Basic Refrigeration and Charging Procedures

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CHAPTER ONE

Refrigerant Pressures, States, and Conditions

The typical vapor compression refrigeration system shown in Figure 1-1 can be divided into two pressures: condensing (high side) and evaporating (low side). These pressures are divided or separated in the system by the compressor discharge valve and the metering device. Listed below are field service terms often used to describe these pressures:

Condensing Pressure	Evaporating Pressure
High side pressure	Low side pressure
Head pressure	Suction pressure
Discharge pressure	Back pressure

Condensing Pressure

The condensing pressure is the pressure at which the refrigerant changes state from a vapor to a liquid. This phase change is referred to as *condensation*. This pressure can be read directly from a pressure gauge connected anywhere between the compressor discharge valve and the entrance to the metering device, assuming there is negligible pressure drop. In reality, line and valve friction and the weight of the liquid itself cause pressure drops from the compressor discharge to the metering device. If a true condensing pressure is needed, the technician must measure the pressure as close to the condenser as possible to avoid these pressure drops. This pressure is usually measured on smaller systems near the compressor valves, Figure 1-2. On small systems, it is not critical where a technician places the pressure gauge (as long as it is on the high side of the system), because pressure drops are negligible. The pressure gauge reads the same no matter where it is on the high side of the system if line and valve losses are negligible.

Evaporating Pressure

The evaporating pressure is the pressure at which the refrigerant changes state from a liquid to a vapor. This phase change is referred to as *evaporation* or *vaporizing*. A pressure gauge placed anywhere between the metering device outlet and the compressor (including compressor crankcase) will read the evaporating pressure. Again, negligible pressure drops are assumed. In reality, there will be line and valve pressure drops as the refrigerant travels through the evaporator and suction line. The technician must measure the pressure as close to the evaporator as possible to get a true evaporating pressure. On small systems where pressure drops are negligible, this pressure is usually measured near the compressor (see Figure 1-2). Gauge placement on small systems is usually not critical as long as it is placed on the low side of the refrigeration system, because the refrigerant

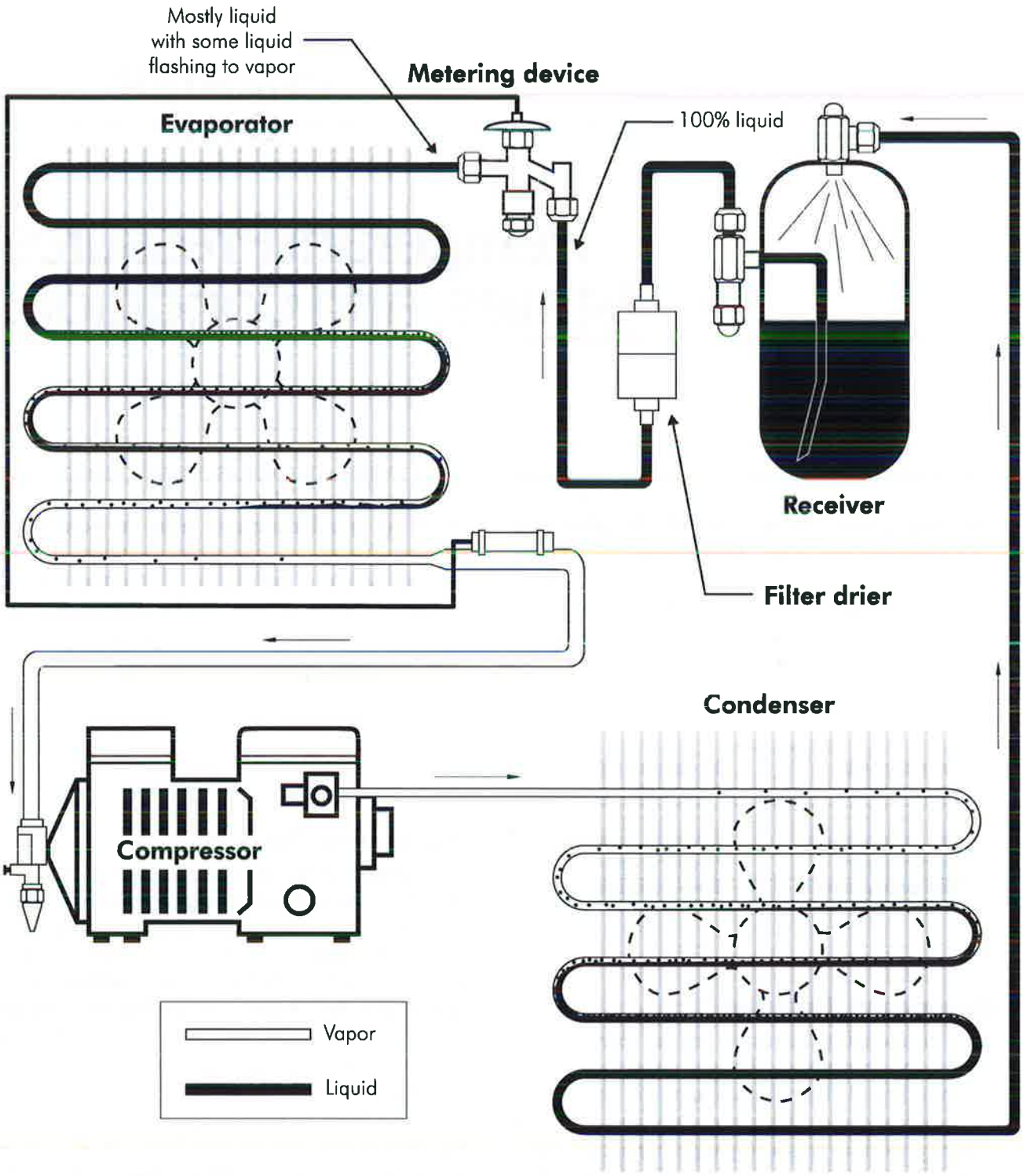


Figure 1-1. Typical compression refrigeration system

vapor pressure acts equally in all directions. If line and valve pressure drops become substantial, gauge placement becomes critical. In larger more sophisticated systems, gauge placement is more critical because of associated line and valve pressure losses. If the system has significant line and valve pressure losses, the technician must place the gauge as close as possible to the component that requires a pressure reading.

