypass line OD. See hot gas reheat line selection, heat pump mode discharge velocities, or hot gas bypass line selection for more details on each selection process.
15. Now click the calculate button to see the results. The outputs will be discussed and explained in the following pages.

31.2 EES Refrigerant Piping Calculator Liquid Line Selection

Change the nominal OD of the liquid line until the values for velocity and sub-cooling for vertical lift are within the limits.

Liquid Line

1/2 od

id_L = 0.436 [in.]

1 $vel_{fpm,L}$ = 211.7 [ft/min]

dt_L = 0.8949 [F]

Elbow and Equivalent Length

$Quantity_L$ = 4

$Length_L$ = 100 [ft.]

Le_L = 103.6 [ft.]

$Pd_{psia,L}$ = 4.283 [psi]

Subcooling to Overcome 1 Foot Liquid Lift

dF/dl = /ft]

2 VerticalLift = 40.0 [ft]

3 SubcoolForVerticalLift = 4.448 [F]

$lbs_{ref,L}$ = 6.348

1. Velocity ($vel_{fpm,L}$) < 500 fpm
   Or < 300 fpm if solenoid or other electrically operated valves are in the line.
   Or < 100 fpm if line is between condenser and receiver
2. Sub-cool For Vertical Lift < 8°F (when sub-cooling = 10°F). This is to ensure at least 2°F of sub-cooling at the TXV which ensures that a pure liquid column will be entering the TXV. Take into consideration the accessories of the line. Some possible factory installed accessories are shut-off valve, liquid filter-drier, check valve. Units that have liquid line receivers will have less than 10°F of sub-cooling available. Units with receivers should try to stay below 2°F of Sub-cool For Vertical Lift. If this is not possible, a heat exchanger can be installed to gain more sub-cooling in the liquid line.

3. The lbs_{ref} value for each line can be added together to estimate how much refrigerant is needed to charge the unit in addition to the unit charge.

Note: The only way to account for the effect of additional accessories using the refrigerant piping calculator is to add to the length:

- See **Table 14 -** Equivalent Length Table for equivalent length of some accessories.
- A liquid line filter-drier is designed for 1psi pressure drop. Since the only method of adding pressure drop is through the length box, we must calculate how many equivalent feet would equal 1psi.
- Once the nominal OD is selected, divide the equivalent feet (Le_L) by the pressure drop ($Pd_{psia,L}$)
- Le_L / $Pd_{psia,L}$ = 103.6 eqft / 4.283 psi = 24 eqft/ 1 psi
- Add this equivalent feet result to the length of the lines (100 + 24 = 124) & then look at the Sub-cool For Vertical Lift and make sure it is still less than 8°F.
- Caution: Do not use the pounds of refrigerant ($lbs_{ref,L}$) while the liquid filter-drier equivalent feet are in there. To get the correct pounds of refrigerant, enter the line length in the length box as well as the quantity of elbows in the elbows box.

Change the nominal OD of the liquid line until the values for velocity and sub-cooling for vertical lift are within the limits.

31.3 EES Refrigerant Piping Calculator Suction Line Selection

Change the nominal OD of the suction line until the values for velocity and temperature loss are within the limits. It is very important to know if the suction refrigerant is flowing up vertical lift or down.

4. The velocity limits for the suction vertical risers depends on if the compressor is digital or non-digital. The minimum velocity for a digital compressor is 1500 fpm at full capacity and the part load capacities do not need to be checked, unless it is a tandem with a digital compressor. A digital tandem compressor must meet or exceed the 1500fpm at 100% capacity and at 50% capacity. This is because of the way the digital compressor operates with a full pulse but at different time intervals for different capacities. The velocity limits for the suction vertical risers on non-digital compressors must be checked at full capacity and at the minimum part load capacity using the following velocity limits (Hint - since velocity and tons are directly related, 50% of capacity (tons) is also 50% of velocity. So multiply the full load velocity by the minimum part load % and then check to make sure that actual velocity is greater than minimum velocity for oil return:

Suction Line

7/8 od

id_s = 0.785 [in.]

$vel_{fpm,s}$ = 1816 [ft/min]

dt_s = 2.09 [F]

Elbow and Equivalent Length

$Quantity_s$ = 4

$Length_s$ = 100 [ft.]

Le_s = 105.6 [ft.]

$Pd_{psia,s}$ = 4.924 [psi]

$MinPDoilReturn_s$ = 0.35 [psi]

$MinTonsOilReturn_s$ = 1.154 [tons]

$vel_{fpmMin,s}$ = 446.1 [ft/min]

$lbs_{ref,s}$ = 0.74

100% Capacity Velocity ($vel_{fpm,s}$) < 4000 fpm

Suction Riser Digital Compressor 100% Capacity Velocity ($vel_{fpm,s}$) > 1500 fpm

Suction Riser Even Tandem Digital Compressor 100% Capacity Velocity ($vel_{fpm,s}$) > 3000 fpm

On/Off Compressor & Suction Down Minimum Capacity Velocity ($vel_{fpm,s}$) > $vel_{fpmMin,s}$

5. The minimum velocity for oil return ($vel_{fpmMin,s}$) is a value calculated from data from ASHRAE Handbook - Refrigeration. The Handbook states that the line selected should provide a pressure drop equal to or greater than 0.35psi/100ft in line sizes 2”ID or less and 0.20psi/100ft in line sizes above 2”ID. From this guideline, along with pipe inside diameter, pipe roughness, refrigerant viscosity and density, the velocity was calculated in the EES Refrigeration Calculator. The minimum limit applies to the minimum part load capacity of the compressors to ensure that the oil is moving through the suction lines with the refrigerant. The upper limit of 4000 fpm is for noise concerns at full capacity. See **Table 2** & **Table 3** for velocity requirements for different compressor types. When suction flow is down, all lines can be sized for $vel_{fpmMin,s}$.

6. Suction Line Loss (dt_s) < 6°F. Check the suction line loss at full load capacity and keep it as low as possible while still meeting velocity requirements. The more temperature loss through the lines, the more capacity loss through the lines.

31.4 EES Refrigerant Piping Calculator Hot Gas Reheat Line Selection

First select Discharge from the drop down menu, and then change the nominal OD of the hot gas reheat line until the values for velocity are within the limits.

Discharge

Choose Discharge HGB

1/2 od

id_d = 0.436 [in.]

7 $vel_{fpm,d}$ = 2268 [ft/min]

dt_d = 6.798 [F]

Elbow and Equivalent Length

$Quantity_d$ = 4

$Length_d$ = 100 [ft.]

Le_d = 103.6 [ft.]

$Pd_{psia,d}$ = 35.95 [psi]

$MinPDoilReturn_d$ = 0.3596 [psi]

$MinTonsOilReturn_d$ = 0.3998 [tons]

$vel_{fpmMin,d}$ = 192.9 [ft/min]

$lbs_{ref,d}$ = 0.5927

$PipeVolume_d$ = 0.1037 [ft^3]

$PipeRefrigerantLiquid_d$ = 6.059 [lbs]

7. 2000 fpm < Velocity ($vel_{fpm,d}$) < 3500 fpm

Keep the velocity as high as possible, but stay under 3500 fpm. It is advantageous to keep the velocity high because the hot gas reheat valve is modulating and normally it will not be running at 100% open. That is why it is important to keep the hot gas reheat line as small as possible.

id_d = inner diameter
$vel_{fpm,d}$ = velocity in the line
dt_d = temperature loss due to friction

$Quantity_d$ = number of 90 long radius bends
Le_d = Equivalent length of discharge line
$Pd_{psia,d}$ = pressure loss due to friction

$MinTonsOilReturn_d$ = the minimum tons of refrigerant required in the line to still return oil up a riser
$vel_{fpmMin,d}$ = the minimum velocity required for oil to return up a riser
$lbs_{ref,d}$ = lbs of refrigerant in the line

$PipeVolume_d$ = the calculated volume of the pipe
$PipeRefrigerantLiquid_d$ = if the refrigerant were a liquid in this line, the mass of the liquid. This value is no longer needed since liquid reheat is no longer used.

31.5 EES Refrigerant Piping Calculator Hot Gas Bypass Line Selection

First select the circuit type from the bottom right of the Refrigerant Piping Calculator Screen. The compressor selection affects the tons of refrigerant used in the HGB calculations and the tonnage used is visible in the bottom left of screen in the box. Notice the single compressor will use 66% of the tonnage, tandem compressor will use 33% of the tonnage, and two-step compressor will use 44% of the tonnage. The hot gas bypass line would never have 100% of the tonnage through it and hot gas bypass will not start running until the compressor is running at its lowest capacity. If too much gas is bypassed, the condenser coil would not have enough refrigerant and the unit would have low head pressure problems. The HGB drop down selection also changes the SCT to 90°F and the SST to 32°F, because these are the conditions that the refrigerant gas would be.

CircuitType$= Single Compressor

Single Compressor
Tandem Compressor
Two-Step

SCT_{HGB} = 90.0 [F] SCP_{HGB} = 274.5 [psig]

SST_{HGB} = 32.0 [F] SSP_{HGB} = 101.1 [psig]

$Tons_{HGB}$ = 3.239 [tons]

Select HGB from the drop down menu, and then change the nominal OD of the hot gas bypass line until the values for velocity & temperature loss are within the limits.

HGB

Choose Discharge HGB

1/2 od

id_d = 0.436 [in.]

8 $vel_{fpm,d}$ = 1576 [ft/min]

dt_d = 3.254 [F]

Elbow and Equivalent Length

$Quantity_d$ = 4

$Length_d$ = 100 [ft.]

Le_d = 103.6 [ft.]

$Pd_{psia,d}$ = 13.28 [psi]

$MinPDoilReturn_d$ = 0.3596 [psi]

$MinTonsOilReturn_d$ = 0.3831 [tons]

$vel_{fpmMin,d}$ = 225.7 [ft/min]

$lbs_{ref,d}$ = 0.4263

$PipeVolume_d$ = 0.1037 [ft^3]

$PipeRefrigerantLiquid_d$ = 6.059 [lbs]

8. 2000 fpm < Velocity ($vel_{fpm,d}$) < 3500 fpm

Keep the velocity as high as possible, but stay under 3500 fpm. It is advantageous to keep the velocity high because the hot gas bypass valve is modulating and normally it will not be running at 100% open. That is why it is important to keep the hot gas bypass line as small as possible.

31.6 EES Refrigerant Piping Calculator Heat Pump Mode Discharge Line Velocities

The heat pump uses the same line for the suction line in cooling mode and the discharge line in heating mode. When sizing lines for heat pump, use the suction line selection process to check suction line criteria, and then use the same OD from the suction line to check the discharge line for heat pump mode. If a line size cannot be found to work for both lines, then a double suction riser will be necessary. It is very important to know which flow is up. If suction flow is up, make sure to evaluate the suction line based on the vertical lift velocities, and the discharge line for minimum velocity for oil return. If discharge is up, make sure to evaluate the discharge line based on the vertical lift velocities and the suction line for minimum velocity for oil return.

