



System Considerations for Refrigerant Blends with Temperature Glide

Theory and Practical Retrofit Guidance



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System Considerations for Refrigerant Blends with Temperature Glide

Theory and Practical Retrofit Guidance for Low and Medium Temperature Refrigeration

1

BLEND AND GLIDE BASICS

Refrigerant choices for refrigeration systems are undergoing significant change. The phase-out of R22 is well into its final few years. Severely restricted now, production and importation of new R22 in the US will cease as of 2020. EPA Significant New Alternatives Policy (SNAP) Rule 20, issued in July 2015 also sets a timeline for de-listing R404A, R507, and others as acceptable alternatives for many commercial refrigeration applications. Some of the first de-listings concerning system retrofits took effect July 20, 2016. EPA has also approved (or SNAP listed) several alternative HFC and HFO-blend refrigerants that may be used to retrofit existing systems, as well as for new systems. Most of these blends are viable replacements for R22, R404A, and similar refrigerants, but have properties that require careful consideration to ensure a successful retrofit.

1.1 SUPERHEAT AND SUBCOOLING OF REFRIGERANT BLENDS

The refrigeration industry has been using refrigerant blends for many years. Some of these are azeotropic, in which the blend constituents form a mixture that changes phase at a constant temperature at a given pressure. This is the same behavior as a pure refrigerant, made up of only one type of molecule. In the evaporator, as the refrigerant boils off from a saturated liquid to a saturated vapor, no sensible heat is gained by the refrigerant until it is 100% vapor. This is a desirable characteristic for most refrigeration systems. Pure refrigerants and azeotropes have a single saturation temperature at a given pressure. Superheat and subcooling are simply calculated from this single-point relationship. Azeotropic blends are assigned ASHRAE R-numbers in the 500-series, such as R502 or R507.

Other blends are zeotropic, in which the blend constituents change temperature with change of phase at constant pressure. This behavior is commonly called temperature glide. As a zeotrope boils off in the evaporator, the constituent with the highest vapor pressure will evaporate first. While the first constituent is evaporating, the remainder of the constituents are gaining sensible heat. As the temperature of the refrigerant increases, the vapor pressure of the next constituent will then be reached, and it will evaporate, while the remaining constituents (including those already evaporated) will continue to gain sensible heat. This will continue for all constituents, until all are boiled off. At a constant pressure, the saturated vapor of the blend will be at a higher temperature than the saturated liquid, thus the term “temperature glide”. This also means that at a given pressure, there are now two saturation temperatures to consider: saturated liquid temperature (also known as “bubble point”) and saturated vapor temperature (also known as “dew point”). Because of this behavior, superheat and subcooling for zeotropes must be calculated using the proper point. Zeotropic blends are assigned R-numbers in the 400-series, such as R404A, R407C, R410A, etc.



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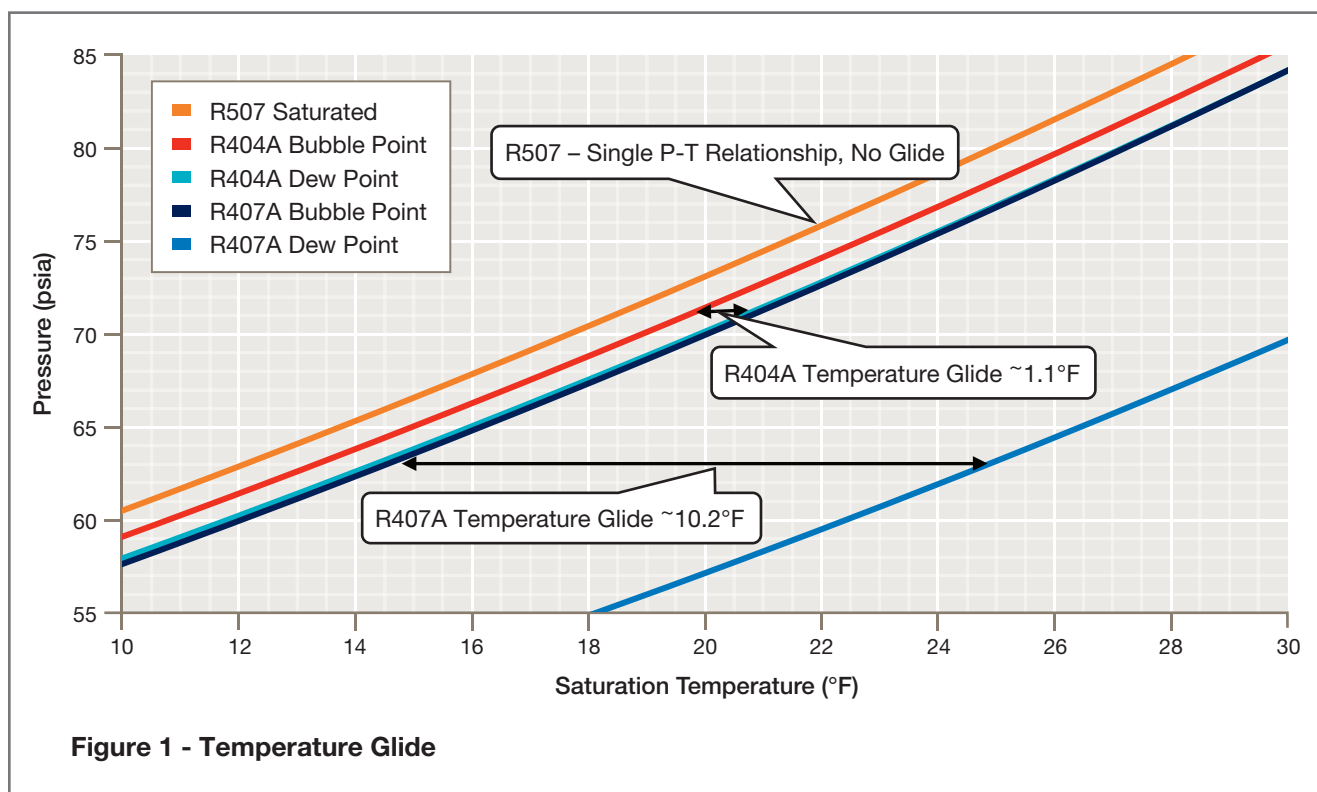
Because having little or no glide is favorable for heat exchanger design, zeotropes with low glide have been more widely used in the past. In the case of blends such as R404A or R408A that have only around 1°F temperature glide, there is very little effect on heat exchanger performance. Even ignoring glide introduces only a small error into superheat and subcooling calculation. With the current movement toward more environmentally-friendly refrigerants, many blends with significantly higher temperature glide (on the order of 10°F) are commonly being used, and more are being introduced to the market. These include but are not limited to R407A, R407C, R407E, R448A, and R449A. With high-glide zeotropic refrigerants, care must be taken to use bubble point to calculate subcooling, and dew point to calculate superheat.

Saturation pressure temperature relationships for R507 (an azeotrope), R404A (a low-glide zeotrope) and R407A (a high-glide zeotrope) are shown below. This is a useful way to visualize temperature glide, and some of the differences between these refrigerant types. The focus is on typical medium temperature refrigeration evaporator range, from 10 to 30°F.

On this plot, temperature glide is the horizontal distance (meaning at fixed pressure) between the bubble point and dew point lines of the zeotropic blends. Azeotropic R507 has a single saturation line, and has no glide. R404A has just over 1°F glide, while R407A has over 10°F temperature glide.

1.2 TEMPERATURE GLIDE AND THE PRESSURE-ENTHALPY DIAGRAM

Temperature glide and subcooling and superheat calculations can also be visualized on a pressure-enthalpy (P-h) diagram. Figure 2 is a P-h diagram for R407A, and illustrates the refrigeration cycle and proper calculations for a high-glide refrigerant. As with all P-h diagrams, the liquid saturation line forms the left side of the saturation dome, and the vapor saturation line forms the right side of the dome. The lines of constant temperature (or isotherms) slope downward left-to-right in the saturated region, showing that R407A is a zeotrope, having temperature glide. For a pure refrigerant or azeotrope, these lines would have no slope: no change in temperature at a given pressure during phase change. Subcooling and superheat are always calculated from the nearest saturation state.