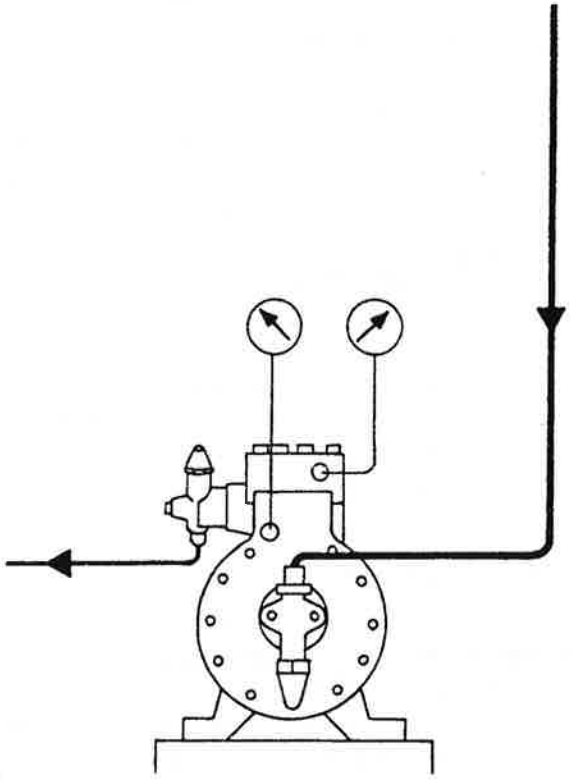


Figure 1-2. Semi-hermetic compressor showing pressure access valves (Courtesy, Danfoss Automatic Controls, Division of Danfoss, Inc.)

Refrigerant States and Conditions

Modern refrigerants exist either in the vapor or liquid state. Refrigerants have such low freezing points that they are rarely in the frozen or solid state. Refrigerants can co-exist as vapor and liquid as long as conditions are right. Both the evaporator and condenser house liquid and vapor refrigerant simultaneously if the system is operating properly. Refrigerant liquid and vapor can exist in both the high or low pressure sides of the refrigeration system.

Along with refrigerant pressures and states are refrigerant conditions. Refrigerant conditions can be *saturated*, *superheated*, or *subcooled.*

Saturation

Saturation is usually defined as a temperature. The saturation temperature is the temperature at which a fluid changes from liquid to vapor or vapor to liquid. At saturation temperature, liquid and vapor are called saturated liquid and saturated vapor, respectively. Saturation occurs in both the evaporator and condenser. At saturation, the liquid experiences its maximum temperature for that pressure, and the vapor experiences its minimum temperature. However, both liquid and vapor are at the same temperature for a given pressure when saturation occurs. Saturation temperatures vary with different refrigerants and pressures. All refrigerants have different vapor pressures. It is vapor pressure that is measured with a gauge.

Vapor Pressure

Vapor pressure is the pressure exerted on a saturated liquid. Any time saturated liquid and vapor are together (as in the condenser and evaporator), vapor pressure is generated. Vapor pressure acts equally in all directions and affects the entire low or high side of a refrigeration system.

As pressure increases, saturation temperature increases; as pressure decreases, saturation temperature decreases. Only at saturation are there pressure/temperature relationships for refrigerants. Table 1-1 shows the pressure/temperature relationship at saturation for refrigerant 134a (R-134a). If one attempts to raise the temperature of a saturated liquid above its saturation temperature, vaporization of the liquid will occur. If one attempts to lower the temperature of a saturated vapor below its saturation temperature, condensation will occur. Both vaporization and condensation occur in the evaporator and condenser, respectively.

The heat energy that causes a liquid refrigerant to change to a vapor at a constant saturation temperature for a given pressure is referred to as *latent heat*. Latent heat is the heat energy that causes a substance to change state without changing the temperature of the substance. Vaporization and condensation are examples of a latent heat process.

Temperature (°F)	Pressure (psig)	Temperature (°F)	Pressure (psig)
-10	1.8		
-9	2.2		
-8	2.6	30	25.6
-7	3.0	31	26.4
-6	3.5	32	27.3
-5	3.9	33	28.1
-4	4.4	34	29.0
-3	4.8	35	29.9
-2	5.3	40	34.5
-1	5.8	45	39.5
0	6.2	50	44.9
1	6.7	55	50.7
2	7.2	60	56.9
3	7.8	65	63.5
4	8.3	70	70.7
5	8.8	75	78.3
6	9.3	80	86.4
7	9.9	85	95.0
8	10.5	90	104.2
9	11.0	95	113.9
10	11.6	100	124.3
11	12.2	105	135.2
12	12.8	110	146.8
13	13.4	115	159.0
14	14.0	120	171.9
15	14.7	125	185.5
16	15.3	130	199.8
17	16.0	135	214.8
18	16.7		
19	17.3		
20	18.0		
21	18.7		
22	19.4		
23	20.2		
24	20.9		
25	21.7		
26	22.4		
27	23.2		
28	24.0		
29	24.8		

Table 1-1. R-134a saturated vapor/liquid pressure/ temperature chart

Superheat

Superheat always refers to a vapor. A superheated vapor is any vapor that is above its saturation temperature for a given pressure. In order for vapor to be superheated, it must have reached its 100% saturated vapor point. In other words, all of the liquid must be vaporized for superheating to occur; the vapor must be removed from contact with the vaporizing liquid. Once all the liquid has been vaporized at its saturation temperature, any addition of heat causes the 100% saturated vapor to start superheating. This addition of heat causes the vapor to increase in temperature and gain *sensible heat*. Sensible heat is the heat energy that causes a change in the temperature of a substance. The heat energy that superheats vapor and increases its temperature is sensible heat energy. Superheating is a sensible heat process. Superheated vapor occurs in the evaporator, suction line, and compressor.