Cooling Section

	Gross	Net
Total Capacity:	*109.90*	*107.21 MBH*
Sensible Capacity:	*76.04*	*73.36 MBH*
Latent Capacity:	*33.86 MBH*	
Mixed Air Temp:	*80.00 °F DB*	*67.00 °F WB*
Entering Air Temp:	*80.00 °F DB*	*67.00 °F WB*
Lv Air Temp (Coil):	*54.33 °F DB*	*53.59 °F WB*
Lv Air Temp (Unit)	*55.19 °F DB*	*53.95 °F WB*
Evap Suction Temp:	*42.98 °F*	
Supply Air Fan:	*1 x 450AQ @ 0.90 BHP*	
SA Fan RPM / Width:	*1258 / 8.030"*	

Heating Section

Primary Heat Type:	*Heat Pump*
Total Capacity:	*125.5 MBH*
OA Temp:	*47.0 DB / 43.0°F WB*
RA Temp:	*75.0 °F DB / 65.0 °F WB*
Entering Air Temp:	*47.0 DB / 43.0 °F WB*
Leaving Air Temp:	*86.8 DB / 60.0°F WB*
Auxiliary Heating Type*:	***No Heat***
Heating CFM:	*2800*
Total Capacity:	
OA Temp:	*47.0 DB / 43.0°F WB*
RA Temp:	*75.0 °F DB / 65.0 °F WB*

Figure 160 - Cooling and Heating Rating Page

Saturated Condensing Temperature

SCT = 115.0 [F] SCP = 391.2 [psig]

Condenser Subcooling

Subcool = 10.00 [F]

Saturated Suction Temperature

SST = 42.98 [F] SSP = 125.27 [psig]

Suction Super heat

SH = 10.00 [F]

Refrigerant$= R410A

Tons = 4.58 [tons]

$m_{ref,s}$ = 0.218 [lb/sec]

$m_{ref,per,min,s}$ = 13.09 [lb/min]

$m_{ref,per,hr}$ = 785.4 [lb/hr]

Figure 161 - Cooling Mode Inputs

Suction Line

7/8 od

id_s = 0.785 [in.]

$vel_{fpm,s}$ = 1754 [ft/min]

dt_s = 1.959 [F]

Elbow and Equivalent Length

$Quantity_s$ = 4

$Length_s$ = 100 [ft.]

Le_s = 105.6 [ft.]

$Pd_{psia,s}$ = 4.647 [psi]

$MinPDoilReturn_s$ = 0.35 [psi]

$MinTonsOilReturn_s$ = 1.161 [tons]

$vel_{fpmMin,s}$ = 444.3 [ft/min]

$lbs_{ref,s}$ = 0.7465

Suction Line

3/4 od

id_s = 0.666 [in.]

$vel_{fpm,s}$ = 2436 [ft/min]

dt_s = 4.39 [F]

Elbow and Equivalent Length

$Quantity_s$ = 4

$Length_s$ = 100 [ft.]

Le_s = 104.8 [ft.]

$Pd_{psia,s}$ = 10.41 [psi]

$MinPDoilReturn_s$ = 0.35 [psi]

$MinTonsOilReturn_s$ = 0.7486 [tons]

$vel_{fpmMin,s}$ = 398.2 [ft/min]

$lbs_{ref,s}$ = 0.5373

Both of these suction line sizes would be acceptable. The velocity is under 4000 fpm and the suction line temperature loss (dts) is less than 6°F.

For part loads, make sure the velocity at the lowest part load exceeds the minimum velocity for oil return (vel fpmMin,s). See suction line selection for digital compressor exceptions.

Now check the discharge line sizes in heating mode to make sure one of these line sizes will also work for the discharge size (see the next page).

Engineering for help sizing the suction/discharge line. They will be able to access the heating mode inputs that are not available through the current ECat rating page.

Refrigerant$= R410A

Tons = 5.23 [tons]

$m_{ref,s}$ = 0.227 [lb/sec]

$m_{ref,per,min,s}$ = 13.62 [lb/min]

$m_{ref,per,hr}$ = 817.2 [lb/hr]

Saturated Condensing Temperature

SCT = 97.0 [F] SCP = 304.1 [psig]

Condenser Subcooling

Subcool = 10.00 [F]

Saturated Suction Temperature

SST = 32.00 [F] SSP = 101.05 [psig]

Suction Super heat

SH = 10.00 [F]

Figure 162 - Heating Mode Inputs

Discharge

Choose Discharge HGB

7/8 od

id_d = 0.785 [in.]

1 $vel_{fpm,d}$ = 906.2 [ft/min]

dt_d = 0.5834 [F]

Elbow and Equivalent Length

$Quantity_d$ = 4

$Length_d$ = 100 [ft.]

Le_d = 105.6 [ft.]

$Pd_{psia,d}$ = 2.562 [psi]

$MinPDoilReturn_d$ = 0.2843 [psi]

$MinTonsOilReturn_d$ = 1.631 [tons]

2 $vel_{fpmMin,d}$ = 282.6 [ft/min]

$lbs_{ref,d}$ = 1.503

$PipeVolume_d$ = 0.3361 [ft^3]

$PipeRefrigerantLiquid_d$ = 20.98 [lbs]

Discharge

Choose Discharge HGB

3/4 od

id_d = 0.666 [in.]

$vel_{fpm,d}$ = 1259 [ft/min]

dt_d = 1.303 [F]

Elbow and Equivalent Length

$Quantity_d$ = 4

$Length_d$ = 100 [ft.]

Le_d = 104.8 [ft.]

$Pd_{psia,d}$ = 5.725 [psi]

$MinPDoilReturn_d$ = 0.3036 [psi]

$MinTonsOilReturn_d$ = 1.091 [tons]

$vel_{fpmMin,d}$ = 262.6 [ft/min]

$lbs_{ref,d}$ = 1.082

$PipeVolume_d$ = 0.2419 [ft^3]

$PipeRefrigerantLiquid_d$ = 15.1 [lbs]

1. Velocity ($vel_{fpm,d}$) < 3500 fpm to prevent noise.

Digital Compressor Discharge Vertical Riser Full Load Velocity - ($vel_{fpm,d}$) > 900 fpm

Even Tandem Digital Compressor Discharge Vertical Riser Full Load Velocity-
($vel_{fpm,d}$) > 1800 fpm

2. On/off Compressor or Discharge Down - Minimum Load Velocity ($vel_{fpm,d}$) > $vel_{fpmMin,d}$

31.7 EES Refrigerant Piping Calculator Quick Reference - Scroll Compressor with HGB

Hot Gas Reheat & Hot Gas Bypass lines must include purge circuit for oil return.

*Tons = Total capacity MBH / 12MBH/ #of circuits
3.9 Tons = 46.4MBH/ 12MBH/ 1 circuit

Refrigerant Piping Calculator

AAON Inc.
2424 South Yukon Ave
Tulsa, Oklahoma 74107
Ph: 918 583 2266 Fx 918 583 6094

Saturated Condensing Temperature
SCT = 115.0 [F] SCP = 391.2 [psig]

Condenser Subcooling
Subcool = 10.00 [F]

Saturated Suction Temperature
SST = 44.50 [F] SSP = 128.90 [psig]

Suction Super heat
SH = 10.00 [F]

Refrigerant$= R410A
* Tons = 3.90 [tons]
$m_{ref,s}$ = 0.185 [lb/sec]
$m_{ref,per,min,s}$ = 11.13 [lb/min]
$m_{ref,per,hr}$ = 667.7 [lb/hr]

Calculate

 # HGB

Choose Discharge HGB

3/8 od
id_d = 0.311 [in.]
$vel_{fpm,d}$ = 2482 [f < 3500 fpm > 2000 fpm
dt_d = 11.51 [F]

Elbow and Equivalent Length
$Quantity_d$ = 4
$Length_d$ = 100 [ft.]
Le_d = 103.2 [ft.]
$Pd_{psia,d}$ = 46.95 [psi]

$MinPDoilReturn_d$ = 0.4116 [psi]
$MinTonsOilReturn_d$ = 0.1667 [tons]
$vel_{fpmMin,d}$ = 193 [ft/min]
$lbs_{ref,d}$ = 0.2169
$PipeVolume_d$ = 0.05275 [ft^3]
$PipeRefrigerantLiquid_d$ = 3.083 [lbs]

Suction Line

7/8 od
id_s = 0.785 [in.]
$vel_{fpm,s}$ = 1452 [ft/min] < 4000 fpm
dt_s = 1.42 [F] < 2°F to minimize capacity loss

Elbow and Equ
$Quantity_s$ = 6
$Length_s$ = 100 [ft.]
Le_s = 108.4 [ft.]
$Pd_{psia,s}$ = 3.433 [psi]

$MinPDoilReturn_s$ = 0.35 [psi]
$MinTonsOilReturn_s$ = 1.179 [tons]
$vel_{fpmMin,s}$ = 439 [ft/min] < velocity < 1452 fpm
$lbs_{ref,s}$ = 0.7662

Liquid Line

Select small line to minimize refrigerant charge

3/8 od
id_L = 0.311 [in.]
$vel_{fpm,L}$ = 344.5 [ft/min] < 500 fpm < 300 fpm (if solenoid valve in line)
dt_L = 3.282 [F]

Elbow and Equivalent Length
$Quantity_L$ = 4
$Length_L$ = 100 [ft.]
Le_L = 103.2 [ft.]
$Pd_{psia,L}$ = 15.71 [psi]

Subcooling to Overcome 1 Foot Liquid Lift
dF/dl = :/ft]
VerticalLift = 40.0 [ft]
SubcoolForVerticalLift = 6.836 [F < 8°F to ensure no flashing at TXV
$lbs_{ref,L}$ = 3.23

SCT_{HGB} = 90.0 [F]	SCP_{HGB} = 274.5 [psig]
SST_{HGB} = 32.0 [F]	SSP_{HGB} = 101.1 [psig]
$Tons_{HGB}$ = 2.596 [tons]	

CircuitType$= Single Compressor

Select Single Compressor for this example. This input only affects the HGB line sizing. Notice the tons used for HGB and SCT & SST conditions change for HGB.

31.8 EES Refrigerant Piping Calculator Quick Reference - 2-Step Scroll Compressor with HGB

Hot Gas Reheat & Hot Gas Bypass lines must include purge circuit for oil return.