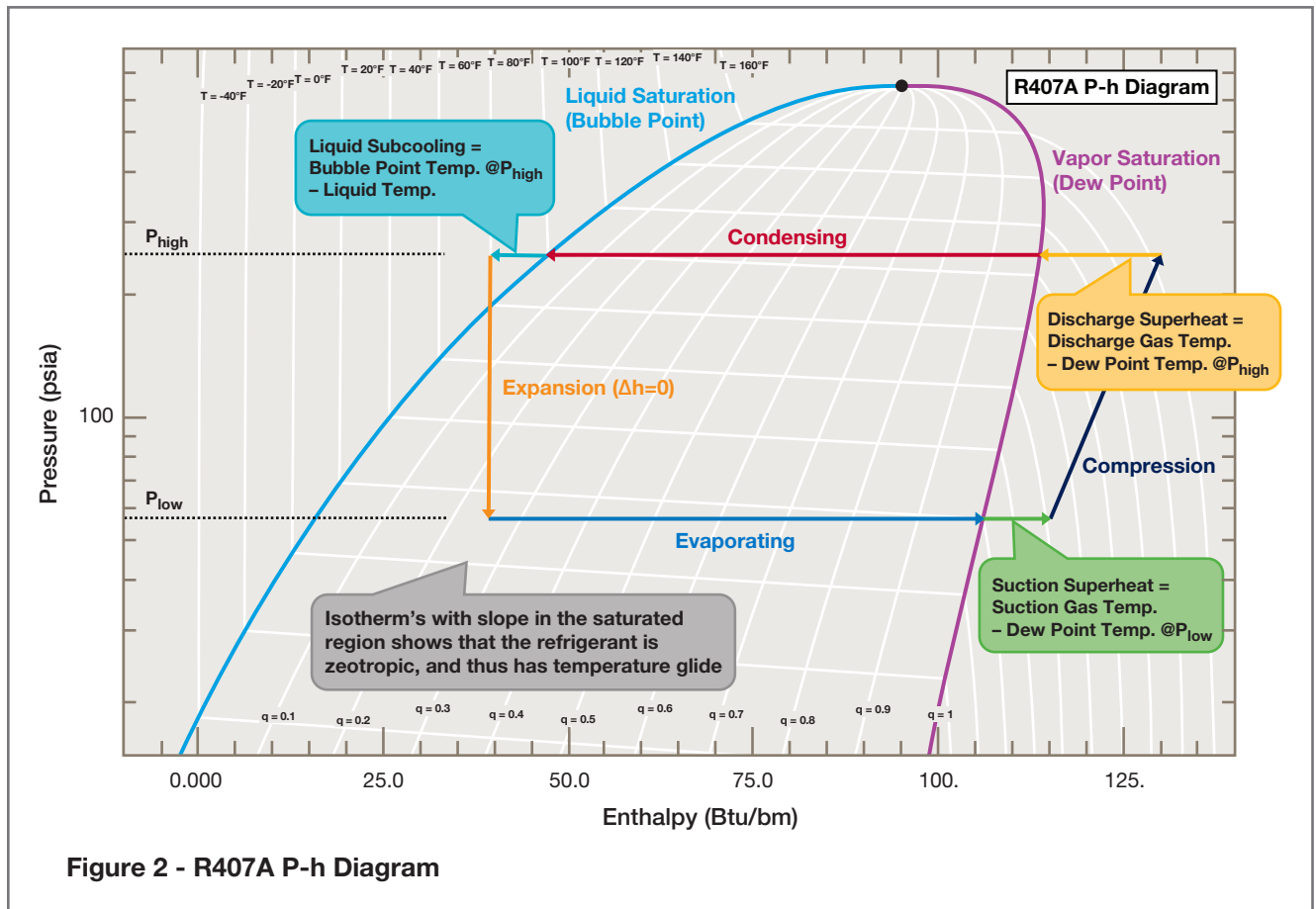


Figure 2 depicts a R407A refrigeration system operating at medium temperature conditions. The measured high and low side system pressures are $P_{high} = 252$ psia and $P_{low} = 57$ psia. Using the proper saturation states:

Liquid Subcooling Calculation:

Measured liquid line temperature = 80°F
 From NIST Refprop, R407A **bubble point** @ 252 psia = 100°F
 Liquid Subcooling = $100^\circ\text{F} - 80^\circ\text{F} = 20^\circ\text{F}$

Suction Superheat Calculation:

Measured suction gas into the compressor = 65°F
 From NIST Refprop, R407A **dew point** @ 57 psia = 20°F
 Suction Superheat = $65^\circ\text{F} - 20^\circ\text{F} = 45^\circ\text{F}$

Discharge Superheat Calculation:

Measured compressor discharge gas = 165°F
 From NIST Refprop, R407A **dew point** @ 252 psia = 108°F
 Discharge Superheat = $165^\circ\text{F} - 108^\circ\text{F} = 57^\circ\text{F}$

The calculated numbers can be checked with the P-h diagram, by scaling them with the isotherms. In the case of liquid subcooling, that process extends from the bubble point, leftward into the subcooled liquid region. It extends from the 100°F isotherm to the 80°F isotherm, confirming our calculation of 20°F subcooling.

To show the effect of using the incorrect saturation state, consider Suction Superheat calculated using bubble point:

Measured suction gas into the compressor = 65°F

From NIST Refprop, R407A **bubble point** @ 57 psia = 10°F

Suction Superheat = 65°F – 10°F = **55°F**

An additional 10°F superheat would be calculated that does not actually exist. This amount of error in calculating superheat at the outlet of an evaporator coil would lead to an improperly adjusted TEV and liquid flooding in the suction line, possibly all the way back to the compressor. We also readily see this is incorrect by checking our calculation against the P-h diagram – the closest saturation state is dew point, not bubble point.

1.3 SUPERHEAT CONTROL WITH GLIDE

The thermostatic expansion valve (TEV) has only one function in the system: to meter the proper amount of refrigerant to the evaporator coil by controlling superheat at the bulb location. In order to do this, the TEV is located at the coil inlet. However, conditions at the coil outlet indicate whether the TEV is working properly. For zeotropic refrigerants, superheat is **calculated** based on dew point, the closest saturation condition at the evaporator outlet. It follows naturally that superheat is also **controlled** based on this condition. This means the vapor pressure or dew point of the refrigerant is used to control the TEV. The following diagram and explanation of the fundamental operation of TEVs is excerpted from **Sporlan Bulletin 10-9, Thermostatic Expansion Valves: Theory of Operation, Application, and Selection**.

In order to understand the principles of thermostatic expansion valve operation, a review of its major components is necessary. A sensing bulb is connected to the TEV by a length of capillary tubing which transmits bulb pressure to the top of the valve's diaphragm. The sensing bulb, capillary tubing, and diaphragm assembly is referred to as the thermostatic element. The thermostatic element on all standard Sporlan TEVs is replaceable.

The diaphragm is the actuating member of the valve. Its motion is transmitted to the pin and pin carrier assembly by means of one or two pushrods, allowing the pin to move in and out of the valve port. The superheat spring is located under the pin carrier, and a spring guide sets it in place. On externally adjustable valves, an external valve adjustment permits the spring pressure to be altered.

There are three fundamental pressures acting on the valve's diaphragm which affect its operation: sensing bulb pressure P1, equalizer pressure P2, and equivalent spring pressure P3 (see Figure 1). The sensing bulb pressure is a function of the temperature of the thermostatic charge, i.e., the substance within the bulb. This pressure acts on the top of the valve diaphragm causing the valve to move to a more open position. The equalizer and spring pressures act together underneath the diaphragm causing the valve to move to a more closed position. During normal valve operation, the sensing bulb pressure must equal the equalizer pressure plus the spring pressure, i.e.:

$$P1 = P2 + P3$$

Equivalent spring pressure is defined as the spring force divided by the effective area of the diaphragm. The effective area of the diaphragm is simply the portion of the total diaphragm area which is effectively used by the bulb and equalizer pressures to provide their respective opening and closing forces. Equivalent spring pressure is essentially constant once the valve has been adjusted to the desired superheat. As a result, the TEV functions by controlling the difference between bulb and equalizer pressures by the amount of the spring pressure.