Subcooling

Subcooling always refers to a liquid at a temperature below its saturation temperature for a given pressure. Once all of the vapor changes state to 100% saturated liquid, further removal of heat will cause the 100% liquid to drop in temperature or lose sensible heat. Subcooled liquid results. Subcooling can occur in both the condenser and liquid line and is a sensible heat process. Another method of subcooling liquid, called liquid pressure amplification™, is covered in Chapter Two. This method increases the pressure on subcooled liquid, causing it to be subcooled even more. This creates a liquid with a temperature below its new saturation temperature for the new higher pressure.

A thorough understanding of pressures, states, and conditions of the basic refrigeration system enables the service technician to be a good systematic troubleshooter. It is not until then that a service technician should even attempt systematic troubleshooting.

Basic Refrigeration System

Figure 1-3 illustrates a basic refrigeration system. The basic components of this system are the compressor, discharge line, condenser, receiver, liquid line, metering device, evaporator, and suction line. Mastering the function of each individual component can assist the refrigera-

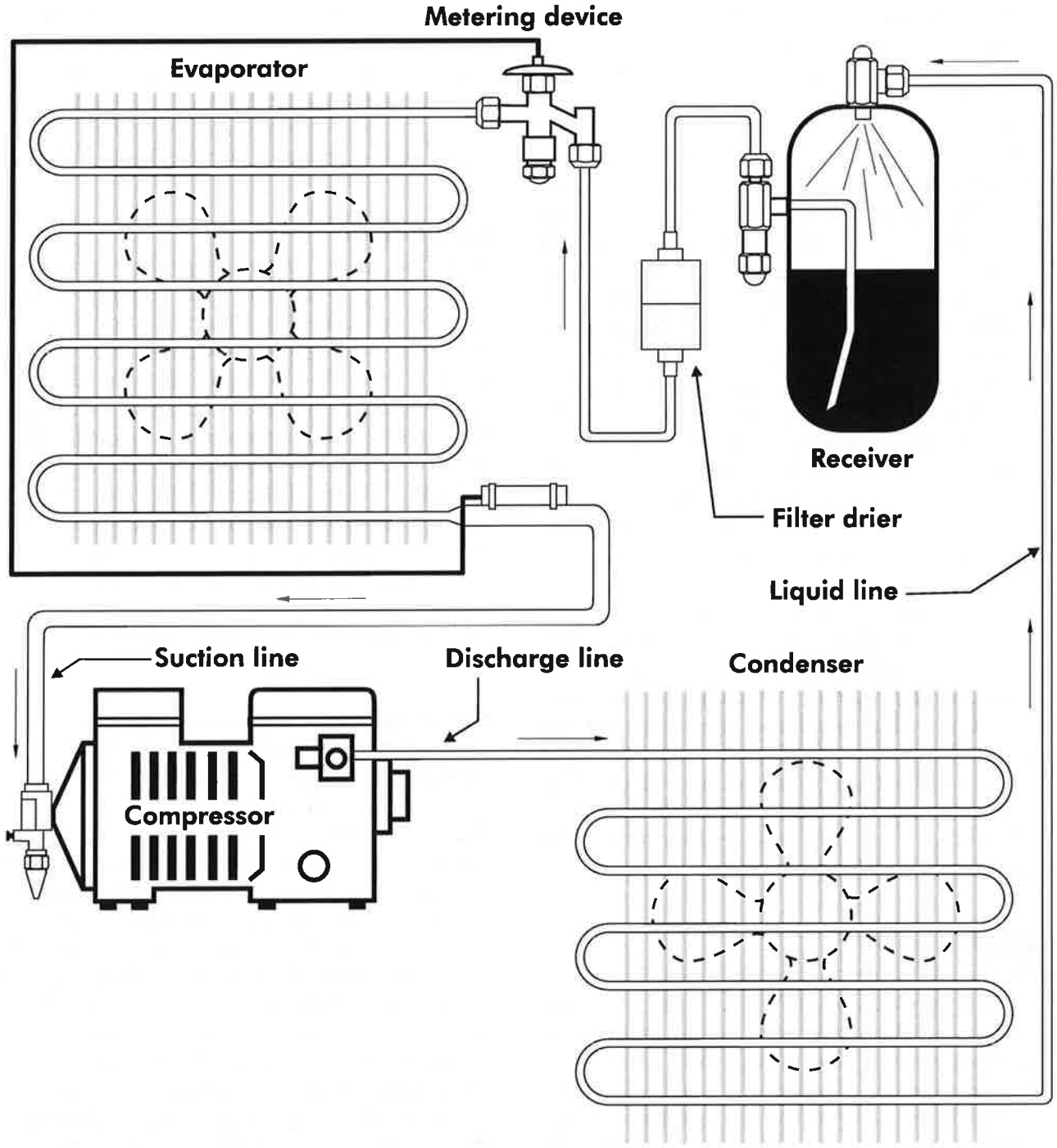


Figure 1-3. Basic refrigeration system

tion technician with analytical troubleshooting skills, saving time and money for both technician and customer.

Compressor

One of the main functions of the compressor is to circulate refrigerant. Without the compressor as a refrigerant pump, refrigerant could not reach other system components to perform its heat transfer functions. The compressor also separates the high pressure from the low pressure side of the refrigeration system. A difference in pressure is mandatory for fluid (gas or liquid) flow, and there could be no refrigerant flow without this pressure separation.

Another function of the compressor is to elevate or raise the temperature of the refrigerant vapor above the ambient (surrounding) temperature. This is accomplished by adding work, or heat of compression, to the refrigerant vapor during the compression cycle. The pressure of the refrigerant is raised, as well as its temperature. By elevating the refrigerant temperature above the ambient temperature, heat absorbed in the evaporator and suction line, and any heat of compression generated in the compression stroke can be rejected to this lower temperature ambient. Most of the heat is rejected in the discharge line and the condenser. Remember, heat flows from hot to cold, and there must be a temperature difference for any heat transfer to take place. The temperature rise of the refrigerant during the compression stroke is a measure of the increased internal kinetic energy added by the compressor.

The compressor also compresses the refrigerant vapors, which increases vapor density. This increase in density helps pack the refrigerant gas molecules together, which helps in the condensation or liquification of the refrigerant gas molecules in the condenser once the right amount of heat is rejected to the ambient. The compression of the vapors during the compression stroke is actually preparing the vapors for condensation or liquification.