*Tons = Total capacity MBH / 12MBH/ #of circuits
3.8 Tons = 45.9MBH/ 12MBH/ 1 circuit

Refrigerant Piping Calculator

AAON Inc.
2424 South Yukon Ave
Tulsa, Oklahoma 74107
Ph: 918 583 2266 Fx: 918 583 6094

Saturated Condensing Temperature
SCT = 115.0 [F] SCP = 391.2 [psig]

Condenser Subcooling
Subcool = 10.00 [F]

Saturated Suction Temperature
SST = 44.90 [F] SSP = 129.87 [psig]

Suction Super heat
SH = 10.00 [F]

Refrigerant$= R410A
* Tons = 3.80 [tons]
$m_{ref,s}$ = 0.181 [lb/sec]
$m_{ref,per,min,s}$ = 10.84 [lb/min]
$m_{ref,per,hr}$ = 650.3 [lb/hr]

Calculate

HGB

Choose Discharge HGB

3/8 od

id_d = 0.311 [in.]
$vel_{fpm,d}$ = 1609 [ft/min] < 3500 fpm; > 2000 fpm
dt_d = 5.067 [F]

Elbow and Equivalent Length
$Quantity_d$ = 4
$Length_d$ = 100 [ft.]
Le_d = 103.2 [ft.]
$Pd_{psia,d}$ = 20.67 [psi]

$MinPDoilReturn_d$ = 0.4116 [psi]
$MinTonsOilReturn_d$ = 0.1667 [tons]
$vel_{fpmMin,d}$ = 193 [ft/min]
$lbs_{ref,d}$ = 0.2169
$PipeVolume_d$ = 0.05275 [ft^3]
$PipeRefrigerantLiquid_d$ = 3.083 [lbs]

Suction Line

7/8 od

id_s = 0.785 [in.]
$vel_{fpm,s}$ = 1405 [ft/min] < 4000 fpm
dt_s = 1.336 [F] < 2°F to minimize capacity loss

Elbow and Equ
$Quantity_s$ = 6
$Length_s$ = 100 [ft.]
Le_s = 108.4 [ft.]
$Pd_{psia,s}$ = 3.246 [psi]

$MinPDoilReturn_s$ = 0.35 [psi]
$MinTonsOilReturn_s$ = 1.183 [tons]
$vel_{fpmMin,s}$ = 437.6 [ft/min] //
$lbs_{ref,s}$ = 0.7715

Liquid Line

Select small line to minimize refrigerant charge

3/8 od

id_L = 0.311 [in.]
$vel_{fpm,L}$ = 335.6 [ft/min] < 500 fpm; < 300 fpm (if solenoid valve in line)
dt_L = 3.126 [F]

Elbow and Equivalent Length
$Quantity_L$ = 4
$Length_L$ = 100 [ft.]
Le_L = 103.2 [ft.]
$Pd_{psia,L}$ = 14.96 [psi]

Subcooling to Overcome 1 Foot Liquid Lift
dF/dl = :/ft]
VerticalLift = 40.0 [ft]
SubcoolForVerticalLift = 6.680 [F] < 8°F to ensure no flashing at TXV
$lbs_{ref,L}$ = 3.23

SCT_{HGB} = 90.0 [F]	SCP_{HGB} = 274.5 [psig]
SST_{HGB} = 32.0 [F]	SSP_{HGB} = 101.1 [psig]
$Tons_{HGB}$ = 1.683 [tons]	

CircuitType$= Two-Step

Select Two-Step for this example. This input only affects the HGB line sizing. Notice the tons used for HGB and SCT & SST conditions change for HGB.

// 2/3 * vel > vel min
2/3 * 1405 fpm > 437.6 fpm
937 fpm > 437.6 fpm

31.9 EES Refrigerant Piping Calculator Quick Reference - Tandem Scroll Compressor with HGB

Hot Gas Reheat & Hot Gas Bypass lines must include purge circuit for oil return.

*Tons = Total capacity MBH / 12MBH/ #of circuits
13.6 Tons = 326MBH/ 12MBH/ 2 circuits

Refrigerant Piping Calculator

AAON Inc.
2424 South Yukon Ave
Tulsa, Oklahoma 74107
Ph: 918 583 2266 Fx 918 583 6094

Saturated Condensing Temperature
SCT = 115.0 [F] SCP = 391.2 [psig]

Saturated Suction Temperature
SST = 40.47 [F] SSP = 119.43 [psig]

Refrigerant$= R410A
* Tons = 13.60 [tons]
$m_{ref,s}$ = 0.650 [lb/sec]
$m_{ref,per,min,s}$ = 38.98 [lb/min]
$m_{ref,per,hr}$ = 2339 [lb/hr]

Condenser Subcooling
Subcool = 10.00 [F]

Suction Super heat
SH = 10.00 [F]

Calculate

HGB

Choose Discharge HGB

1/2 od

id_d = 0.436 [in.]
$vel_{fpm,d}$ = 2360 [< 3500 fpm > 2000 fpm
dt_d = 6.989 [F]

Elbow and Equivalent Length
$Quantity_d$ = 4
$Length_d$ = 100 [ft.]
Le_d = 103.6 [ft.]
$Pd_{psia,d}$ = 28.51 [psi]

$MinPDoilReturn_d$ = 0.3596 [psi]
$MinTonsOilReturn_d$ = 0.3831 [tons]
$vel_{fpmMin,d}$ = 225.7 [ft/min]
$lbs_{ref,d}$ = 0.4263
$PipeVolume_d$ = 0.1037 [ft^3]
$PipeRefrigerantLiquid_d$ = 6.059 [lbs]

Suction Line

1-3/8 od

id_s = 1.245 [in.]
$vel_{fpm,s}$ = 2167 [ft/min] < 4000 fpm
dt_s = 1.805 [F] < 2°F to minimize capacity loss

Elbow and Equ
$Quantity_s$ = 6
$Length_s$ = 100 [ft.]
Le_s = 113.8 [ft.]
$Pd_{psia,s}$ = 4.145 [psi]

$MinPDoilReturn_s$ = 0.35 [psi]
$MinTonsOilReturn_s$ = 3.854 [tons]
$vel_{fpmMin,s}$ = 614.2 [ft/min] //
$lbs_{ref,s}$ = 1.798

Liquid Line

Select small line to minimize refrigerant charge

5/8 od

id_L = 0.555 [in.]
$vel_{fpm,L}$ = 378.9 [ft/min] < 500 fpm
< 300 fpm (if solenoid valve in line)
dt_L = 1.958 [F]

Elbow and Equivalent Length
$Quantity_L$ = 4
$Length_L$ = 100 [ft.]
Le_L = 104 [ft.]
$Pd_{psia,L}$ = 9.371 [psi]

Subcooling to Overcome 1 Foot Liquid Lift
dF/dl = ?/ft]
VerticalLift = 40.0 [ft]
SubcoolForVerticalLift = 5.511 [F] < 8°F to ensure no flashing at TXV
$lbs_{ref,L}$ = 10.29

SCT_{HGB} = 90.0 [F]	SCP_{HGB} = 274.5 [psig]
SST_{HGB} = 32.0 [F]	SSP_{HGB} = 101.1 [psig]
$Tons_{HGB}$ = 4.85 [tons]	

CircuitType$= Tandem Compressor

Select Tandem Compressor for this example. This input only affects the HGB line sizing. Notice the tons used for HGB and SCT & SST conditions change for HGB.

// 1/2 * vel > vel min
1/2 * 2167 fpm > 614 fpm
1084 fpm > 614 fpm

31.10 EES Refrigerant Piping Calculator Quick Reference - Digital Scroll Compressor with HGRH

\# Hot Gas Reheat & Hot Gas Bypass lines must include purge circuit for oil return.

*Tons = Total capacity MBH / 12MBH/ #of circuits
4.67 Tons = 112.2MBH/ 12MBH/ 2 circuit

Refrigerant Piping Calculator

AAON Inc.
2424 South Yukon Ave
Tulsa, Oklahoma 74107
Ph: 918 583 2266 Fx: 918 583 6094

Saturated Condensing Temperature
SCT = 115.0 [F]   SCP = 391.2 [psig]

Condenser Subcooling
Subcool = 10.00 [F]

Saturated Suction Temperature
SST = 42.41 [F]   SSP = 123.93 [psig]

Suction Super heat
SH = 10.00 [F]

Refrigerant$= R410A
* Tons = 4.67 [tons]
$m_{ref,s}$ = 0.223 [lb/sec]
$m_{ref,per,min,s}$ = 13.36 [lb/min]
$m_{ref,per,hr}$ = 801.3 [lb/hr]

Calculate

Choose Discharge for Hot Gas Reheat

\# **Discharge**   Choose Discharge HGB

1/2 od
id_d = 0.436 [in.]
$vel_{fpm,d}$ = 2254 [f   < 3500 fpm
> 2000 fpm
dt_d = 6.718 [F]

Elbow and Equivalent Length
$Quantity_d$ = 4
$Length_d$ = 100 [ft.]
Le_d = 103.6 [ft.]
$Pd_{psia,d}$ = 35.53 [psi]

$MinPDoilReturn_d$ = 0.3596 [psi]
$MinTonsOilReturn_d$ = 0.3997 [tons]
$vel_{fpmMin,d}$ = 192.9 [ft/min]
$lbs_{ref,d}$ = 0.5926
$PipeVolume_d$ = 0.1037 [ft^3]
$PipeRefrigerantLiquid_d$ = 6.059 [lbs]

Suction Line

7/8 od
id_s = 0.785 [in.]   > 1500 fpm (if suction up)
$vel_{fpm,s}$ = 1807 [ft   < 4000 fpm
dt_s = 2.124 [F]   < 2°F to minimize capacity loss

Elbow and Equi
$Quantity_s$ = 6
$Length_s$ = 100 [ft.]
Le_s = 108.4 [ft.]
$Pd_{psia,s}$ = 5 [psi]

$MinPDoilReturn_s$ = 0.35 [psi]
$MinTonsOilReturn_s$ = 1.154 [tons]
$vel_{fpmMin,s}$ = 446.4 [ft/min]   //
$lbs_{ref,s}$ = 0.7392

Liquid Line   Select small line to minimize refrigerant charge

1/2 od
id_L = 0.436 [in.]
$vel_{fpm,L}$ = 210.4 [ft/min]   < 500 fpm
< 300 fpm (if solenoid valve in line)
dt_L = 0.8846 [F]

Elbow and Equivalent Length
$Quantity_L$ = 4
$Length_L$ = 100 [ft.]
Le_L = 103.6 [ft.]
$Pd_{psia,L}$ = 4.234 [psi]

Subcooling to Overcome 1 Foot Liquid Lift
dF/dl = :/ft]
VerticalLift = 40.0 [ft]
SubcoolForVerticalLift = 4.438 [F]   < 8°F to ensure no flashing at TXV
$lbs_{ref,L}$ = 6.348

SCT_{HGB} = 115.0 [F]	SCP_{HGB} = 391.2 [psig]
SST_{HGB} = 42.4 [F]	SSP_{HGB} = 123.9 [psig]
$Tons_{HGB}$ = 4.67 [tons]	

CircuitType$= Single Compressor

This selection only affects the HGB line sizing. It does not matter what is in here when sizing hot gas reheat lines.

// Digital compressor suction up line velocity must > 1500 fpm
Vel(min) used on suction down

31.11 EES Refrigerant Piping Calculator Quick Reference - Tandem Digital Scroll Compressor with HGRH

Hot Gas Reheat & Hot Gas Bypass lines must include purge circuit for oil return.

*Tons = Total capacity MBH / 12MBH/ #of circuits
11.43 Tons = 274.3MBH/ 12MBH/ 2 circuits

Refrigerant Piping Calculator

AAON Inc.
2424 South Yukon Ave
Tulsa, Oklahoma 74107
Ph: 918 583 2266 Fx: 918 583 6094

Refrigerant$= R410A

Saturated Condensing Temperature
SCT = 115.0 [F] SCP = 391.2 [psig]

Saturated Suction Temperature
SST = 45.07 [F] SSP = 130.28 [psig]

* Tons = 11.43 [tons]

$m_{ref,s}$ = 0.543 [lb/sec]

Condenser Subcooling
Subcool = 10.00 [F]

Suction Super heat
SH = 10.00 [F]

$m_{ref,per,min,s}$ = 32.60 [lb/min]

$m_{ref,per,hr}$ = 1956 [lb/hr]

Calculate

Choose Discharge for Hot Gas Reheat

Discharge

Choose Discharge HGB

3/4 od

id_d = 0.666 [in.]