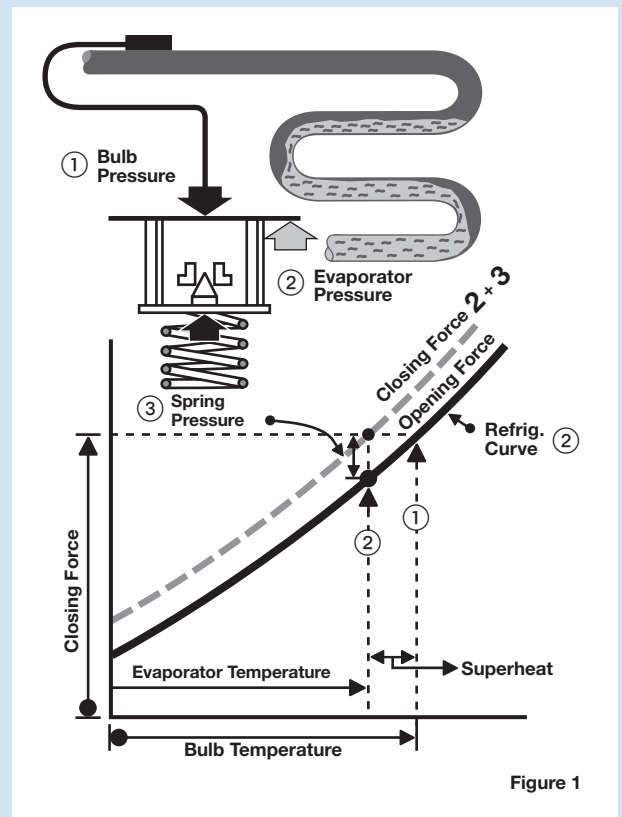


Figure 1

Based on the characteristics of high-glide zeotropic blends, we see the evaporator pressure referred to is in fact vapor pressure or dew point. Considering the three pressures acting on a TEV, P1 (bulb pressure) and P2 (vapor pressure) are acting on opposite sides of the same diaphragm. TEVs are designed for these two pressures to be very close to one another. These will then be the fundamental forces acting to position the pin in the port, producing the best superheat response. Since each of these two similar pressures is acting on the same area, only a small amount of additional force (via spring pressure, P3) is needed to fine tune the balance through adjustment. This is illustrated in the above figure by the small size of P3 in comparison to the other two forces. The stroke of the valve is relatively very small, and the force the spring needs to provide is very small, so the spring constant is low. The adjustment assembly is typically designed to provide 8-12 full rotations from minimum to maximum design spring force. This allows for virtually infinite tuning, but only within the design range of the spring. It follows that the thermostatic element charge should be designed to be close to the vapor pressure (dew point) of a zeotropic refrigerant to provide the best TEV performance.

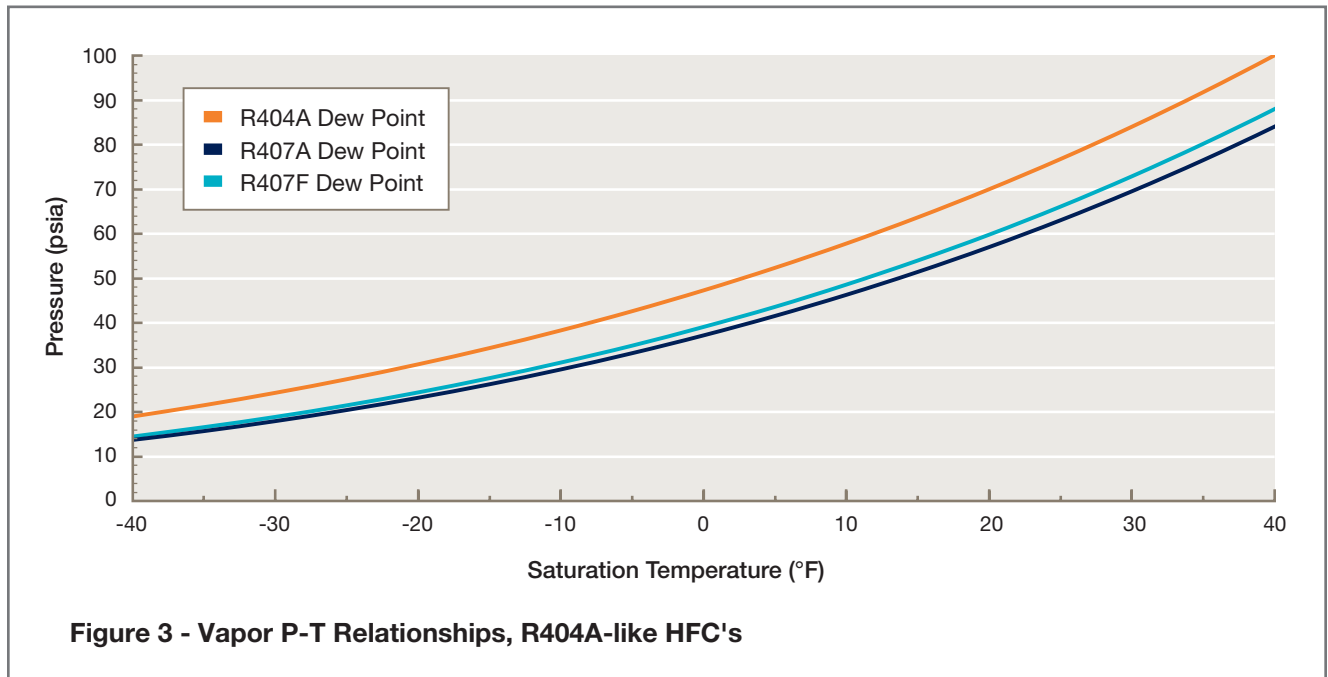
What if the thermostatic element charge is significantly lower in pressure than the vapor pressure at the evaporator outlet? The opening force on the TEV will be very low, and the closing forces will keep the pin and port closed until the bulb gets much warmer, resulting in inadequate refrigerant feed to the evaporator and high superheat. Adjusting spring pressure lower will help compensate by balancing the forces at a more open position. If evaporator outlet pressure is too much higher, even removing all spring pressure may not be sufficient. The TEV will be operating outside its design parameters, desired superheat may not be obtained, and the evaporator will be fed too little refrigerant for its load.

What if the thermostatic element charge is significantly higher in pressure than the vapor pressure at the evaporator outlet? The opening force on the TEV will be very high, moving the pin and port more open, and flooding will occur. Adjusting spring pressure higher will help compensate by balancing the forces at a less open position. However, if too much additional spring pressure is needed, maximum design spring force may be reached, and the valve will be operating outside its design parameters. Proper superheat adjustment may not be possible, and superheat response will be compromised. Liquid flood back may occur, particularly at high condensing pressure conditions. TEVs, like most control systems, operate best in the middle portion of their design range.

2

THERMOSTATIC CHARGES AND CAPACITY FOR ALTERNATE REFRIGERANTS

Many alternates to R404A, R507, and similar refrigerants have been developed. Most of these can also be used as alternates to R22 in refrigeration applications. At this time, the alternates that are generating the greatest interest and gaining market momentum include R407A, R407F, R448A, and R449A. R407A and R407F are HFC blends containing R32, R125, and R134a. R448A and R449A are blends of HFC and HFO constituents, containing R32, R125, R134a, and R1234yf. R448A also contains a small percentage of R1234ze. These alternate refrigerants all have similar temperature glide, in the range of 10°F, due to the common constituents among them. Section 2 examines how these alternates



compare to R404A and R22 from the standpoint of TEV thermostatic charges and volumetric capacity.

2.1 THERMOSTATIC CHARGES FOR HIGH-GLIDE ALTERNATES

Let's relate the design and operational characteristics of TEVs to some of the high-glide refrigerants. Over the past few decades, many different refrigerants have been used. Sporlan has developed thermostatic charges to function properly with most of these. Since the charge in the thermostatic element does not function as a heat transfer fluid, we are primarily concerned that its pressure-temperature relationship is close to the vapor pressure-temperature relationship of the system refrigerant.

During the early stage of thermostatic charge development, Sporlan engineers will compare the vapor pressure-temperature relationship of the new refrigerant to that of a refrigerant with existing thermostatic charges. If they are a close match, that is a strong indication that an existing thermostatic charge will also function well with the new refrigerant. Figure 3 compares the vapor pressure-temperature relationships of R407A and R407F with R404A, across the normal ranges of low and medium temperature refrigeration.

The vapor pressures of R407A and R407F are similar to one another, but both are significantly lower than R404A. Is there an existing refrigerant that may be a better match, for which thermostatic charges have already been designed? Figure 4 adds R22 to the same chart.