Discharge Line

One function of the discharge line is to carry the high pressure superheated vapor from the compressor discharge valve to the entrance of the condenser. The discharge line also acts as a desuperheater, cooling the superheated vapors that the compressor has compressed and giving that heat up to the ambient (surroundings). These compressed vapors contain all of the heat that the evaporator and suction line have absorbed, along with the heat of compression of the compression stroke. Any generated motor winding heat may also be contained in the discharge line refrigerant, which is why the beginning of the discharge line is the hottest part of the refrigeration system. On hot days when the system is under a high load and may have a dirty condenser, the discharge line can reach over 400°F. By desuperheating the refrigerant, the vapors will be cooled to the saturation temperature of the condenser. Once the vapors reach the condensing saturation temperature for that pressure, condensation of vapor to liquid will take place as more heat is lost.

Condenser

The first passes of the condenser desuperheat the discharge line gases. This prepares the high pressure superheated vapors coming from the discharge line for condensation, or the phase change from gas to liquid. Remember, these superheated gases must lose all of their superheat before reaching the condensing temperature for a certain condensing pressure. Once the initial passes of the condenser have rejected enough superheat and the condensing temperature or saturation temperature has been reached, these gases are referred to as 100% saturated vapor. The refrigerant is then said to have reached the 100% saturated vapor point, Figure 1-4.

One of the main functions of the condenser is to condense the refrigerant vapor to liquid. Condensing is system dependent and usually takes place in the lower two-thirds of the condenser. Once the saturation or condensing temperature is reached in the condenser and the refrigerant gas has reached 100% saturated vapor, condensation can take place if more heat is removed. As more heat is taken away from the 100% saturated vapor, it will force the vapor to become a liquid or to condense. When condensing, the vapor will gradually phase change to liquid until 100% liquid is all that remains. This phase change, or change of state, is an example of a latent heat rejection process, as the heat removed is latent heat not sensible heat. The phase change will happen at one temperature even though heat is being removed. *Note: An exception to this is a near-azeotropic blend of refrigerants where there is a temperature glide or range of temperatures when phase changing (see Chapter Eight on blend temperature glide).* This one temperature is the saturation temperature corresponding to the saturation pressure in the condenser. As mentioned before, this pressure can be measured anywhere

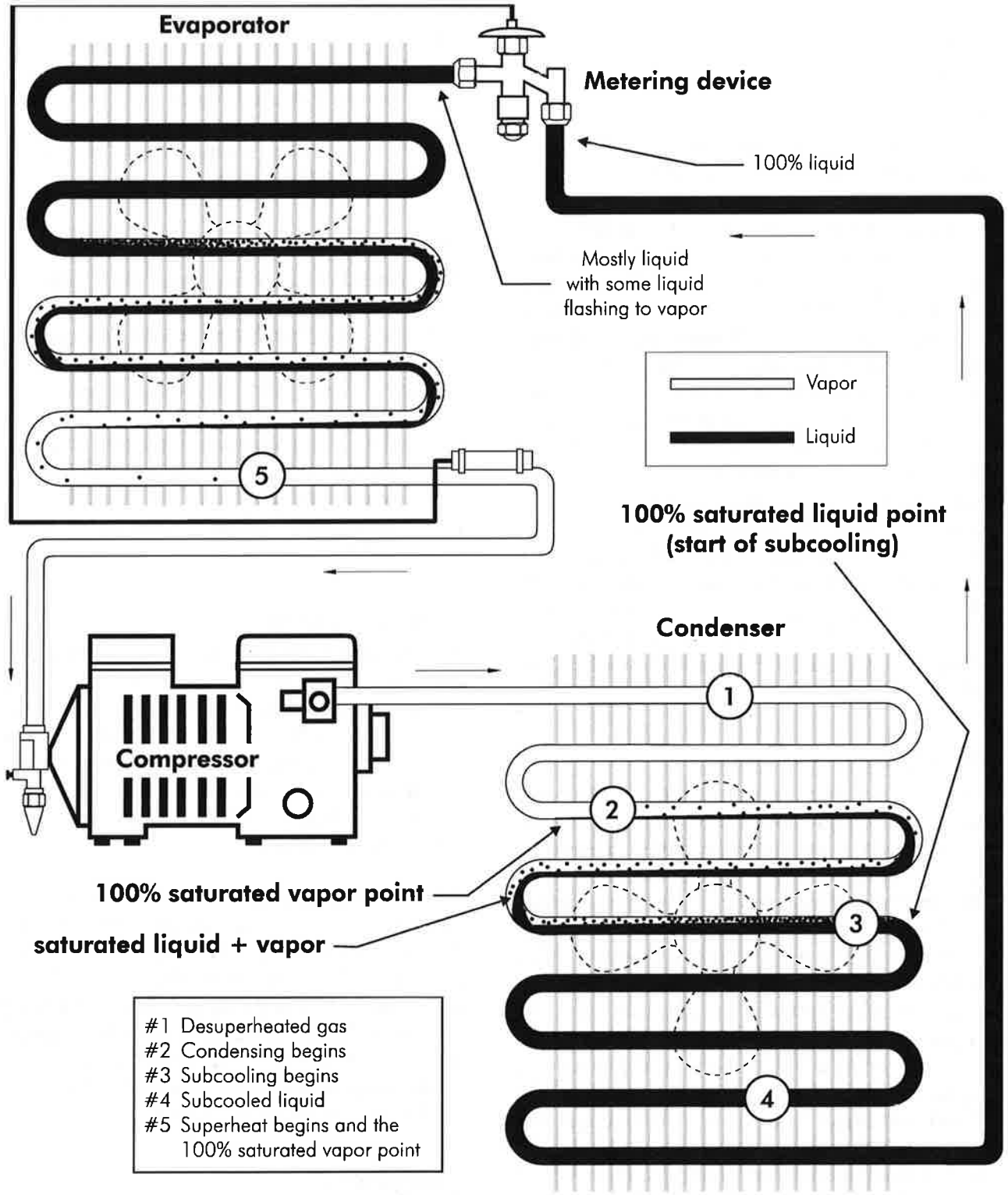


Figure 1-4. Basic refrigeration system showing 100% saturated vapor and liquid points

on the high side of the refrigeration system as long as line and valve pressure drops and losses are negligible.