$vel_{fpm,d}$ = 2336 [

< 3500 fpm
> 2000 fpm

dt_d = 4.425 [F]

Elbow and Equivalent Length

$Quantity_d$ = 4

$Length_d$ = 100 [ft.]

Le_d = 104.8 [ft.]

$Pd_{psia,d}$ = 23.4 [psi]

$MinPDoilReturn_d$ = 0.3036 [psi]

$MinTonsOilReturn_d$ = 1.138 [tons]

$vel_{fpmMin,d}$ = 232.7 [ft/min]

$lbs_{ref,d}$ = 1.395

$PipeVolume_d$ = 0.2419 [ft^3]

$PipeRefrigerantLiquid_d$ = 14.14 [lbs]

Suction Line

1-1/8 od

id_s = 1.025 [in.]

//

$vel_{fpm,s}$ = 2471 [ft/min] < 4000 fpm

dt_s = 2.856 [F]

< 2°F to minimize capacity loss

Elbow and Equi

$Quantity_s$ = 6

$Length_s$ = 100 [ft.]

Le_s = 110.2 [ft.]

$Pd_{psia,s}$ = 6.954 [psi]

$MinPDoilReturn_s$ = 0.35 [psi]

$MinTonsOilReturn_s$ = 2.411 [tons]

$vel_{fpmMin,s}$ = 521.1 [ft/min]

$lbs_{ref,s}$ = 1.319

Liquid Line

Select small line to minimize refrigerant charge

5/8 od

id_L = 0.555 [in.]

$vel_{fpm,L}$ = 316.9 [ft/min]

< 500 fpm
< 300 fpm (if solenoid valve in line)

dt_L = 1.407 [F]

Elbow and Equivalent Length

$Quantity_L$ = 4

$Length_L$ = 100 [ft.]

Le_L = 104 [ft.]

$Pd_{psia,L}$ = 6.733 [psi]

Subcooling to Overcome 1 Foot Liquid Lift

dF/dl = :/ft]

VerticalLift = 40.0 [ft]

SubcoolForVerticalLift = 4.960 [F] < 8°F to ensure no flashing at TXV

$lbs_{ref,L}$ = 10.29

SCT_{HGB} = 115.0 [F]	SCP_{HGB} = 391.2 [psig]
SST_{HGB} = 45.1 [F]	SSP_{HGB} = 130.3 [psig]
$Tons_{HGB}$ = 11.43 [tons]	

CircuitType$= Single Compressor

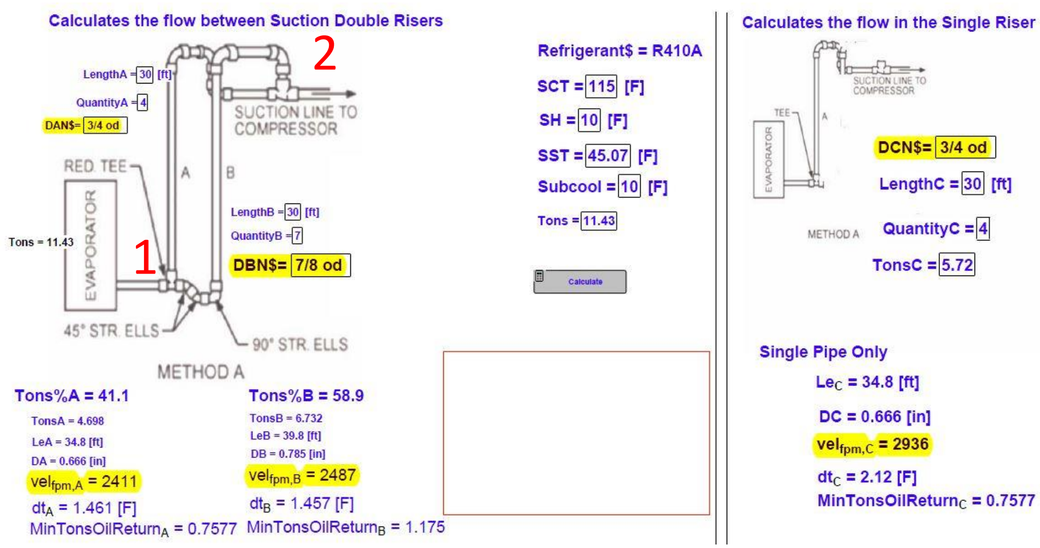
This selection only affects the HGB line sizing. It does not matter what is in here when sizing hot gas reheat lines.

// 1/2 * vel > 1500 fpm (digital)
1/2 * 2471 fpm > 1500 fpm
1236 fpm is not > 1500 fpm
DOUBLE SUCTION RISER IS NEEDED

31.12 EES Double Suction Riser Quick Reference - Tandem Digital Scroll Compressor with HGRH

The double suction riser calculator is in the EES Toolkit and it is 311 AAON Double Riser Flow Split. The calculator works based on the fact that the pressure drop from (1) to (2) will be the same regardless of which path is taken. Path B has a few more bends, so that is why even with the same line OD for both path A and path B, the values are slightly different. The inputs for this toolkit are:

- Length A = Length B = Length C= vertical riser height
- Quantity A = Quantity C = number of bends in path A
- Quantity B = number of bends in path B
- Enter the same full load refrigerant conditions that were entered into the Refrigerant Piping Calculator
- Tons C = minimum capacity tonnage
- Change the drop down line sizes to get acceptable velocities at full load and part load
    - If DAN and DBN are different sizes, make sure DBN is the larger size
    - DAN = DCN



31.13 EES Refrigerant Piping Calculator Quick Reference - VFD Controlled Scroll Compressor with HGRH

Hot Gas Reheat & Hot Gas Bypass lines must include purge circuit for oil return.

*Tons = Total capacity MBH / 12MBH/ #of circuits
32 Tons = 770MBH/ 12MBH/ 2 circuits

Refrigerant Piping Calculator

AAON Inc.
2424 South Yukon Ave
Tulsa, Oklahoma 74107
Ph: 918 583 2266 Fx: 918 583 6094

Refrigerant$= R410A

* Tons = 32.00 [tons]

$m_{ref,s}$ = 1.521 [lb/sec]

$m_{ref,per,min,s}$ = 91.26 [lb/min]

$m_{ref,per,hr}$ = 5476 [lb/hr]

Calculate

Saturated Condensing Temperature

SCT = 115.0 [F] SCP = 391.2 [psig]

Condenser Subcooling

Subcool = 10.00 [F]

Saturated Suction Temperature

SST = 45.00 [F] SSP = 130.11 [psig]

Suction Super heat

SH = 10.00 [F]

Choose Discharge for Hot Gas Reheat

Discharge

Choose Discharge HGB

1-1/8 od

id_d = 1.025 [in.]

$vel_{fpm,d}$ = 2762 [f < 3500 fpm > 2000 fpm

dt_d = 3.728 [F]

Elbow and Equivalent Length

$Quantity_d$ = 4

$Length_d$ = 100 [ft.]

Le_d = 106.8 [ft.]

$Pd_{psia,d}$ = 19.72 [psi]

$MinPDoilReturn_d$ = 0.2556 [psi]

$MinTonsOilReturn_d$ = 3.263 [tons]

$vel_{fpmMin,d}$ = 281.7 [ft/min]

$lbs_{ref,d}$ = 3.304

$PipeVolume_d$ = 0.573 [ft^3]

$PipeRefrigerantLiquid_d$ = 33.49 [lbs]

Suction Line

1-5/8 od //

id_s = 1.481 [in.]

$vel_{fpm,s}$ = 3318 [ft/min] < 4000 fpm

dt_s = 3.545 [F] ? < 2°F to minimize capacity loss

Elbow and Equi

$Quantity_s$ = 8

$Length_s$ = 100 [ft.]

Le_s = 120.8 [ft.]

$Pd_{psia,s}$ = 8.626 [psi]

$MinPDoilReturn_s$ = 0.35 [psi]

$MinTonsOilReturn_s$ = 6.392 [tons]

$vel_{fpmMin,s}$ = 662.8 [ft/min]

$lbs_{ref,s}$ = 2.751

// 50% * vel > vel minimum
50% * 3318 fpm > 662.8 fpm
1659 fpm > 662.8 fpm

Liquid Line

Select small line to minimize refrigerant charge

1-1/8 od

id_L = 1.025 [in.]

$vel_{fpm,L}$ = 260.1 [ft/min] < 500 fpm
< 300 fpm (if solenoid valve in line)

dt_L = 0.4808 [F]

Elbow and Equivalent Length

$Quantity_L$ = 4

$Length_L$ = 100 [ft.]

Le_L = 106.8 [ft.]

$Pd_{psia,L}$ = 2.301 [psi]

Subcooling to Overcome 1 Foot Liquid Lift

dF/dl = :/ft]

VerticalLift = 40.0 [ft]

SubcoolForVerticalLift = 4.034 [F] < 8°F to ensure no flashing at TXV

$lbs_{ref,L}$ = 35.08

CircuitType$= Single Compressor

This selection only affects the HGB line sizing. It does not matter what is in here when sizing hot gas reheat lines.

? The suction temp loss is greater than 2°F, but the 50% velocity on the 2-1/8"OD line was too close to the minimum velocity for oil return, so the decision is to take the capacity loss to ensure oil return at minimum load.

31.14 EES Refrigerant Piping Calculator Quick Reference - Heat Pump Scroll Compressor

Heat pumps must be checked in cooling mode and heating mode conditions. Velocities are more critical in the mode where the refrigerant is traveling up the line. The ECat ratings do not make it possible to know the SST & SCT for the heating mode, so ask Applications Engineering for help. The discharge line OD must be the same OD as the suction line.

Heat Pump Cooling Mode

*Tons = Total capacity MBH / 12MBH/ #of circuits
3.88 Tons = 46.5MBH/ 12MBH/ 1 circuit

Choose Discharge and change the OD to match the suction line OD since this same line is used as both a suction and discharge line. This will give a general idea of which size line will work for both discharge and suction. The heating mode conditions will be different. Ask the AAON Applications department for the heating mode SST, SCT, and capacity.