The saturation curve of R22 is a much closer match to the vapor pressure curves of R407A and R407F. Consider typical low and medium evaporator operating temperatures of -20°F and 20°F:

TEMPERATURE °F	VAPOR PRESSURE (PSIA)			
	R404A	R407A	R407F	R22
-20	30.7	23.2	24.4	24.9
20	70.0	57.0	59.8	57.8

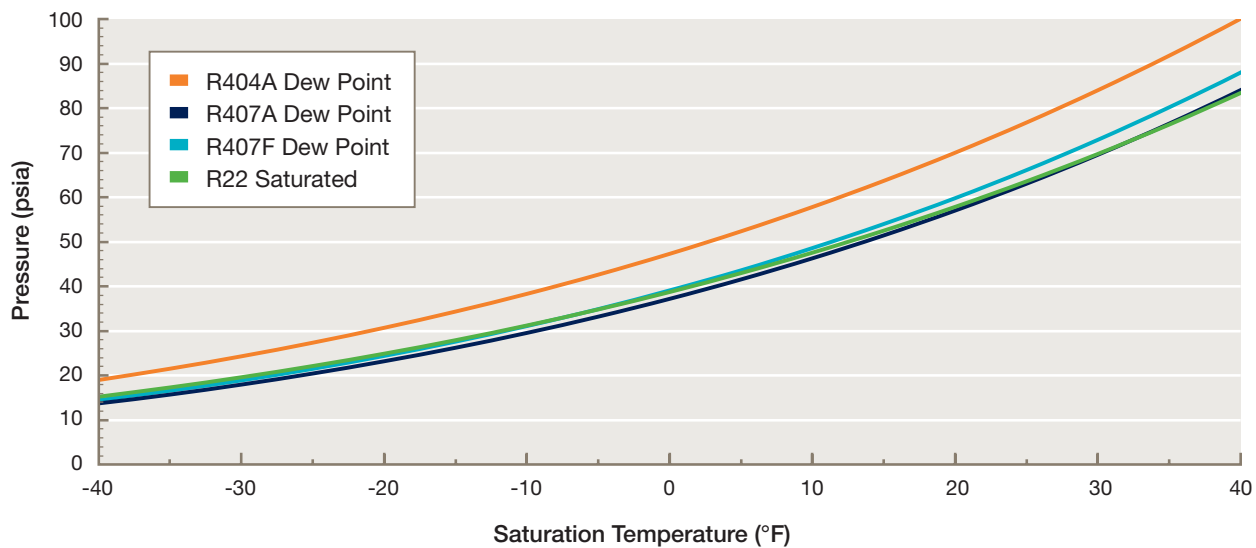


Figure 4 - Vapor P-T Relationships, R404A-like HFC's

R22 saturation pressure varies by less than ± 1 psi at -20°F , and by ± 2 psi at 20°F . This indicates thermostatic charges for R22 will likely be applicable to R407A and R407F. However, the vapor pressure of R404A exceeds those of R407A and R407F on average by 7 psi at -20°F and nearly 12 psi at 20°F .

Figure 5 is the same chart as above, but the vapor pressure curves of R407A and R407F are replaced with those of R448A and R449A. It should be noted that in Figure 5, the dew point curve of R448A lies directly under that of R449A. Though it may appear that it was omitted, these two are instead so similar they cannot be distinguished at the scale presented.

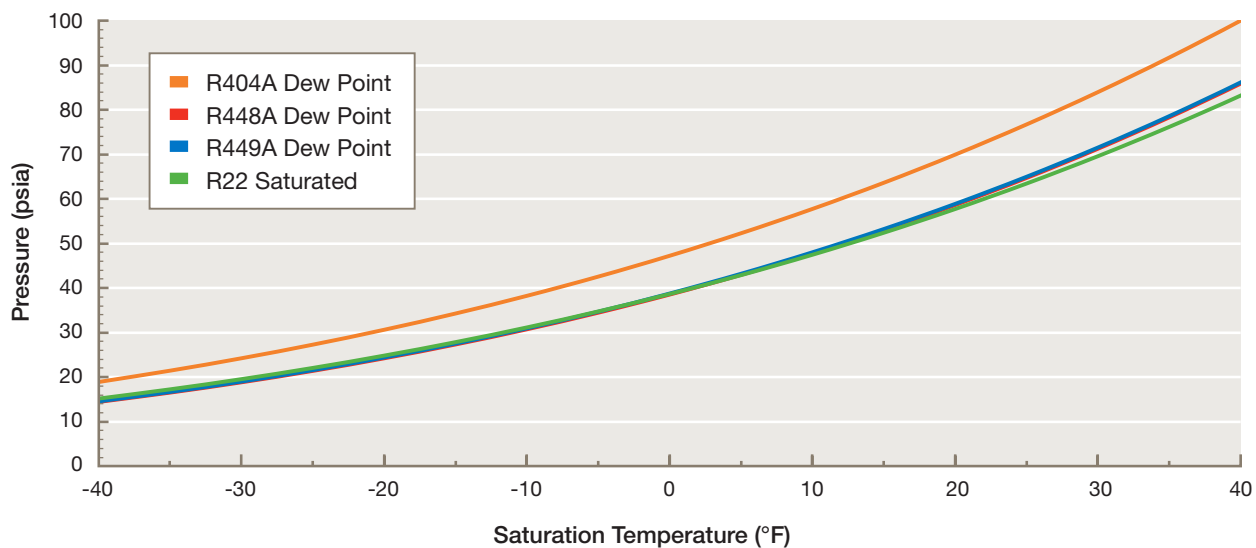


Figure 5 - Vapor P-T Relationships, R404A-like HFO Blends

The saturation curve of R22 appears to be an even closer match to the vapor pressure curves of R448A and R449A. Looking more closely at the -20°F and 20°F points:

TEMPERATURE °F	VAPOR PRESSURE (PSIA)			
	R404A	R448A	R449F	R22
-20	30.7	24.5	24.6	24.9
20	70.0	59.3	59.1	57.8

We find a variation of only +0.4 psi at -20°F, and -1.5 psi at 20°F for R22. R404A vapor pressure exceeds those of R448A and R449A by 6 psi at -20°F and by 11 psi at 20°F. These straightforward comparisons strongly indicate that thermostatic charges designed for R22 will function well for R407A, R407F, R448A, and R449A throughout the range of low and medium temperature refrigeration applications. An extensive program of charge characteristic testing performed in the Sporlan Engineering Lab confirmed both low and medium temperature thermostatic charges for R22 are also the best choice for R407A, R407F, R448A, and R449A.

What is the effect of using thermostatic charges designed for R404A with these refrigerants? We have established that small variations in vapor pressure from one refrigerant to another, around 5% or less, pose no issue with TEV adjustment or operation. What if that variation is on the order of 20%? From the TEV balance equation ($P_1 = P_2 + P_3$), if P_1 becomes much larger than P_2 , P_3 must increase to compensate. However, we also know that:

- P_3 should be kept small so that P_1 and P_2 are the fundamental forces acting to position the pin;
- By adjusting P_3 very high the design range of the spring may be exceeded;
- The TEV will not be operating in its effective design range, where it will provide the best superheat response.

Thermostatic charges designed for refrigerants with much higher vapor pressure, including R404A, should not be used with R407A, R407F, R448A, and R449A. Thermostatic charges designed for R22 are the proper choice. Conversely, lower-pressure thermostatic charges for R22, R407A, R407F, R448A, or R449A should not be applied with higher vapor pressure refrigerants such as R404A.

2.2 CAPACITY OF ALTERNATE REFRIGERANTS

The capacity (Q) of a direct expansion heat exchanger is calculated by the following equation:

$$Q = \dot{m}(h_{out} - h_{in}), \text{ where:}$$

\dot{m} = mass flow rate of the refrigerant;

h_{out} = the enthalpy of the refrigerant leaving the heat exchanger;

h_{in} = the enthalpy of the refrigerant entering the heat exchanger

Refrigeration systems are almost always designed to a specific cooling capacity, so the Q of greatest interest is the capacity of the evaporator. The $(h_{out} - h_{in})$ part of the equation is also known as net refrigerating effect, or NRE. It is closely approximated by subtracting saturated liquid enthalpy into the TEV from saturated vapor enthalpy at the evaporator pressure. Refrigerant choice and high and low side operating conditions determine NRE. So for a given Q , \dot{m} will be determined by the NRE of the refrigerant.

Let's look at the NRE of some of the alternates, using 90°F saturated liquid, and saturated vapor at -20°F and 20°F, calculated by NIST Refprop 9.1. We will compare with R22 and R404A.

EVAPORATOR TEMPERATURE °F	NET REFRIGERATING EFFECT (Btu/lbm)					
	R404A	R407A	R407F	R448A	R449A	R22
-20	44.1	57.5	64.8	59.0	58.2	66.1
20	49.4	62.6	69.7	64.1	63.3	70.1

Using NRE, we will calculate the required mass flow of each refrigerant per ton (12,000 Btu/hr) capacity.