The last function of the condenser is to subcool the liquid refrigerant. Subcooling is defined as any sensible heat taken away from 100% saturated liquid. Technically, subcooling is defined as the difference between the measured liquid temperature and the liquid saturation temperature at a given pressure. Once the saturated vapor in the condenser has phase changed to saturated liquid, the 100% saturated liquid point has been reached. If any more heat is removed, the liquid will go through a sensible heat rejection process and lose temperature as it loses heat. The liquid that is cooler than the saturated liquid in the condenser is subcooled liquid. Subcooling is an important process, because it starts to lower the liquid temperature to the evaporator temperature. This will reduce flash loss in the evaporator so more of the vaporization of the liquid in the evaporator can be used for useful cooling of the product load (see Chapter Two on the importance of liquid subcooling).

Receiver

The receiver acts as a surge tank. Once the subcooled liquid exits the condenser, the receiver receives and stores the liquid. The liquid level in the receiver varies depending on whether the metering device is throttling opened or closed. Receivers are usually used on systems in which a thermostatic expansion valve (TXV or TEV) is used as the metering device. The subcooled liquid in the receiver may lose or gain subcooling depending on the surrounding temperature of the receiver. If the subcooled liquid is warmer than receiver surroundings, the liquid will reject heat to the surroundings and subcool even more. If the subcooled liquid is cooler than receiver surroundings, heat will be gained by the liquid and subcooling will be lost.

A receiver bypass is often used to bypass liquid around the receiver and route it directly to the liquid line and filter drier. This bypass prevents subcooled liquid from sitting in the receiver and losing its subcooling. A thermostat with a sensing bulb on the condenser outlet controls the bypass solenoid valve by sensing liquid temperature coming to the receiver, Figure 1-5. If the liquid is subcooled to a predetermined temperature, it will bypass the receiver and go to the filter drier.

Liquid Line

The liquid line transports high pressure subcooled liquid to the metering device. In transport, the liquid may either lose or gain subcooling depending on the surrounding temperature. Liquid lines may be wrapped around suction lines to help them gain more subcooling, Figure 1-6. Liquid/suction line heat exchangers can be purchased and installed in existing systems to gain subcooling. The importance of liquid subcooling will be covered more extensively in Chapter Two.

Metering Device

The metering device meters liquid refrigerant from the liquid line to the evaporator. There are several different styles and kinds of metering devices on the market with different functions. Some metering devices control evaporator superheat and pressure, and some even have pressure limiting devices to protect compressors at heavy loads.

The metering device is a restriction that separates the high pressure side from the low pressure side in a refrigeration system. The compressor and the metering device are the two components that separate pressures in a refrigeration system. The restriction in the metering device causes liquid refrigerant to flash to a lower temperature in the evaporator because of its lower pressure and temperature.

Evaporator

The evaporator, like the condenser, acts as a heat exchanger. Heat gains from the product load and outside ambient travel through the sidewalls of the evaporator to vaporize any liquid refrigerant. The pressure drop through the metering device causes vaporization of some