Refrigerant Piping Calculator

AAON Inc.
2424 South Yukon Ave
Tulsa, Oklahoma 74107
Ph: 918 583 2266 Fx: 918 583 6094

Saturated Condensing Temperature
SCT = 115.0 [F] SCP = 391.2 [psig]

Condenser Subcooling
Subcool = 10.00 [F]

Saturated Suction Temperature
SST = 44.55 [F] SSP = 129.02 [psig]

Suction Super heat
SH = 10.00 [F]

Refrigerant$ = R410A
* Tons = 3.88 [tons]
$m_{ref,s}$ = 0.185 [lb/sec]
$m_{ref,per,min,s}$ = 11.07 [lb/min]
$m_{ref,per,hr}$ = 664.2 [lb/hr]

Calculate

Discharge (#)

Choose Discharge HGB

3/4 od

id_d = 0.666 [in.]
$vel_{fpm,d}$ = 794.9 [ft/min] <>
dt_d = 0.5755 [F]

Elbow and Equivalent Length
$Quantity_d$ = 4
$Length_d$ = 100 [ft.]
Le_d = 104.8 [ft.]
$Pd_{psia,d}$ = 3.043 [psi]

$MinPDoilReturn_d$ = 0.3036 [psi]
$MinTonsOilReturn_d$ = 1.137 [tons]
$vel_{fpmMin,d}$ = 232.9 [ft/min] <>
$lbs_{ref,d}$ = 1.393
$PipeVolume_d$ = 0.2419 [ft^3]
$PipeRefrigerantLiquid_d$ = 14.14 [lbs]

Suction Line

3/4 od

id_s = 0.666 [in.]
$vel_{fpm,s}$ = 2006 [ft/min] < 4000 fpm
dt_s = 3.197 [F]

? < 2°F to minimize capacity loss

Elbow and Equ
$Quantity_s$ = 8
$Length_s$ = 100 [ft.]
Le_s = 109.6 [ft.]
$Pd_{psia,s}$ = 7.734 [psi]

$MinPDoilReturn_s$ = 0.35 [psi]
$MinTonsOilReturn_s$ = 0.7607 [tons]
$vel_{fpmMin,s}$ = 393.2 [ft/min]
$lbs_{ref,s}$ = 0.552

Liquid Line

Select small line to minimize refrigerant charge

3/8 od

id_L = 0.311 [in.]
$vel_{fpm,L}$ = 342.8 [ft/min] < 500 fpm
< 300 fpm (if solenoid valve in line)
dt_L = 3.251 [F]

Elbow and Equivalent Length
$Quantity_L$ = 4
$Length_L$ = 100 [ft.]
Le_L = 103.2 [ft.]
$Pd_{psia,L}$ = 15.56 [psi]

Subcooling to Overcome 1 Foot Liquid Lift
dF/dl = :/ft]
VerticalLift = 40.0 [ft]
SubcoolForVerticalLift = 6.804 [F] < 8°F to ensure no flashing at TXV
$lbs_{ref,L}$ = 3.23

vel > vel min <>
794.9 fpm >232.9
This line must also be checked using heating mode conditions

? The suction temp loss is greater than 2°F, but since the suction line size must also be the discharge line, the decision is to take the capacity loss to ensure oil return in heating mode.

CircuitType$ = Single Compressor

This selection only affects the HGB line sizing. It does not matter what is in here when sizing heat pump discharge lines.

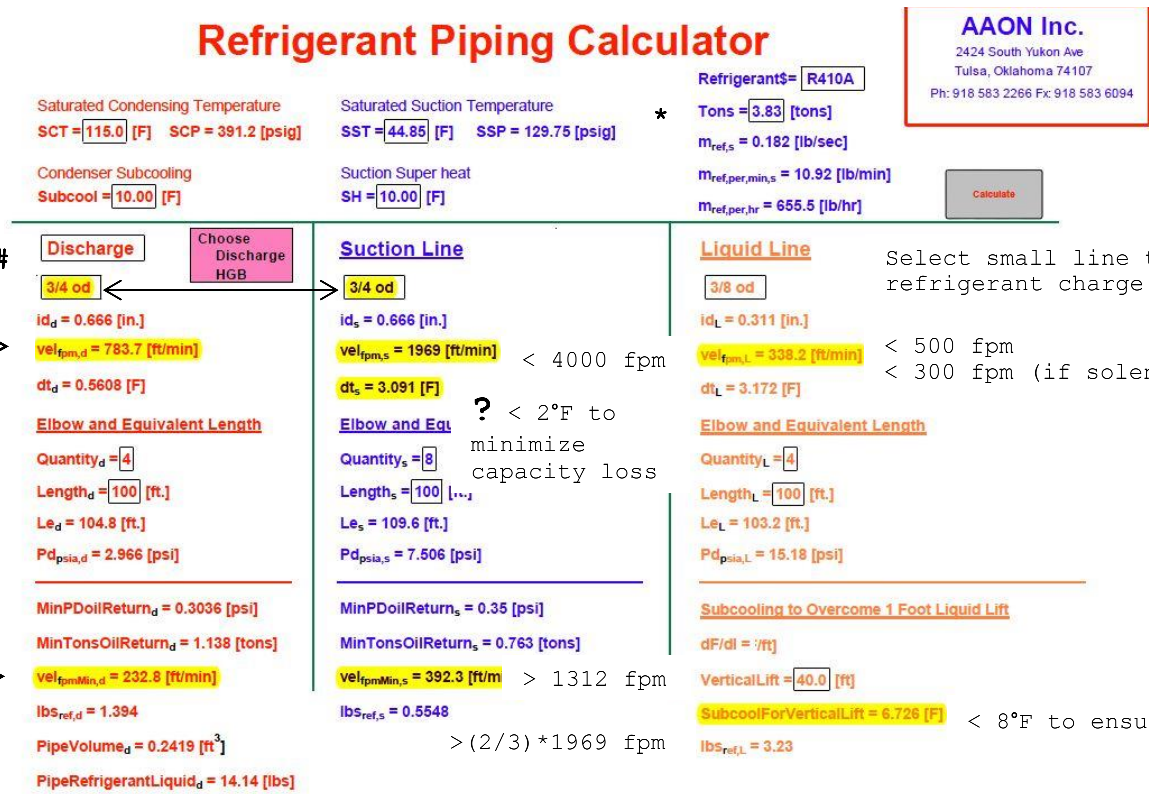
31.15 EES Refrigerant Piping Calculator Quick Reference - Heat Pump 2-Step Scroll Compressor

Heat pumps must be checked in cooling mode and heating mode conditions. Velocities are more critical in the mode where the refrigerant is traveling up the line. The ECat ratings do not make it possible to know the SST & SCT for the heating mode, so ask Applications Engineering for help. The discharge line OD must be the same OD as the suction line.

Heat Pump Cooling Mode

*Tons = Total capacity MBH / 12MBH/ #of circuits
3.83 Tons = 46.0MBH/ 12MBH/ 1 circuit

Choose Discharge and change the OD to match the suction line OD since this same line is used as both a suction and discharge line. This will give a general idea of which size line will work for both discharge and suction. The heating mode conditions will be different. Ask the AAON Applications department for the heating mode SST, SCT, and capacity.



*

\#

<>

< 4000 fpm

? < 2°F to minimize capacity loss

> 1312 fpm

>(2/3)*1969 fpm

Select small line to minimize refrigerant charge

< 500 fpm
< 300 fpm (if solenoid valve in line)

< 8°F to ensure no flashing at TXV

<>

2/3*vel > vel min
523 fpm > 232.8
but this line must still be checked using heating mode conditions

<>

? The suction temp loss is greater than 2°F, but since the suction line size must also be the discharge line, the decision is to take the capacity loss to ensure oil return in heating mode. If discharge flow is up, the vertical riser should be sized using a smaller OD than the horizontal runs.

This selection only affects the HGB line sizing. It does not matter what is in here when sizing heat pump discharge lines.

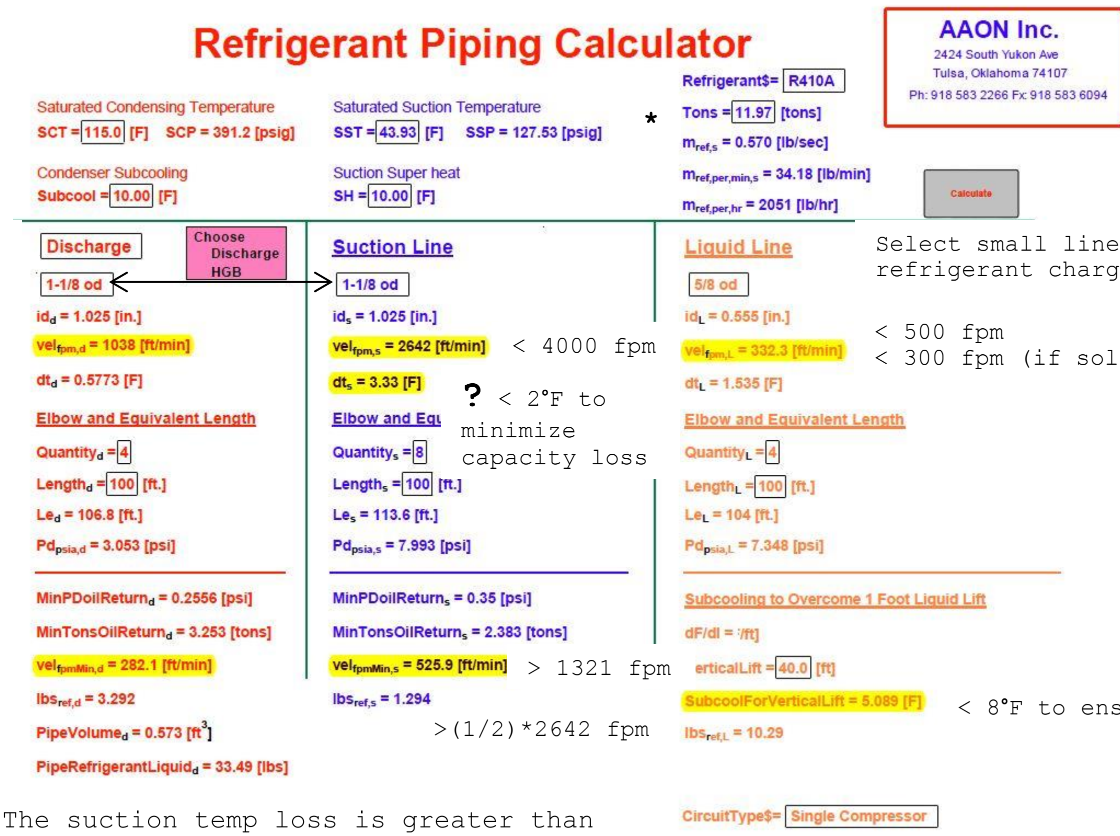
31.16 EES Refrigerant Piping Calculator Quick Reference - Heat Pump Tandem Scroll Compressor

Heat pumps must be checked in cooling mode and heating mode conditions. Velocities are more critical in the mode where the refrigerant is traveling up the line. The ECat ratings do not make it possible to know the SST & SCT for the heating mode, so ask Applications Engineering for help. The discharge line OD must be the same OD as the suction line.

Heat Pump Cooling Mode

*Tons = Total capacity MBH / 12MBH/ #of circuits
11.97 Tons = 287.3MBH/ 12MBH/ 2 circuits

Choose Discharge and change the OD to match the suction line OD since this same line is used as both a suction and discharge line. This will give a general idea of which size line will work for both discharge and suction. The heating mode conditions will be different. Ask the AAON Applications department for the heating mode SST, SCT, and capacity.



\# <> < 4000 fpm

? < 2°F to minimize capacity loss

<> > 1321 fpm

>(1/2)*2642 fpm

Select small line to minimize refrigerant charge

< 500 fpm
< 300 fpm (if solenoid valve in line)

< 8°F to ensure no flashing at TXV

<>

1/2*vel > vel min
519 fpm > 232.8
but this line must still be checked using heating mode conditions

? The suction temp loss is greater than 2°F, but since the suction line size must also be the discharge line, the decision is to take the capacity loss to ensure oil return in heating mode. The 7/8 OD for suction line exceeds the maximum velocity of 4000fpm

This selection only affects the HGB line sizing. It does not matter what is in here when sizing heat pump discharge lines.