EVAPORATOR TEMPERATURE °F	MASS FLOW PER TON (lbm/min)					
	R404A	R407A	R407F	R448A	R449A	R22
-20	4.54	3.48	3.09	3.39	3.44	3.03
20	4.05	3.19	2.87	3.12	3.16	2.85

There is quite significant variation in mass flow required per ton capacity at the same conditions for these refrigerants. The mass flow of R407F is very close to that of R22, and both are ~32% lower than R404A. The mass flow of R407A, R448A, and R449A are similar, and are ~23% lower than R404A, but still ~12% higher than R22.

It is also useful to compare the volumetric flow of these refrigerants, in order to better understand the sizing of liquid and suction control valves. All valves fundamentally control flow by volume, not mass directly. Density or specific volume is required to calculate volumetric flow. We will use -20°F low temperature saturated evaporating, and 20°F medium temperature saturated evaporating, and 100°F condensing with 10°F subcooling for both. So as not to overly complicate the analysis, high and low side pressure drops will be ignored. From Refprop 9.1, densities for -20°F saturated vapor, 20°F saturated vapor, and 90°F saturated liquid are:

CONDITION °F	REFRIGERANT DENSITY (lbm/ft³)					
	R404A	R407A	R407F	R448A	R449A	R22
-20 Sat. Vapor	0.69	0.47	0.45	0.48	0.48	0.48
20 Sat. Vapor	1.53	1.11	1.06	1.11	1.12	1.07
90 Sat. Liquid	62.92	69.34	67.64	66.42	66.41	72.52

Mass flow per ton is divided by liquid and vapor density to determine the liquid volumetric flow and vapor volumetric flow per ton for each refrigerant, at low and medium temperature system operating conditions.

LOW TEMP (-20/100)	VOLUMETRIC FLOW PER TON (ft³/min)					
	R404A	R407A	R407F	R448A	R449A	R22
Vapor	6.58	7.40	6.87	7.06	7.17	6.31
Liquid	0.072	0.050	0.046	0.051	0.052	0.042

MEDIUM TEMP (20/100)	VOLUMETRIC FLOW PER TON (ft ³ /min)					
	R404A	R407A	R407F	R448A	R449A	R22
Vapor	2.65	2.87	2.71	2.81	2.82	2.66
Liquid	0.064	0.046	0.042	0.047	0.048	0.039

This comparison highlights the differences on the liquid side, and the similarities on suction side of the system. The lower liquid density of R404A in comparison with R22, along with higher mass flow requirements means that an R404A liquid side valve must flow an even greater volume than a simple mass flow comparison of these two would indicate. R407A, R407F, R448A, and R449A are again in the middle, around 30% lower than R404A and 20% higher than R22. On the suction side of the system, the refrigerants are a much closer match. Suction volumetric flow of R404A and R22 are very similar, with R407A, R407F, R448A, and R449A being 4 – 10% higher than both at medium temperature and 9 – 17% higher at low temperature conditions.

3

SYSTEM RETROFITS TO R407A, R407F, R448A, OR R449A: LIQUID SIDE

There is much interest and significant activity with these refrigerants for system retrofits, mostly due to regulatory actions. These include the ongoing phase-out of R22, which will no longer be able to be produced or imported into the US as of January 1, 2020, and the pending removal of R404A and similar refrigerants from EPA's approved list of refrigerants for retrofit supermarket systems (July 20, 2016), new supermarket systems (January 1, 2017), and new condensing units (January 1, 2018). Refrigerant will still be available for service into the foreseeable future, but prices will rise. Blends like R404A present more difficulties with reclaiming and recycling than pure refrigerants such as R22.

When considering retrofits of refrigeration systems, we will assume that one goal is to replace as few system components as possible, in the interest of reducing both cost and system downtime. Section 3 examines how R407A, R407F, R448A, and R449A may function with existing valves. The large differences in mass flow and liquid volumetric flow rate between R404A and R22 noted in Section 2.2 mean that retrofits of each to alternate refrigerants must be addressed separately.

The valve sizing and selections in all the subsequent sections of this document are carried out with Version 5.09.15 of the Sporlan Selection Program. This is proprietary software that employs the NIST Refprop DLL for all refrigerant property calculations. Flow and loading calculations are based on experimentally determined valve characteristics, and results have been experimentally validated. In the following sections, results that are outside the normally accepted range are highlighted in red.

3.1 TEV CAPACITY WHEN RETROFITTING FROM R404A

These refrigerants are mostly used in systems having multi-circuit evaporator coils, which requires the use of a refrigerant distributor. Pressure drop across the distributor reduces the pressure drop across the TEV, so that must be determined before selecting TEVs. The Sporlan selection program will be used to calculate the distributor nozzle and tube pressure drop for

use in selecting TEVs. We will use a 1 ton (12,000 Btu/hr) load, and assume a 3 circuit evaporator with 3/16" OD, 18" long distributor tubes. From this a nozzle is sized for R404A, then that nozzle size is applied to the alternate refrigerants. The selection program will calculate tube ΔP , nozzle ΔP , and distributor total ΔP . Distributor total ΔP is then subtracted from system high-to-low ΔP to obtain ΔP across the TEV, the driving force for flow across the TEV.

LOW TEMP (-20/100)	REFRIGERANT DISTRIBUTOR				
	R404A	R407A	R407F	R448A	R449A
Orifice #	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2
Tube ΔP (psi)	28	22	19	21	22
Nozzle ΔP (psi)	17	13	11	13	13
Dist ΔP (psi)	45	35	30	34	35
Hi-Lo ΔP (psi)	221	229	240	234	230
TEV ΔP (psi)	176	194	210	200	195

MEDIUM TEMP (20/100)	REFRIGERANT DISTRIBUTOR				
	R404A	R407A	R407F	R448A	R449A
Orifice #	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2
Tube ΔP (psi)	14	11	9	10	11
Nozzle ΔP (psi)	23	18	15	17	18
Dist ΔP (psi)	36	28	24	27	29
Hi-Lo ΔP (psi)	182	195	204	199	196
TEV ΔP (psi)	146	167	180	172	167

For a given distributor orifice size, the total pressure drop across it for the alternates is lower by 10 – 15 psi at low temperature conditions, and 8 – 12 psi at medium temperature conditions. While these lower distributor ΔP 's are still adequate for the function of the distributor, there is more ΔP across the TEV for R407A, R407F, R448A, and R449A. Using the R404A distributor ΔP , a TEV is selected for R404A. The same TEV (same port size, pin angle, and stroke) is then applied to the alternate refrigerants, employing their distributor ΔP 's, in order to compare sizing.

LOW TEMP (-20/100)	R404A TEV SELECTION: FBSE-2				
	R404A	R407A	R407F	R448A	R449A
Dist ΔP (psi)	45	35	30	34	35
TEV ΔP (psi)	176	194	210	200	195
TEV % loading	64	44	38	43	44

MEDIUM TEMP (20/100)	R404A TEV SELECTION: FBSE-1				
	R404A	R407A	R407F	R448A	R449A
Dist ΔP (psi)	36	28	24	27	29
TEV ΔP (psi)	146	167	180	172	167
TEV % loading	65	45	40	44	45

From this comparison, we see that a TEV and distributor sized properly for R404A may be oversized for the alternates. Percent loading for R407F reduces approximately 25%, the others reduce approximately 20%. While

there are no hard rules that apply to all TEVs on all systems, loadings below 50% begin to become a concern. Mass and volumetric flow requirements for R407A, R407F, R448A, and R449A are lower, yet there is greater ΔP across the TEV, to drive flow higher. Keep in mind the TEV is sized at a nominal condition. Under low-load conditions, the TEV will be even more oversized. When thermostatic charge is also considered (as discussed in Section 2.1), there is a high risk that existing R404A TEVs will provide poor superheat control when retrofitted to R407A, R407F, R448A, or R449A. There is also a high risk of liquid flood back.

Replacement of TEVs may be the best choice in most retrofits from R404A to R407A, R407F, R448A, or R449A. Most existing distributors will perform acceptably well. Sizing of existing R404A distributors and TEVs should be checked before replacement.

3.2 TEV CAPACITY WHEN RETROFITTING FROM R22

The same exercise can be performed with an R22 system. We will again use a 1 ton (12,000 Btu/hr) load, and assume a 3 circuit evaporator with 3/16" OD, 18" long distributor tubes. A nozzle is sized for R22, then applied to the alternate refrigerants.