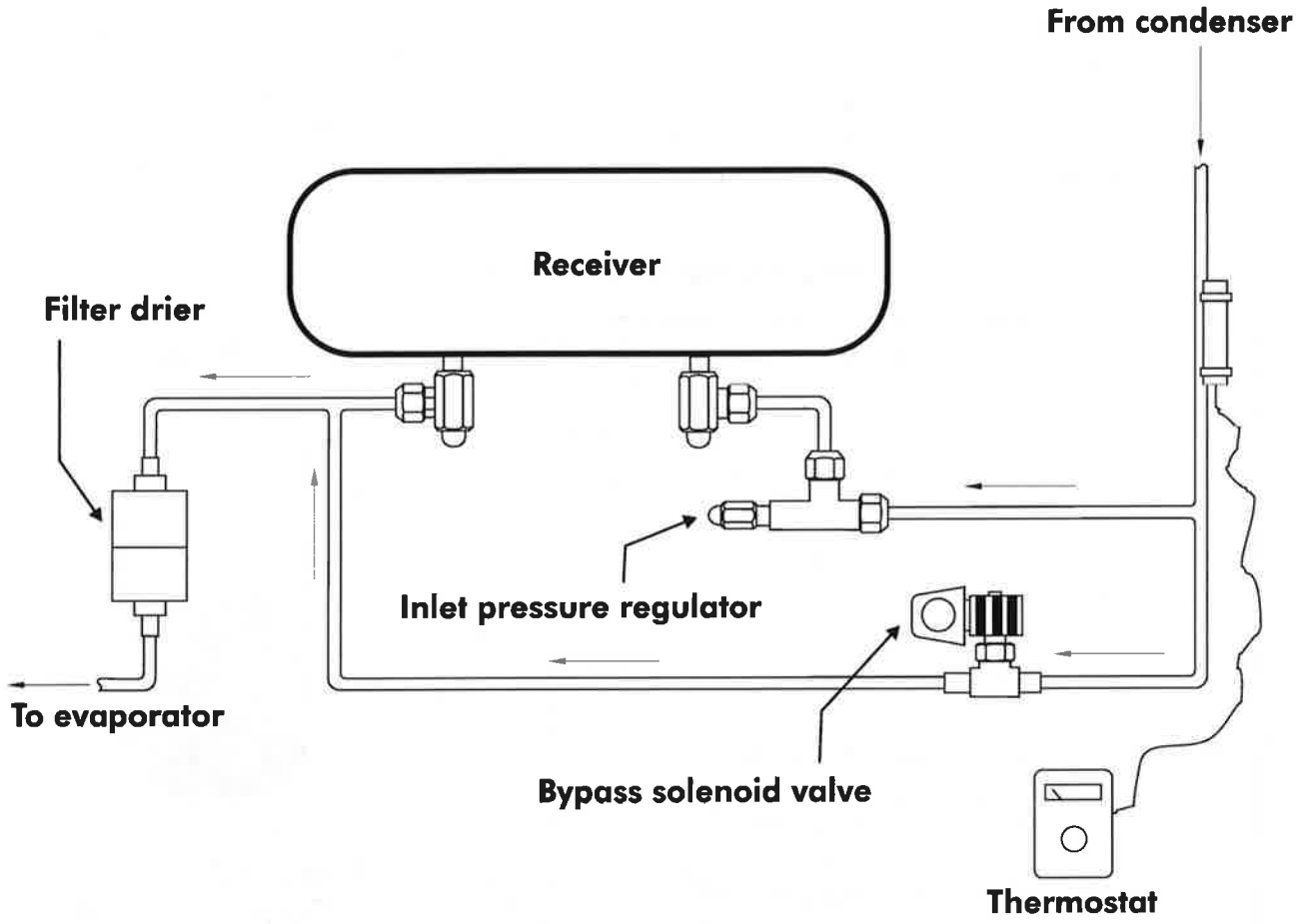


Figure 1-5. Receiver with thermostatically controlled liquid bypass



Figure 1-6. Liquid/suction line heat exchanger (Courtesy, Refrigeration Research, Inc.)

of the refrigerant and causes a lower saturation temperature in the evaporator. This temperature difference between the lower pressure refrigerant and the product load is the driving potential for heat transfer to take place.

The last pass of the evaporator coil acts as a superheater to ensure all liquid refrigerant has been vaporized. This protects the compressor from any liquid slopover, which may result in valve damage or diluted oil in the crankcase. The amount of superheat in the evaporator is usually controlled by a thermostatic expansion type of metering device.

Suction Line

The suction line transports low pressure superheated vapor from the evaporator to the compressor. There may be other components in the suction line such as suction accumulators, crankcase pressure regulators, p-traps, filters, and screens. Liquid/suction line heat exchangers are often mounted in the suction line to transfer heat away from the liquid line (subcool) and into the suction line, Figure 1-7.

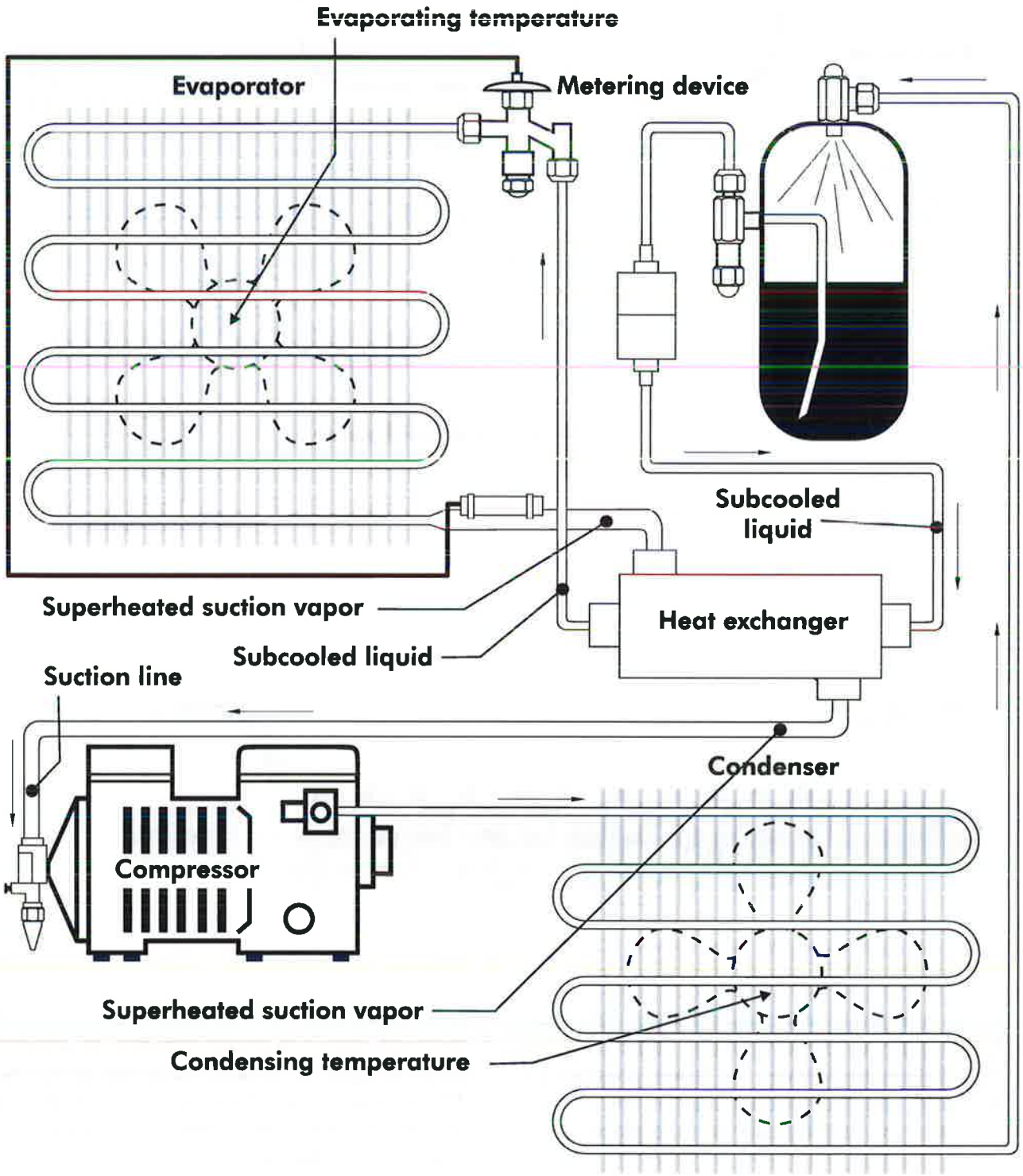


Figure 1-7. Refrigeration system showing liquid/suction line heat exchanger

Another function of the suction line is to superheat the vapor as it approaches the compressor. Even though suction lines are usually insulated, sensible heat stills penetrates the refrigerant vapor and adds more superheat. This additional superheat decreases the density of the refrigerant vapor to prevent compressor overload, resulting in lower amp draws. This additional superheat also helps ensure that the compressor will see vapor only under low loading conditions. Many metering devices have a tendency to lose control of evaporator superheat at low loads. It is recommended that systems should have at least 20°F of total superheat at the compressor to prevent liquid slugging and/or flooding of the compressor at low loadings. This topic will be covered extensively in Chapter Two.