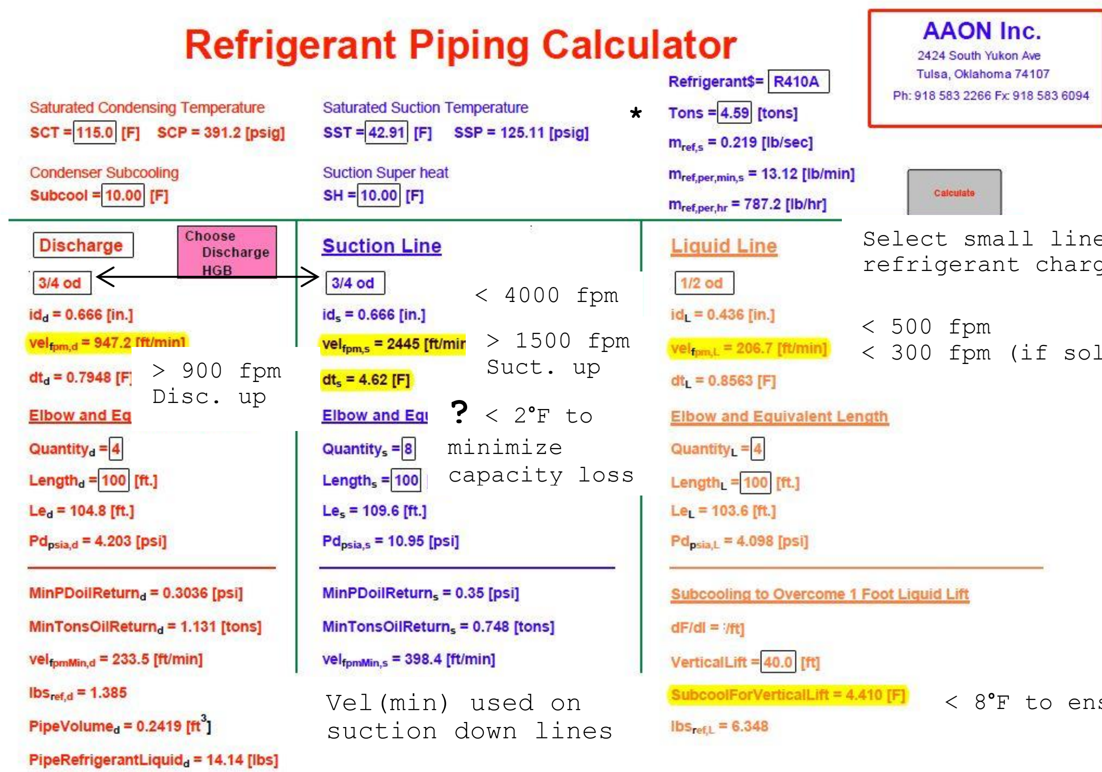
31.17 EES Refrigerant Piping Calculator Quick Reference - Heat Pump Digital Scroll Compressor

Heat pumps must be checked in cooling mode and heating mode conditions. Velocities are more critical in the mode where the refrigerant is traveling up the line. The ECat ratings do not make it possible to know the SST & SCT for the heating mode, so ask Applications Engineering for help. The discharge line OD must be the same OD as the suction line.

Heat Pump Cooling Mode

*Tons = Total capacity MBH / 12MBH/ #of circuits
4.59 Tons = 110.2MBH/ 12MBH/ 2 circuits

Choose Discharge and change the OD to match the suction line OD since this same line is used as both a suction and discharge line. This will give a general idea of which size line will work for both discharge and suction. The heating mode conditions will be different. Ask AAON Applications for the heating mode SST, SCT, and capacity. For digital compressors discharge velocity must be > 900fpm if discharge flow is up. If discharge flow is down, keep velocity above min velocity for oil return.



Select small line to minimize refrigerant charge

< 500 fpm
< 300 fpm (if solenoid valve in line)

< 8°F to ensure no flashing at TXV

This selection only affects the HGB line sizing. It does not matter what is in here when sizing heat pump discharge lines.

? The suction temp loss is greater than 2°F, but since the suction line size must also be the discharge line, the decision is to take the capacity loss to ensure oil return in heating mode. If discharge flow is up, the vertical riser should be sized using a smaller OD than the horizontal

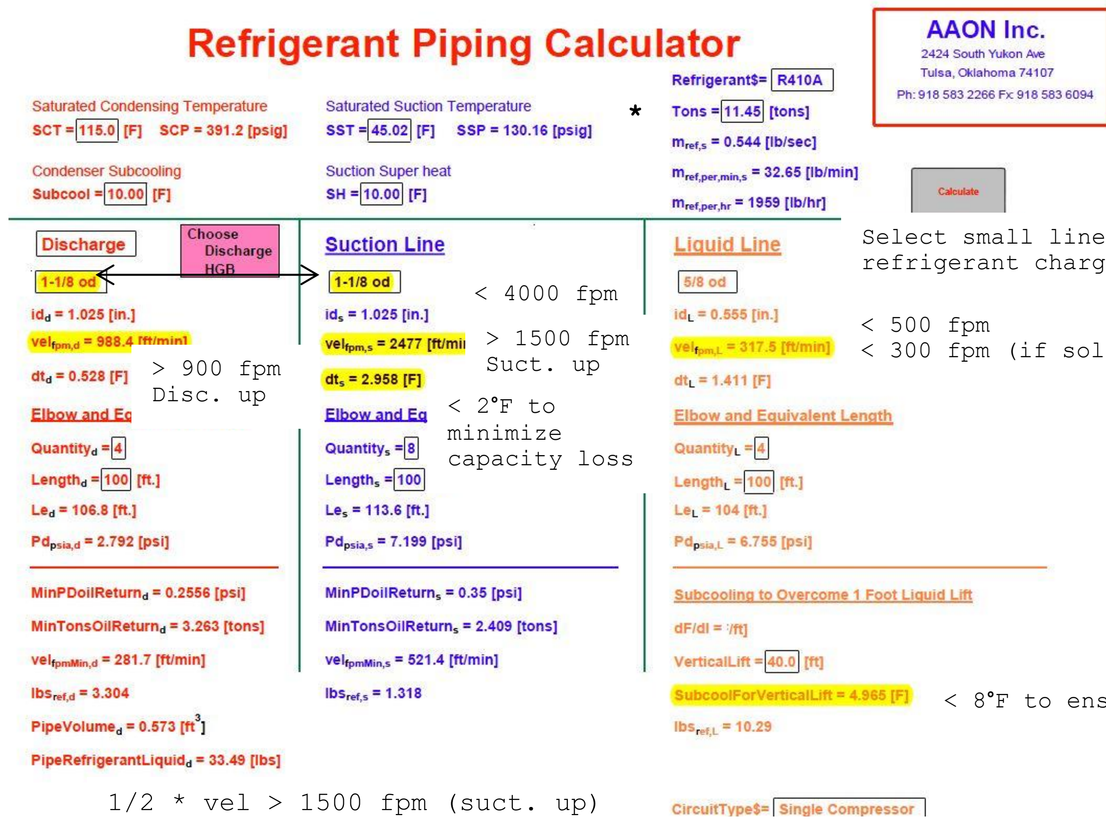
31.18 EES Refrigerant Piping Calculator Quick Reference - Heat Pump Tandem Digital Scroll Compressor

Heat pumps must be checked in cooling mode and heating mode conditions. Velocities are more critical in the mode where the refrigerant is traveling up the line. The ECat ratings do not make it possible to know the SST & SCT for the heating mode, so ask Applications Engineering for help. The discharge line OD must be the same OD as the suction line.

Heat Pump Cooling Mode

*Tons = Total capacity MBH / 12MBH/ #of circuits
11.45 Tons = 274.9MBH/ 12MBH/ 2 circuits

Choose Discharge and change the OD to match the suction line OD since this same line is used as both a suction and discharge line. This will give a general idea of which size line will work for both discharge and suction. The heating mode conditions will be different. Ask AAON Applications for the heating mode SST, SCT, and capacity.
For digital compressors discharge velocity must be > 900fpm if discharge flow is up. If discharge flow is down, keep velocity above min velocity for oil return.



#

> 900 fpm Disc. up

< 4000 fpm

> 1500 fpm Suct. up

< 2°F to minimize capacity loss

Select small line to minimize refrigerant charge

< 500 fpm
< 300 fpm (if solenoid valve in line)

< 8°F to ensure no flashing at TXV

This selection only affects the HGB line sizing. It does not matter what is in here when sizing heat pump discharge lines.

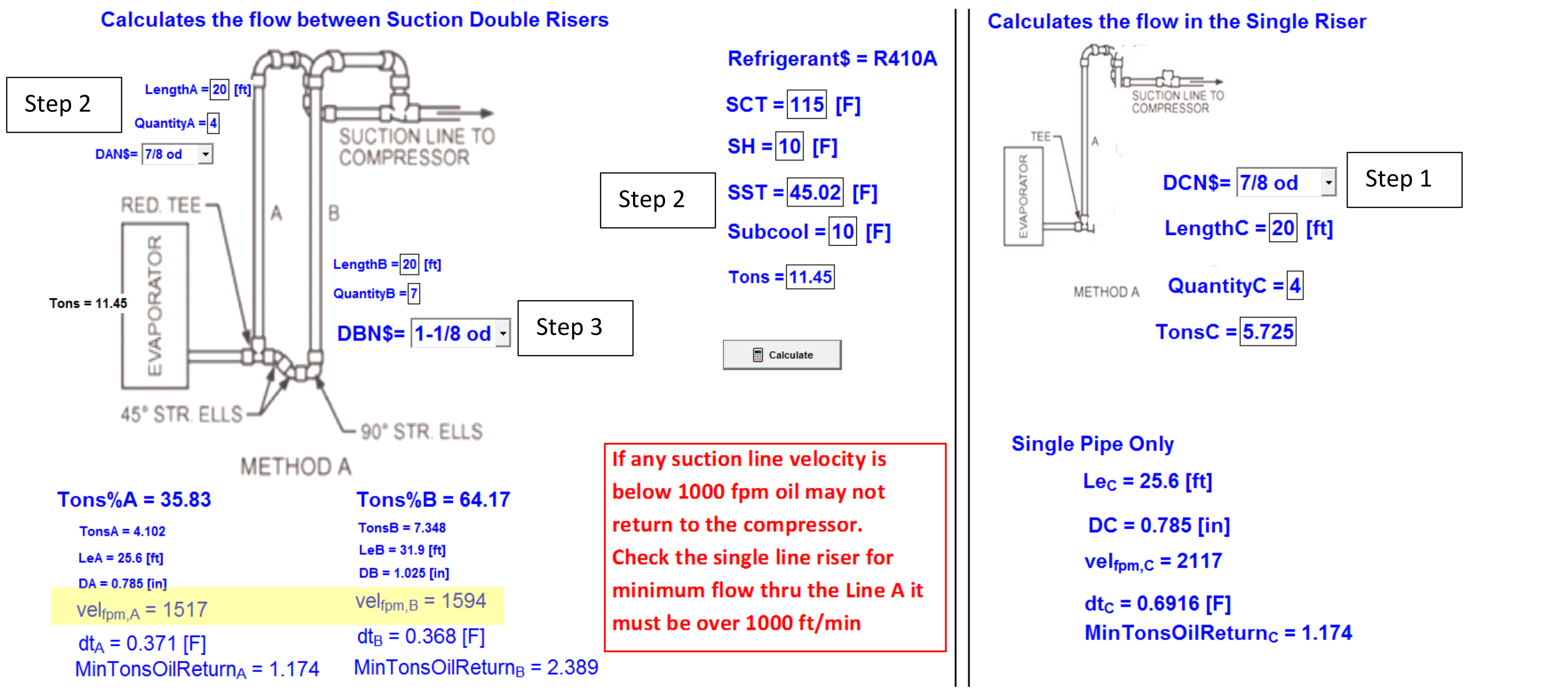
1/2 * vel > 1500 fpm (suct. up)
1/2 * 2477 fpm > 1500 fpm
1239 fpm is not > 1500 fpm
DOUBLE SUCTION RISER IS NEEDED IF SUCTION IS UP

1/2 * vel > 900 fpm (dis. up)
1/2 * 988 fpm > 900 fpm
494 fpm is not > 900 fpm
DOUBLE DISCHARGE RISER IS NEEDED IF DISCHARGE IS UP

31.19 EES Double Suction Riser Quick Reference - Heat Pump Tandem Digital Scroll Compressor

In this example, the suction flow is up, but a single line does not fit the requirements for 100% and 50% capacity.