LOW TEMP (-20/100)	REFRIGERANT DISTRIBUTOR				
	R22	R407A	R407F	R448A	R449A
Orifice #	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2
Tube ΔP (psi)	17	22	19	21	21
Nozzle ΔP (psi)	21	27	24	27	28
Dist ΔP (psi)	38	49	43	48	49
Hi-Lo ΔP (psi)	185	229	240	234	230
TEV ΔP (psi)	148	180	197	186	181

MEDIUM TEMP (20/100)	REFRIGERANT DISTRIBUTOR				
	R22	R407A	R407F	R448A	R449A
Orifice #	1	1	1	1	1
Tube ΔP (psi)	8	11	9	10	11
Nozzle ΔP (psi)	23	30	26	30	30
Dist ΔP (psi)	31	41	35	40	41
Hi-Lo ΔP (psi)	153	195	204	199	196
TEV ΔP (psi)	122	154	169	159	155

With a distributor orifice sized for R22, total pressure drop for the alternates is higher by 4 - 11 psi at both low and medium temperature conditions. Yet because of their significantly greater total high-to-low side ΔP , there is still as much as 47 - 49 psi greater ΔP across the TEV at both low and medium temperature conditions. Using the R22 distributor ΔP , a TEV is selected for R22. The same TEV (same port size, pin angle, and stroke) is then applied to the alternate refrigerants, in order to compare sizing.

LOW TEMP (-20/100)	R22 TEV SELECTION: FBVE-2				
	R22	R407A	R407F	R448A	R449A
Dist ΔP (psi)	38	49	43	48	49
TEV ΔP (psi)	148	180	197	186	181
TEV % loading	66	70	60	68	69

MEDIUM TEMP (20/100)	R22 TEV SELECTION: FBVE-1				
	R22	R407A	R407F	R448A	R449A
Dist ΔP (psi)	31	41	35	40	41
TEV ΔP (psi)	122	154	169	159	155
TEV % loading	75	76	66	81	83

With a TEV and distributor sized properly for R22, percent loading changes little for the alternates. Percent loading for R407F is somewhat lower at both medium and low temperature conditions, but well within an acceptable range. Loadings for R407A, R448A, and R449A are just 2 – 4% percent higher at low temperature conditions. At medium temperature conditions, loading with R448A and R449A increases by 6 – 8%. All these would be considered within an acceptable range. Coupled with the knowledge that thermostatic charges for R22 are the proper choice for these refrigerants as well, there is a high likelihood that existing R22 TEVs and distributors will provide good superheat control when retrofitted to R407A, R407F, R448A, or R449A. Sizing should be checked to confirm this.

3.3 LIQUID LINE SOLENOID VALVE CAPACITY WHEN RETROFITTING FROM R404A

Another commonly used control valve in refrigeration systems is the solenoid valve. These are most commonly used to control liquid flow, but are also sometimes used to control suction gas flow. We will first look at liquid line solenoid valves. Unlike an expansion valve, the purpose of which is to precisely meter refrigerant to the evaporator, the solenoid valve's purpose is to simply provide electrically actuated on/off refrigerant flow control. Most solenoid valves are “piloted”. They are opened by a coil which applies electromagnetic pull on the stem, but rely on a light spring and a small amount of pressure drop across the valve to close it. At sizing conditions this pressure drop should be 1 psi or greater. If a solenoid valve is too large, there may not be enough ΔP across it to initially close it, particularly at low load conditions.

If a solenoid valve is too small, there will be more pressure drop across it than is desired. This causes loss of subcooling in a liquid line. If there is very little subcooling, liquid can flash across the valve. A general guideline is that the pressure drop should be no greater than 1°F of saturation (bubble point) drop for the system refrigerant. For all the refrigerants in our discussion, 1°F of saturation is 3.0 – 3.5 psi at 100°F.

For TEVs and distributors, we used 1 ton (12,000 Btu/hr) loads in our comparisons. This is appropriate because there is typically one TEV per case or unit cooler in a system. Often one solenoid valve will control flow on a system branch supplying liquid to a few or several cases or unit coolers. For solenoid comparisons, we will size for a two ton (24,000 Btu/hr) load at low temperature conditions (-20°F evaporator, 100°F condensing) and a four ton (48,000 Btu/hr) load at medium temperature conditions (20°F evaporator,

100°F condensing). A liquid line solenoid valve will be selected for R404A, then that valve will be applied to R407A, R407F, R448A, and R449A to see the effect on sizing. Selecting a solenoid valve for R404A that will have ΔP greater than 1 psi, but less than 3.5 psi:

LOW TEMP (-20/100)	R404A LL SOLENOID SELECTION: E6				
	R404A	R407A	R407F	R448A	R449A
Flow (lbm/min)	9.07	6.96	6.17	6.79	6.88
Sol ΔP (psi)	1.3	0.7	0.6	0.7	0.7
Sol % loading	66	48	42	48	49

MEDIUM TEMP (20/100)	R404A LL SOLENOID SELECTION: E8				
	R404A	R407A	R407F	R448A	R449A
Flow (lbm/min)	16.2	12.8	11.6	12.5	12.6
Sol ΔP (psi)	2.6	1.5	1.2	1.4	1.5
Sol % loading	92	68	61	68	69

An E6 liquid line solenoid valve is selected for the R404A low temperature system, with 1.3 psi ΔP . This is on the low end of the preferred range, so what if the next smaller size E5 is selected instead? ΔP would be 4.2 psi instead, with 117% loading. In this case, either would likely function without issue. Usually the lower ΔP would be chosen for an actual system, so the E6 will be used for comparison. As shown in the table above, that choice results in low ΔP for all the alternate refrigerants. The situation is different for the medium temperature system with a four ton load. An E8 solenoid valve is chosen for R404A, with 2.6 psi ΔP . When applied with the alternate refrigerants, ΔP 's from 1.2 to 1.5 psi are calculated. These are obviously lower, but still acceptable.

When retrofitting an existing R404A system to R407A, R407F, R448A, or R449A, liquid line solenoid valve sizing should be checked. If ΔP will be lower than 1 psi with the alternate refrigerant, appropriately sized solenoid valves should be selected to replace the existing ones.

3.4 LIQUID LINE SOLENOID VALVE CAPACITY WHEN RETROFITTING FROM R22

Let's perform the same comparison with R22 as a baseline, again using a two ton (24,000 Btu/hr) load at low temperature conditions and a four ton (48,000 Btu/hr) load at medium temperature conditions. A liquid line solenoid valve will be selected for R22, then that valve will be applied to R407A, R407F, R448A, and R449A to see the effect on sizing. 1°F saturation drop for R22 at 100°F is 2.9 psi, so we select a solenoid valve for R22 that will have ΔP greater than 1 psi but less than 3 psi.

LOW TEMP (-20/100)	R22 LL SOLENOID SELECTION: E5				
	R22	R407A	R407F	R448A	R449A
Flow (lbm/min)	6.05	6.96	6.17	6.79	6.88
Sol ΔP (psi)	1.6	2.2	1.8	2.2	2.3
Sol % loading	79	85	74	84	86

MEDIUM TEMP (20/100)	R22 LL SOLENOID SELECTION: E6				
	R22	R407A	R407F	R448A	R449A
Flow (lbm/min)	11.4	12.8	11.6	12.5	12.6
Sol ΔP (psi)	1.8	2.4	2.0	2.4	2.5
Sol % loading	84	89	79	88	89

An E5 liquid line solenoid valve is selected for the R22 low temperature system, with 1.6 psi ΔP. This would be common choice in an actual system. As shown in the table above, that selection results in ΔP's from 1.8 to 2.3 psi for the alternate refrigerants. The E5 would function well, without excessive ΔP for the alternate refrigerants. Results for the medium temperature system mirror the low temperature system. An E6 solenoid valve is selected for R22, with 1.8 psi ΔP. When applied with the alternate refrigerants, ΔPs of 2.0 to 2.5 psi are calculated, again well within an acceptable range.

When retrofitting an existing R22 system to R407A, R407F, R448A, or R449A, existing liquid line solenoid valve sizing will in most situations be acceptable, though it should be checked. If ΔP will be excessively high with the alternate refrigerant, appropriately sized solenoid valves should be selected to replace the existing ones.