Application of Pressures, States, and Conditions

Systematic troubleshooting requires mastering the function of all refrigeration system components. It is also important to be able to recognize the pressure, state, and condition of the working fluid (refrigerant) in the refrigeration system components. Figure 1-8 illustrates the basic refrigeration system. The legend lists refrigerant pressures, conditions, and states for the points shown in Figure 1-8. An explanation of the pressure, condition, and state of each point should clarify any system weaknesses.

Assume the following conditions for Figure 1-8:

- Refrigerant = R-134a
- Discharge (condensing) pressure = 124 psig (100°F)
- Suction (evaporating) pressure = 6 psig (0°F)
- Discharge temperature = 180°F
- Condenser outlet temperature = 90°F
- TXV inlet temperature = 80°F
- Evaporator outlet temperature = 10°F
- Compressor inlet temperature = 40°F

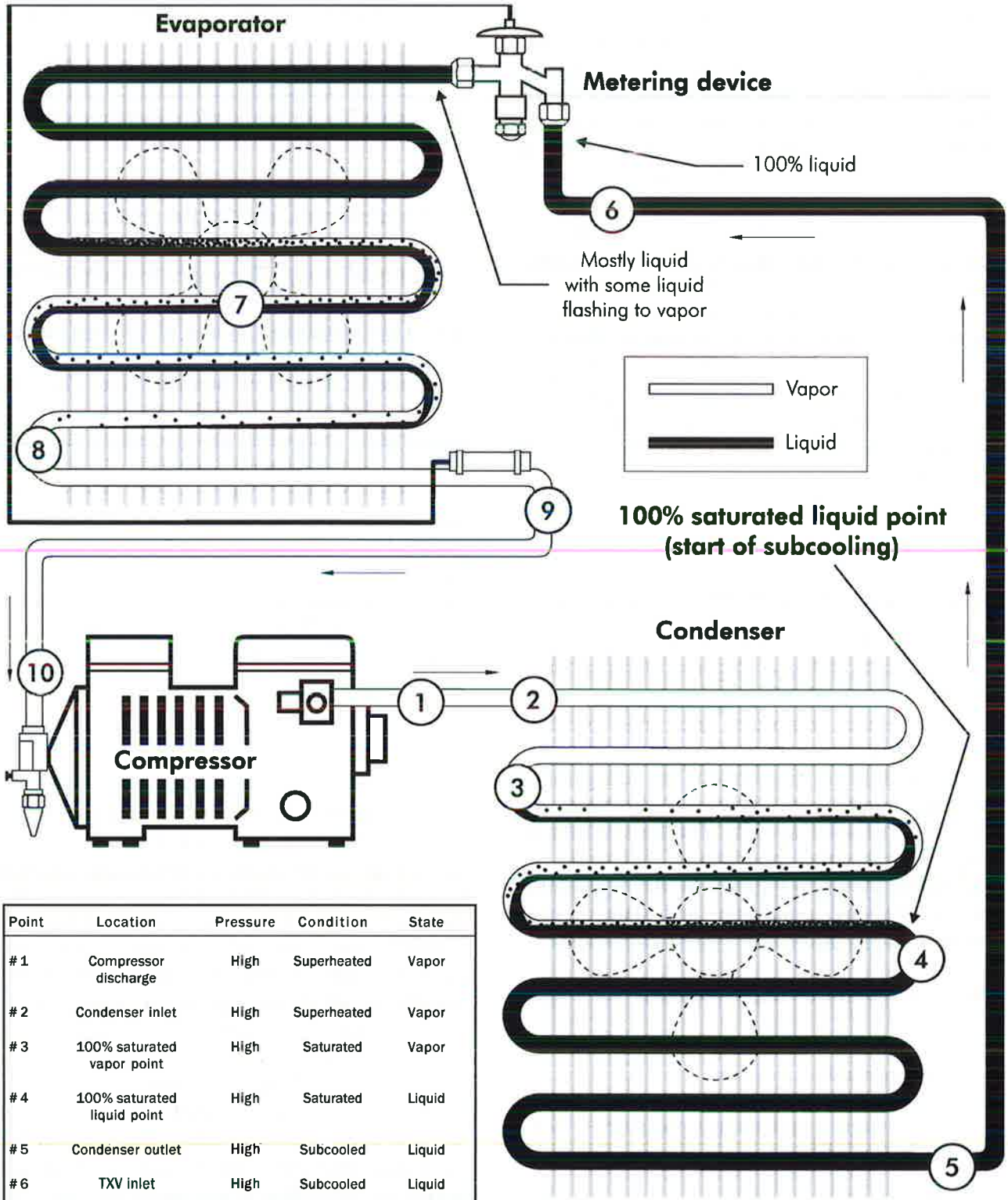
See Table 1-1 for pressure/temperature relationships.

Compressor Discharge (Point #1)

The refrigeration compressor is a vapor pump, not a liquid pump. The vapor leaving the compressor will be high pressure superheated vapor. The compressor is one of the two components in the system that separates the high pressure side from the low pressure side. The compressor discharge is high pressure, and the compressor suction is low pressure. The compressor discharge vapor receives its superheat from sensible heat coming from the evaporator, suction line, motor windings, friction, and internal heat of compression from the compression stroke. Since the vapor is superheated, no pressure/temperature relationship exists. Its temperature is well above the saturation temperature of 100°F for the given saturation (condensing) pressure of 124 psig.