Step 1 - Open 311 Double Riser Calculator, and determine the correct line for 50% capacity in the second part of the calculator (DCN).
Step 2 - Input the 100% capacity data into the first section and change DAN to the OD determined in **Step 1**.
Step 3 - Change DBN to get velocity A and velocity B above 1500fpm.
Step 4 - Go back into the 307 Refrigerant Piping Calculator and check discharge velocities for line A and the closest line that would represent the cross sectional area of line A & line B combined. The cross sectional area of a 7/8"OD line is 0.48 in^2 and of a 1-1/8"OD is 0.83in^2, so the combined area is 1.31in^2. A 1-3/8"OD line has a 1.22in^2 cross sectional area, while a 1-5/8"OD line has a 1.72in^2 cross sectional area. The values at 1-3/8"OD will be the closest to the velocities in the discharge mode.



EES Double Suction Riser Quick Reference - Heat Pump Tandem Digital Scroll Compressor Continued:

Saturated Condensing Temperature

SCT = 110.0 [F]   SCP = 365.4 [psig]

Condenser Subcooling

Subcool = 15.00 [F]

Saturated Suction Temperature

SST = 35.34 [F]   SSP = 108.05 [psig]

Suction Super heat

SH = 10.00 [F]

Refrigerant$= R410A

Tons = 12.04 [tons]

$m_{ref,s}$ = 0.544 [lb/sec]

$m_{ref,per,min,s}$ = 32.65 [lb/min]

$m_{ref,per,hr}$ = 1959 [lb/hr]

Discharge

Choose Discharge HGB

1-3/8 od

id_d = 1.245 [in.]

$vel_{fpm,d}$ = 733 [ft/min]

dt_d = 0.2178 [F]

Elbow and Equivalent Length

$Quantity_d$ = 0

$Length_d$ = 100 [ft.]

Le_d = 100 [ft.]

$Pd_{psia,d}$ = 1.094 [psi]

$MinPDoilReturn_d$ = 0.2364 [psi]

$MinTonsOilReturn_d$ = 5.254 [tons]

$vel_{fpmMin,d}$ = 319.9 [ft/min]

$lbs_{ref,d}$ = 4.454

$PipeVolume_d$ = 0.8454 [ft^3]

Discharge

Choose Discharge HGB

7/8 od

id_d = 0.785 [in.]

$vel_{fpm,d}$ = 1844 [ft/min]

dt_d = 2.138 [F]

Elbow and Equivalent Length

$Quantity_d$ = 0

$Length_d$ = 100 [ft.]

Le_d = 100 [ft.]

$Pd_{psia,d}$ = 10.75 [psi]

$MinPDoilReturn_d$ = 0.2843 [psi]

$MinTonsOilReturn_d$ = 1.706 [tons]

$vel_{fpmMin,d}$ = 261.2 [ft/min]

$lbs_{ref,d}$ = 1.771

$PipeVolume_d$ = 0.3361 [ft^3]

$PipeRefrigerantLiquid_d$ = 20.04 [lbs]

The 1-3/8"OD is the closest cross-sectional area to the combined lines and since discharge is down, the velocity is greater than the minimum velocity for oil return so this would be an acceptable solution.

The 7/8"OD is the smaller of the two lines. Divide velocity by 2 (for one compressor off), the velocity in this line is 922 fpm at 50% capacity which exceeds the minimum velocity for oil return.

32 General Control Sequences

32.1 Constant Air Volume (CAV)

The airflow rate through this air handling unit remains constant. CAV units typically use space temperature control, which means the decision to cool, heat, or vent is made by comparing the measured space temperature to the space temperature setpoint. Once a mode of operation is energized by the controller, the cooling or heating is controlled to a supply air setpoint.

32.2 Variable Air Volume (VAV)

The airflow rate through this air handling unit varies to maintain a constant supply duct static pressure in the main duct. The cooling only VAV unit typically supplies 55°F supply air temperature to many different zones in the building. The different zones have field supplied VAV boxes that control how much air enters the space based on space temperature control. If the VAV unit includes a heating source, the heat can be operated to provide morning warm up or to provide supply air tempering. The VAV boxes provided by others may have a heating source if temperatures higher than 55°F are desired.

32.3 Single Zone Variable Air Volume (SZ VAV)

Single Zone VAV systems modulate the supply fan speed to maintain a comfortable and consistent space temperature in a single zone. The variable capacity compressors modulate to maintain a constant supply air temperature, and if the unit also has a modulating heat source, the heat source modulates to maintain a constant supply air temperature. The space temperature sensor determines the cooling, heating, or venting mode of operation. If the heat source is not modulating, the heat mode will operate as a CAV with 100% supply fan speed.

32.4 Make Up Air (MUA)

The Make Up Air unit uses 100% outside air with a constant airflow rate. MUA units use outside air temperature control, which means the decision to cool, heat, or vent is made by comparing the measured outside air temperature to the outside air temperature setpoint. Once a mode of operation is energized by the controller, the cooling or heating is controlled to a supply air setpoint.

32.5 CAV/MUA Dual Mode

CAV/MUA Dual Mode is also known as Hood On application. It is used for a lab with exhaust hood or kitchen applications with exhaust hood. The unit is CAV controlled until the exhaust hoods come on. When the exhaust hoods come on, the OA damper goes full open to bring outside air into the space and uses the OA sensor as the controlling sensor. When the hood is off, the controlling sensor will be the space sensor using a CAV control sequence.

32.6 Dehumidification Mode

In all the control sequences previously discussed, if the unit also needs to provide humidity control, it will read a humidity sensor as the controlling sensor. The controlling sensor for CAV, VAV, & single zone VAV will be a space humidity sensor, and the controlling sensor for MUA will be an outside air humidity sensor. The compressors are controlled to a saturation suction temperature setpoint and the modulating hot gas reheat valves are controlled to a supply air temperature setpoint.

33 EER, SEER, IEER

Question: Will looking at an EER, SEER, or IEER comparison between AAON equipment and competitor equipment show all of the energy saving benefits of AAON equipment?

Answer: EER, SEER and IEER are all single numbers representing efficiency, each number represents a different calculation of efficiency and therefore are not comparable between differing metrics because there is not a direct correlation.

The **Energy Efficiency Ratio (EER)** is a ratio of the cooling capacity in Btu/h to the power input value in Watts at a SINGLE set of Rating Conditions expressed in Btu/(W·h). In the EER test the air conditions entering the indoor coil are a SINGLE value of 80/67°F and a SINGLE outdoor condition of 95/75°F.

The **Seasonal Energy Efficiency Ratio (SEER)** is the total heat removed from the conditioned space during an annual cooling SEASON, expressed in Btu's, divided by the total electrical energy consumed by the air conditioner or heat pump during the same season, expressed in Watt-hours. In AHRI Standard 210/240 one can see that the air conditions entering the indoor coil stay at 80/67°F for all test points and use a maximum of three fan speeds. The outdoor coil conditions change between 92°F and 67°F DB.

The **Integrated Energy Efficiency Ratio (IEER)** is a single number representing cooling part-load efficiency weighted at 100%, 75%, 50%, and 25% capacity. In AHRI Standard 340/360 one can see that the air conditions entering the indoor coil stay at 80/67°F for all test points, but if a unit is capable of VAV operation it is considered in the test, in that, VAV units adjust the airflow rate at part load to maintain the full load measured leaving air dry-bulb temperature. The outdoor coil condition changes based on the initial Full Load Unit Net Capacity.

Equipment can be designed for very high peak efficiency (EER) and can have very poor part load efficiency (SEER/IEER). A car with a single gear is designed for high speeds or low speeds but not both. HVAC units with a single on/off compressor and on/off fan are designed for optimum performance at a single point and operate at less than peak efficiency at all other points. The EER measurement is an energy metric comparison of the past.

By introducing variable capacity compressors and variable speed indoor and outdoor fans, AAON equipment is similar to an automatic transmission, where the equipment can operate efficiently at many speeds (loads). The part load standards of SEER and IEER still only test at a few points, so the true benefit of fuel (energy consumption) efficiency is not shown in the single number. The standards are progressing but they lag the industry, not lead it, because the standards are developed via committees dominated by the large players in the market, that are slow to adopt new technologies, not the smaller, innovative companies like AAON.

Just as the MPG on the car window is an estimate of driving conditions and varies based on highway conditions, driver style, etc. SEER and IEER are a limited in the representation of a HVAC system because they do not consider dehumidification/re-evaporation, operation at conditions other than the points in the standard or optimized sequences like Single Zone VAV that varies the fan speed based on space temperature and the compressor capacity based on supply air temperature.

The best comparison between differing technologies is not an AHRI standard with limited testing points, but is an energy analysis that shows total power used between the alternate designs over a given time period. The AAON Energy and Economics Analysis Program (found in the ECat software) can and should be used for this purpose.