4

SYSTEM RETROFITS TO R407A, R407F, R448A, OR R449A: SUCTION SIDE

Section 2.2 notes the large differences in mass flow between R404A and R22. This coupled with R22's higher liquid density means that on the liquid side of the system, a valve has to flow a much greater volume of R404A than R22 to deliver a given amount of refrigeration capacity. However, the density relationship between R22 and R404A reverses for saturated vapor. R404A vapor is much denser than R22 vapor. Because of this, the volume of vapor that a valve on the suction side of the system must flow to deliver a given capacity is actually quite similar for both. Section 4 will examine how existing mechanical evaporator pressure regulators, electric evaporator pressure regulators, and suction solenoid valves will perform when a system is retrofitted from R22 or R404A to R407A, R407F, R448A, or R449A.

4.1 EPR CAPACITY WHEN RETROFITTING FROM R22 OR R404A

An evaporator pressure regulator (EPR) valve is installed between the outlet of the evaporator and the suction header of a supermarket refrigeration system. Its function is to maintain the desired pressure in the evaporator while suction pressure downstream of it is lower, so it is a "holdback" or open on rise of inlet pressure device. Most evaporators in a system operate best when set at a design pressure, and therefore operate at the proper temperature. The evaporator will then deliver the proper temperature air to the case or walk-in cooler, and not frost up too quickly. Systems typically have loads that require different evaporator temperatures, but the lowest evaporator temperature on a system dictates the common pressure at which the suction header must be maintained. The EPR allows evaporators designed to operate at higher temperatures to do so, and maintains this pressure within a small range to buffer the evaporator from suction pressure fluctuations.

EPRs are sometimes used to control individual loads, but are more often used to control suction pressure on a system branch or circuit with multiple loads. For this reason, we will size for a two ton (24,000 Btu/hr) load at low temperature conditions (-20°F evaporator, 100°F condensing) and a four ton (48,000 Btu/hr) load at medium temperature conditions (20°F evaporator, 100°F condensing). We will assume the suction header is running at -25°F saturated for the low temperature system, and at 17°F for the medium temperature system. In an actual system, it would be typical to have 5°F difference on a low temperature header and 3°F on a medium temperature header. Sporlan offers EPRs with three different modes of operation: direct acting, internally piloted, and externally piloted. Our comparison will consider the externally piloted type, because it is the most common in this application.

The suction vapor volumetric flow of R404A and R22 is about the same for a given capacity under like conditions. Does that result in the selection of the same EPR? When using the low temperature system parameters already discussed, the Sporlan selection program gives the following results for R404A and R22:

EVAPORATOR PRESSURE REGULATOR SELECTION DATA (Using NIST RefProp DLL)			
Refrigerant	R- 404A HP62)	Liquid Temperature (°F):	90
Common Suction Temp (°F):	-25	Suction Vapor Temp (°F):	0
EPR Valve Setting (°F):	-20	Evaporator Capacity (tons):	2.00
Pressure Drop Available Across Valve (psi): 3.3			
Valve Type Description	Valve Type	Req'd Pressure Drop (psi)	Percentage of Rated Capacity
(S)ORIT-12 0/100	Piloted Externally	3.0	95

EVAPORATOR PRESSURE REGULATOR SELECTION DATA (Using NIST RefProp DLL)			
Refrigerant	R- 22	Liquid Temperature (°F):	90
Common Suction Temp (°F):	-25	Suction Vapor Temp (°F):	0
EPR Valve Setting (°F):	-20	Evaporator Capacity (tons):	2.00
Pressure Drop Available Across Valve (psi): 2.8			
Valve Type Description	Valve Type	Req'd Pressure Drop (psi)	Percentage of Rated Capacity
(S)ORIT-12 0/100	Piloted Externally	1.8	83

While percent loading varies some, the same SORIT-12 is chosen for both.
For medium temperature conditions:

EVAPORATOR PRESSURE REGULATOR SELECTION DATA			
(Using NIST RefProp DLL)			
Refrigerant	R- 404A HP62)	Liquid Temperature (°F):	90
Common Suction Temp (°F):	17	Suction Vapor Temp (°F):	40
EPR Valve Setting (°F):	20	Evaporator Capacity (tons):	4.00
Pressure Drop Available Across Valve (psi): 3.8			
Valve Type	Valve Type	Req'd Pressure	Percentage of
Description		Drop (psi)	Rated Capacity
(S)ORIT-15 0/100	Piloted Externally	1.4	62

EVAPORATOR PRESSURE REGULATOR SELECTION DATA			
(Using NIST RefProp DLL)			
Refrigerant	R- 22	Liquid Temperature (°F):	90
Common Suction Temp (°F):	17	Suction Vapor Temp (°F):	40
EPR Valve Setting (°F):	20	Evaporator Capacity (tons):	4.00
Pressure Drop Available Across Valve (psi): 3.2			
Valve Type	Valve Type	Req'd Pressure	Percentage of
Description		Drop (psi)	Rated Capacity
(S)ORIT-15 0/100	Piloted Externally	1.0	57

A SORIT-15 is chosen for both, and loading is again within a few percent difference. Let’s examine how these EPRs may function with the alternate refrigerants, starting with the low temperature example from above.

LOW TEMP (-25/-20/100)	R404A/R22 EPR SELECTION: SORIT-12					
	R404A	R22	R407A	R407F	R448A	R449A
ΔP Available (psi)	3.3	2.8	2.7	2.9	2.9	2.8
ΔP Required (psi)	3.0	1.8	2.6	2.1	2.4	2.4
EPR % Loading	95	83	97	86	92	93

Sizing for all the alternates is acceptable, with percent loading falling in between that of R404A and R22 for most, and R407A being slightly higher.
For the medium temperature condition and load:

MEDIUM TEMP (17/20/100)	R404A/R22 EPR Selection: SORIT-15					
	R404A	R22	R407A	R407F	R448A	R449A
ΔP Available (psi)	3.8	3.2	3.4	3.6	3.5	3.5
ΔP Required (psi)	1.4	1.0	1.2	1.0	1.2	1.2
EPR % Loading	62	57	61	55	59	60

Sizing for the alternates is acceptable at medium temperature conditions as well, with little variation from that of R404A or R22. When retrofitting an existing low or medium temperature R404A or R22 system to R407A, R407F, R448A, or R449A, existing evaporator pressure regulator valve sizing will likely be acceptable. It is good practice to check EPR sizing for confirmation.

While externally piloted EPRs were used in this example, there are many internally piloted EPRs in use today, particularly on loop-piped systems. The function of internally piloted EPRs can be more dependent on proper sizing than externally piloted ones. Thus, they may be more sensitive to a refrigerant retrofit. If a retrofit is to be performed on a system with internally piloted EPRs, sizing must be checked with the alternate refrigerant to ensure that sizing guidelines will not be exceeded. The manufacturer of the valve will be able to provide guidance on how the existing valve may function with an alternate refrigerant, and whether or not replacement is recommended.

4.2 EEPR CAPACITY WHEN RETROFITTING FROM R22 OR R404A

Electric evaporator pressure regulators are also widely used in supermarket systems. EEPRs perform the same function as described in Section 4.1 for an EPR, but are operated by an electric stepper motor, and controlled via a pressure transducer and electronic controller. We also expect EEPRs for R22 and R404A systems to be sized similarly. We will perform the same comparison for our two ton low and four ton medium temperature systems with EEPRs sized for R22 and R404A, then look at how the selected valves might perform with the alternates.

LOW TEMP (-25/-20/100)	R404A/R22 EEPR SELECTION: CDS-7					
	R404A	R22	R407A	R407F	R448A	R449A
ΔP Available (psi)	3.3	2.8	2.7	2.9	2.9	2.8
ΔP Required (psi)	1.6	1.0	1.4	1.1	1.3	1.3
EPR % Loading	72	63	74	65	70	71

MEDIUM TEMP (17/20/100)	R404A/R22 EEPR SELECTION: CDS-7					
	R404A	R22	R407A	R407F	R448A	R449A
ΔP Available (psi)	3.8	3.2	3.4	3.6	3.5	3.5
ΔP Required (psi)	2.4	1.6	2.0	1.7	1.9	1.9
EPR % Loading	80	72	78	70	75	76

As with mechanical EPRs, EEPRs for R404A and R22 are acceptable for the alternates. When retrofitting an existing low or medium temperature R404A or R22 system to R407A, R407F, R448A, or R449A, existing electric evaporator pressure regulator valve sizing will likely be acceptable. It is good practice to check EEPR sizing for confirmation.