34 Appendix

Table 13 - R-410A Refrigerant Temperature-Pressure Chart
Based on a Standard Atmosphere at Sea Level (14.696 psia)

Psia = 14.696 + Psig

°F	PSIG	°F	PSIG	°F	PSIG	°F	PSIG	°F	PSIG
20	78.3	**47**	134.7	**74**	213.7	**101**	321.0	**128**	463.2
21	80.0	**48**	137.2	**75**	217.1	**102**	325.6	**129**	469.3
22	81.8	**49**	139.7	**76**	220.6	**103**	330.2	**130**	475.4
23	83.6	**50**	142.2	**77**	224.1	**104**	334.9	**131**	481.6
24	85.4	**51**	144.8	**78**	227.7	**105**	339.6	**132**	487.8
25	87.2	**52**	147.4	**79**	231.3	**106**	344.4	**133**	494.1
26	89.1	**53**	150.1	**80**	234.9	**107**	349.3	**134**	500.5
27	91.0	**54**	152.8	**81**	238.6	**108**	354.2	**135**	506.9
28	92.9	**55**	155.5	**82**	242.3	**109**	359.1	**136**	513.4
29	94.9	**56**	158.2	**83**	246.0	**110**	364.1	**137**	520.0
30	96.8	**57**	161.0	**84**	249.8	**111**	369.1	**138**	526.6
31	98.8	**58**	163.8	**85**	253.7	**112**	374.2	**139**	533.3
32	100.9	**59**	166.7	**86**	257.5	**113**	379.4	**140**	540.1
33	102.9	**60**	169.6	**87**	261.4	**114**	384.6	**141**	547.0
34	105.0	**61**	172.5	**88**	265.4	**115**	389.9	**142**	553.9
35	107.1	**62**	175.4	**89**	269.4	**116**	395.2	**143**	560.9
36	109.2	**63**	178.4	**90**	273.5	**117**	400.5	**144**	567.9
37	111.4	**64**	181.5	**91**	277.6	**118**	405.9	**145**	575.1
38	113.6	**65**	184.5	**92**	281.7	**119**	411.4	**146**	582.3
39	115.8	**66**	187.6	**93**	285.9	**120**	416.9	**147**	589.6
40	118.1	**67**	190.7	**94**	290.1	**121**	422.5	**148**	596.9
41	120.3	**68**	193.9	**95**	294.4	**122**	428.2	**149**	604.4
42	122.7	**69**	197.1	**96**	298.7	**123**	433.9	**150**	611.9
43	125.0	**70**	200.4	**97**	303.0	**124**	439.6		
44	127.4	**71**	203.6	**98**	307.5	**125**	445.4		
45	129.8	**72**	207.0	**99**	311.9	**126**	451.3		
46	132.2	**73**	210.3	**100**	316.4	**127**	457.3		

To use this table at altitude follow the example below:
Denver Colorado Airport is 5430 ft above sea level, the standard barometric pressure is 12.032 psia
A gage zeroed at denver will be off by (14.696-12.032)= 2.664 psia. To use the above chart at altitude
Adjust the gage value as follows:

Table Psig Lookup = Measured gage pressure + (Pb sea level – Pb altitude)
Gage reads 115.4 psig altitude
Table Psig Lookup = 115.4 + 2.664 = 118.064 rounds to 118.1 psig (40°F)

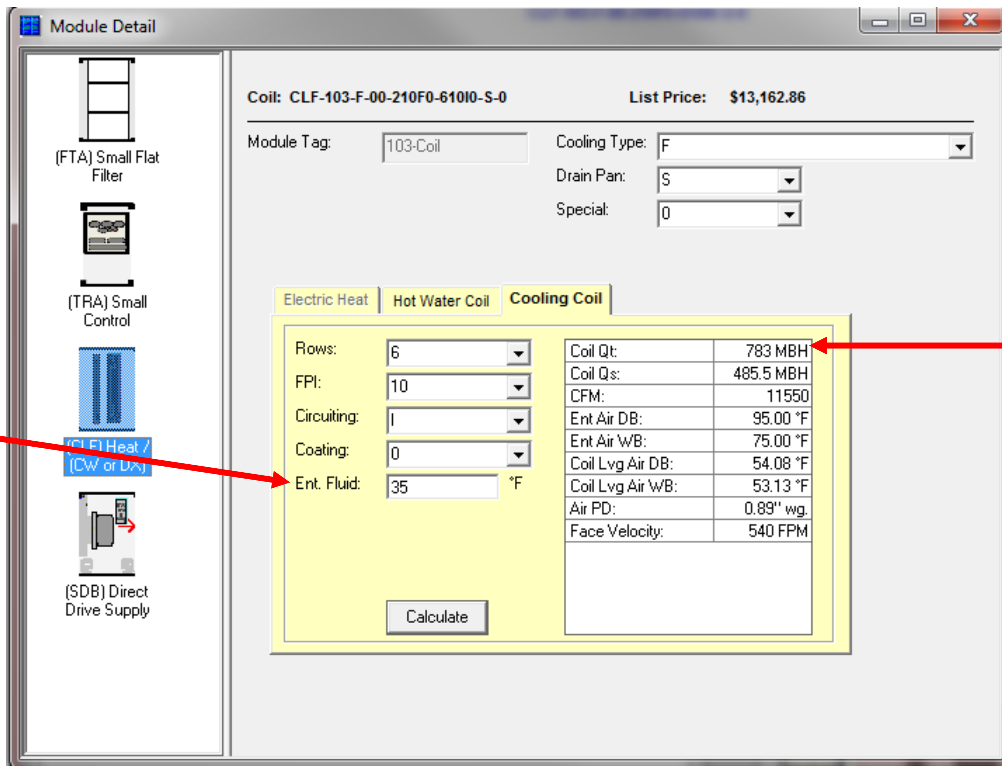
Table 14 - Equivalent Length Table

Equivalent Length Table											
OD Nominal	ID	Le LR Elbow	Le Short Radius Elbow	Le 45 deg Elbow	Close Pattern 180 deg Bend	Le Tee Straight Thru	Le Tee Flow Thru Branch	Le Ball Valve Unrestricted	Le Sight Glass	Le Globe Valve Conventional	Le Y Pattern Globe Valve
		20.0	30.0	16.0	50.0	20.0	60.0	3.15	16.8	340.0	145.0
		L/D = 20	L/D = 30	L/D = 16	L/D = 50	L/D=20	L/D =60	L/D = 3.15	L/D = 16.8	L/D = 340	L/D = 145
Source	Copper Handbook	L/D from Crane Handbook						ChE Calculator	ASHRAE expansion + contraction + expansion 3/4	L/D from Crane Handbook	
Inch	Inch	feet	feet	feet	feet	feet	feet	feet	feet	feet	feet
1/8 od	0.065	0.1	0.2	0.1	0.3	0.1	0.3	0.02	0.1	1.8	0.8
3/8 od	0.315	0.5	0.8	0.4	1.3	0.5	1.6	0.1	0.4	8.9	3.8
1/2 od	0.430	0.7	1.1	0.6	1.8	0.7	2.2	0.1	0.6	12.2	5.2
5/8 od	0.545	0.9	1.4	0.7	2.3	0.9	2.7	0.1	0.8	15.4	6.6
3/4 od	0.666	1.1	1.7	0.9	2.8	1.1	3.3	0.2	0.9	18.9	8.0
7/8 od	0.785	1.3	2.0	1.0	3.3	1.3	3.9	0.2	1.1	22.2	9.5
1-1/8 od	1.025	1.7	2.6	1.4	4.3	1.7	5.1	0.3	1.4	29.0	12.4
1-3/8 od	1.265	2.1	3.2	1.7	5.3	2.1	6.3	0.3	1.8	35.8	15.3
1-5/8 od	1.505	2.5	3.8	2.0	6.3	2.5	7.5	0.4	2.1	42.6	18.2
2-1/8 od	1.985	3.3	5.0	2.6	8.3	3.3	9.9	0.5	2.8	56.2	24.0
2-5/8 od	2.465	4.1	6.2	3.3	10.3	4.1	12.3	0.6	3.5	69.8	29.8
3-1/8 od	2.945	4.9	7.4	3.9	12.3	4.9	14.7	0.8	4.1	83.4	35.6
3-5/8 od	3.425	5.7	8.6	4.6	14.3	5.7	17.1	0.9	4.8	97.0	41.4
4-1/8 od	3.905	6.5	9.8	5.2	16.3	6.5	19.5	1.0	5.5	110.6	47.2

34.1 Cross Plot Example

To get the total heat absorptions from the M2 air handler:

1. Select the M2 unit
2. Enter the unit conditions
3. Double-click on the coil module
4. Select the desired coil
5. Change the Entering Fluid temperature to 35°F
6. Record the Coil Qt
7. Repeat steps 5 & 6 for fluid temps of 40°F, 45°F, & 50°F



To get the total absorptions for the condensing unit:

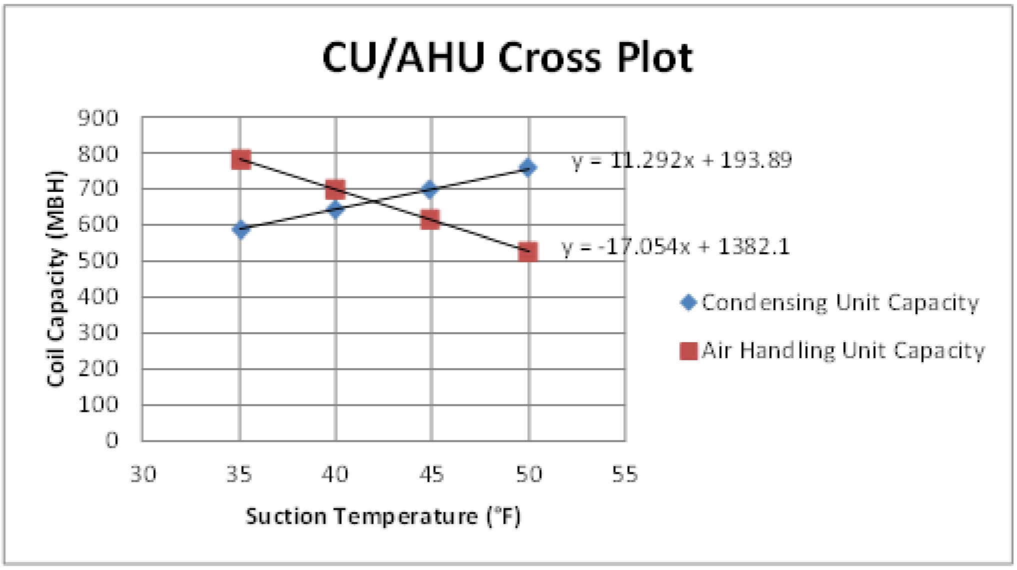
8. First determine which size CN is required
9. To get a rough estimate, divide the Coil Qt by 12MBH. So this example would be 783 / 12 = 65.3 tons
10. Select a CNA-065 in ECat and print the Unit Rating Page
11. Find the Condenser Capacity section and find the suction temperatures of 35°F, 40°F, 45°F, 50°F matched with heat absorptions.

Capacity (MBH)

Suction Temp	Total Unit	Circuit 1	Circuit 2
Design (50°)	759.9	412.4	347.5
35°	590.6	321.6	269.1
40°	644.0	350.3	293.7
45°	700.7	380.7	320.0
50°	759.9	412.4	347.5

12. Now Plot the Suction Temperature (°F) vs Qt (MBH).
13. There will be one line for the air handler & one line for the condenser.
14. The two lines should cross somewhere between 35°F & 50°F. If they do not cross, then the CN and M2 are not compatible.

Suction Temp °F	Condensing Unit CN Capacity MBH	Air Handling Unit DX Capacity MBH
35	590.6	783.0
40	644.0	702.2
45	700.7	616.6
50	759.9	527.3



When both equations equal the same coil capacity, the system is balanced
Set the equations equal to each other and solve for X
X = suction temperature input for M2 program

$11.292x + 193.89 = -17.054x + 1382.1$

$x = 41.9$

35 Index

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AAON
2425 South Yukon Ave.
Tulsa, OK 74107-2728
Phone: 918-583-2266
Fax: 918-583-6094
www.aaon.com

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