4.3 SUCTION SOLENOID VALVE CAPACITY WHEN RETROFITTING FROM R22 OR R404A

The purpose of a suction solenoid valve is to provide electrically actuated on/off refrigerant flow control. Most are piloted valves, as discussed in Section 3.3. Suction solenoid valves are opened by the coil applying electromagnetic pull on the stem, but require a small amount of pressure drop across the valve to close it. At sizing conditions this pressure drop should be 1 psi or greater. If a solenoid valve is too large, there may not be enough ΔP across it to initially close it, particularly at low load conditions.

If a solenoid valve is too small, there will be more pressure drop across it than is desired. In a suction line, pressure drop is a system inefficiency – it causes the compressor to operate at a lower suction pressure than it would otherwise. For the refrigerants in this exercise, 1.7 – 2.0 psi of drop is 3°F of saturation at -20°F, and 2.1 – 2.6 psi of drop is 2°F saturation at 20°F. Because of this, suction solenoid valves with no greater than 3°F saturation drop at low temperature conditions, and no greater than 2°F saturation drop at medium temperature conditions should be selected. Within this range, the suction solenoid valve will function reliably without causing excessive loss of system efficiency. These valves are then applied to the alternate refrigerants.

LOW TEMP (-20/100)	R404A/R22 SUCTION SOLENOID SELECTION: E25					
	R404A	R22	R407A	R407F	R448A	R449A
Flow (lbm/min)	9.07	6.05	6.96	6.17	6.79	6.88
Sol ΔP (psi)	1.9	1.2	1.7	1.4	1.6	1.6
Sat Drop (°F)	2.7	2.2	2.5	2.0	2.3	2.3

MEDIUM TEMP (20/100)	R404A/R22 SUCTION SOLENOID SELECTION: E25					
	R404A	R22	R407A	R407F	R448A	R449A
Flow (lbm/min)	16.2	11.4	12.8	11.5	12.5	12.6
Sol ΔP (psi)	2.7	1.9	2.4	2.0	2.2	2.3
Sat Drop (°F)	2.1	1.8	1.8	1.5	1.6	1.7

An E25 solenoid valve is chosen for both the two ton low temperature load, and the four ton medium temperature load with R22 or R404A. When applied to the alternate refrigerants, the goal of maintaining at least 1 psi ΔP to ensure operation of the solenoid valve was met, while not excessively reducing system efficiency. Saturation drop was acceptable for all refrigerants at both medium and low temperature conditions.

When retrofitting an existing low or medium temperature R404A or R22 system to R407A, R407F, R448A, or R449A, existing suction solenoid valve sizing will likely be acceptable. It is good practice to check suction solenoid valve sizing for confirmation.

5

OTHER CONSIDERATIONS AND RETROFIT SUMMARY

There are several other areas that must be considered when performing a refrigerant retrofit. Among the most important of these are filter changes, oil changes, and replacement of elastomeric seals. This section will touch on these but not cover them exhaustively. There are many good refrigerant retrofit guides that cover these topics in detail.

5.1 SYSTEM FILTRATION

Any system refrigerant retrofit should include a change of all system filters, filter-driers or filter-drier cores, regardless of refrigerant. Clean filters minimize pressure drop, reducing startup difficulty and maximizing efficiency. It is best practice to check for significant pressure drop due to contamination after a few hours of system operation, and change filters again if significant contamination is found. Retrofits from R22/mineral oil

to HFC/POE oil are known to cause contamination deposits to dislodge and move through the system, so filter changes are a very important step in any retrofit of an R22 system. Filter manufacturers provide guidelines on filter changes for system cleanup and for normal operation.

New filter-driers or cores will also help ensure the system is dry as possible, benefiting system startup and operation. When the system is re-started following a retrofit, liquid line moisture indicators should be reviewed after 24 hours of operation. If moisture is still indicated in the system, change the filter-driers or cores again. This process should be repeated until the system is indicated dry.

There will normally be little concern with the size of liquid or suction filters when retrofitting from either R404A or R22 to R407A, R407F, R448A, or R449A. Filters are usually selected with some amount of additional capacity. It is good practice check the size of filter-driers, to confirm there will be both adequate flow capacity and moisture removal capability in relation to the size of the system. All Sporlan filter-driers, cores, and suction filters are fully compatible with these alternate refrigerants.

5.2 OIL CHANGES

When retrofitting an R22 system to R407A, R407F, R448A, or R449A, the existing mineral oil in the system must be changed to a synthetic POE oil. Some R22 systems may have POE oil instead of mineral oil. Recommended procedures for HCFC to HFC retrofits apply to the blends with HFO content as well. Established guidelines call for reducing mineral oil content to 5% or less through successive oil changes. Specific oil type and viscosity recommendations from the system or compressor manufacturer should be followed.

An R404A system will already have POE oil instead of mineral oil. When retrofitting to any of these alternates, a change of oil types is not necessary. It is still good practice to change oil in conjunction with a retrofit, in order to remove as many contaminants from the system as possible. The cleaner the system is, the more efficiently and reliably it will operate. In this situation of a POE-for-POE change, multiple oil changes (as may be necessary with a mineral-to-POE change) are likely unnecessary, unless there is severe oil contamination.

5.3 ELASTOMERIC SEALS

When retrofitting a system to R407A, R407F, R448A, or R449A, it is recommended to change the elastomeric seals in the system to minimize the possibility of external leaks. This is especially important with an R22 system retrofit. Elastomeric gaskets and O-rings that have been exposed long-term to R22 and mineral oil have a greatly increased risk of leaking after a deep vacuum being applied to the system, followed by exposure to HFC refrigerants and POE oil.

When retrofitting an R404A system, the seals are not exposed to different refrigerant and oil types after the retrofit. However, the existing seals will likely have been in place for a number of years. Pulling a deep vacuum on the system will still alter the set the seals have taken from being under internal pressure, and cause small amounts of refrigerant the seals have absorbed over time to begin to release from them. While the risk of leaks may be less for an HFC-to-HFC (or HFO) and POE-to-POE retrofit, it is

nonetheless best practice to change the elastomeric seals. Minimizing external refrigerant leaks benefits both the system's owner and the environment.

5.4 SUMMARY FOR RETROFITS TO R407A, R407F, R448A, OR R449A

In a retrofit from R404A, existing liquid side components may be too large, and TEV thermostatic elements are not compatible. Replacement of at least TEVs may be necessary. R404A suction side components may be okay. In a retrofit from R22, both liquid and suction side components are likely okay. Though most critical for an R22 system, any retrofit should include measures to ensure a clean and dry system, and minimize external leaks, including changing of filters, refrigeration oil, and elastomeric seals. Figure 6 summarizes Sections 2.1 – 5.3.

Figure 6 - Guidelines for Retrofits to R407A, R407F, R448A, or R449A

COMPONENT	FROM R404A			FROM R22		
	Existing is ...	Check Sizing?	Recommendation	Existing is ...	Check Sizing?	Recommendation
Distributor	Likely Okay	Required*	Replace only if sizing requires	Likely Okay	Required*	Replace only if sizing requires
TEV (Sizing)	Likely Oversized	Required	Replace TEV	Likely Okay	Required	Replace only if sizing requires
TEV (Thermostatic Element)	Not Compatible	N/A	Replace with R22 element**	Compatible	N/A	Don't replace
Liquid Line Solenoid Valve	Possibly Oversized	Required	Replace only if sizing requires	Likely Okay	Good Practice	Replace only if sizing requires
Externally Piloted EPR	Likely Okay	Good Practice	Replace only if sizing requires	Likely Okay	Good Practice	Replace only if sizing requires
Internally Piloted EPR	Likely Okay	Required	Replace only if sizing requires	Likely Okay	Required	Replace only if sizing requires
Electric EPR	Likely Okay	Good Practice	Replace only if sizing requires	Likely Okay	Good Practice	Replace only if sizing requires
Suction Solenoid Valve	Likely Okay	Good Practice	Replace only if sizing requires	Likely Okay	Good Practice	Replace only if sizing requires
Filter-Drier	Likely OK size, but contaminated	Good Practice	Change filter-drier	Likely OK size, but contaminated	Good Practice	Change filter-drier
Suction Filter	Likely OK size, but contaminated	Good Practice	Change suction filter	Likely OK size, but contaminated	Good Practice	Change suction filter
Refrigeration Oil	Compatible	N/A	Change to de-contaminate	Not Compatible	N/A	Change until <5% MO content
Elastomeric Seals	Leaks Possible	N/A	Best practice to change	Leaks Likely	N/A	Change seals

* In order to accurately check TEV sizing.

** Only if TEV sizing is OK, best practice is to replace the TEV.



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