Condenser Inlet (Point #2)

As high pressure superheated refrigerant leaves the compressor, it instantly begins to lose superheat and cool in temperature. Its heat is usually given up to the surroundings. As mentioned before, this process is called desuperheating. Even though this refrigerant vapor is going through a desuperheating process, it is still superheated vapor. Pressure acts equally in all directions, so the vapor will be high pressure, or the same pressure as the compressor discharge, assuming that any line and valve pressure drops are ignored. Remember, the refrigerant is superheated and not saturated, so there is no pressure/temperature relationship. This high pressure superheated vapor is also above its saturation temperature of 100°F for the given discharge or condensing pressure of 124 psig. This point can be referred to as high pressure superheated vapor. This process of desuperheating will continue until the 100% saturated vapor point in the condenser is reached.



Point	Location	Pressure	Condition	State
#1	Compressor discharge	High	Superheated	Vapor
#2	Condenser inlet	High	Superheated	Vapor
#3	100% saturated vapor point	High	Saturated	Vapor
#4	100% saturated liquid point	High	Saturated	Liquid
#5	Condenser outlet	High	Subcooled	Liquid
#6	TXV inlet	High	Subcooled	Liquid
#7	Middle of evaporator	Low	Saturated	Vapor and liquid
#8	100% saturated vapor point	Low	Saturated	Vapor
#9	Evaporator outlet	Low	Superheated	Vapor
#10	Compressor inlet	Low	Superheated	Vapor

Figure 1-8. Basic refrigeration system showing refrigerant pressures, states, and condition locations

100% Saturated Vapor Point (Point #3)

Once all the superheat is rejected from the refrigerant gas, the saturation temperature of 100°F is finally reached for the condensing pressure of 124 psig (see Table 1-1 for pressure/temperature relationship). The vapor has now reached the 100% saturated vapor point and is at the lowest temperature it can be and still remain a vapor. This temperature is referred to as both its saturation and condensing temperature, and a pressure/temperature relationship exists. Any heat lost past the 100% saturated vapor point will gradually phase change the vapor to liquid (condensing). The heat removed from the vapor turning to liquid is referred to as latent heat and happens at a constant temperature of 100°F. As the vapor condenses to liquid, refrigerant molecules actually become more dense and get closer together. This molecular joining is what gives up most of the latent heat energy. This point is on the high side of the refrigeration system and is referred to as high pressure saturated vapor.

100% Saturated Liquid Point (Point #4)

Soon all of the vapor will give up its latent heat and turn to saturated liquid at a constant condensing temperature of 100°F. Any more heat given up by the refrigerant after this point will be sensible heat, because the phase change from vapor to liquid is complete. This point is still on the high side of the refrigeration system and can be referred to as high pressure saturated liquid. The entire condensing process takes place between the 100% saturated vapor point and the 100% saturated liquid point. Any sensible heat lost past the 100% saturated liquid point is referred to as subcooling.

Condenser Outlet (Point #5)

Once the 100% saturated liquid point is reached in the condenser, liquid subcooling occurs. Remember, any heat lost in the liquid past its 100% saturated liquid point is subcooling. Liquid subcooling can continue all the way to the entrance of the metering device if conditions are right. This point is on the high side of the system and is all subcooled liquid, so it is referred to as high pressure subcooled liquid. There is no pressure/temperature relationship at the subcooled condition, only at saturation. At a pressure of 124 psig, the corresponding temperature of the 100% saturated liquid point in the condenser is 100°F. The difference between 100°F and the condenser outlet temperature of 90°F is 10°F of condenser subcooling. Subcooling calculations will be covered much more extensively in Chapter Two.

TXV Inlet (Point #6)

The inlet to the TXV is on the high side of the system and consists of subcooled liquid. This subcooling should continue from the 100% saturated liquid point in the condenser. The tubing from the condenser outlet to the TXV inlet is often referred to as the liquid line. The liquid line may be exposed to very high or low roof temperatures depending on the time of year. This will seriously affect whether or not subcooling takes place and to what magnitude. If the liquid line is exposed to hot temperatures, liquid line flashing may occur (covered in Chapter Two). Since this point is on the high side of the system and is subcooled liquid, it is referred to as high pressure subcooled liquid.

Middle of Evaporator (Point #7)

When the subcooled liquid enters the TXV, flashing of the liquid occurs. Once in the evaporator, the liquid refrigerant experiences a severe drop in pressure to the new saturation (evaporator) pressure of 6 psig. This pressure decrease causes some of the liquid to flash to vapor in order to reach the new saturation temperature in the evaporator of 0°F. Once this new evaporator temperature is reached, the liquid/vapor mixture starts absorbing heat from the product load and continues to change from liquid to vapor. This process happens at the new saturation temperature of 0°F, corresponding to the saturation (evaporator) pressure of 6 psig. This is a classic example of heat absorbed by the refrigerant without increasing in temperature, which is called the latent heat of vaporization. The heat energy absorbed in the

refrigerant breaks the liquid molecules into vapor molecules instead of increasing its temperature. Since the refrigerant is both saturated liquid and vapor and is on the low side of the refrigeration system, it is referred to as a low pressure saturated liquid and vapor.

100% Saturated Vapor Point (Point #8)

Once all of the liquid changes to vapor in the evaporator, the 100% saturated vapor point is reached. This point is still at the evaporator saturation temperature of 0°F. Any more heat absorbed by the refrigerant vapor will result in a temperature rise of the refrigerant. This heat energy goes into increasing the velocity and spacing of the vapor molecules, because there is no more liquid to be vaporized. This increase in molecular velocity can be measured in degrees. Any heat added past this 100% saturated vapor point is superheat. Since this 100% saturated vapor point is in the low side of the system and is saturated vapor, it is referred to as low pressure saturated vapor.

Evaporator Outlet (Point #9)

The evaporator outlet temperature is used for evaporator superheat calculations. This point is located at the evaporator outlet next to the TXV remote bulb. Because it is located downstream of the 100% saturated vapor point, it is superheated. This point is in the low side of the refrigeration system and is referred to as low pressure superheated vapor. The difference between the 100% saturated vapor temperature of 0°F and the evaporator outlet temperature of 10°F is called evaporator superheat. In this example, there are 10°F of evaporator superheat.

Compressor Inlet (Point #10)

The compressor inlet consists of low pressure superheated vapor. This vapor feeds the compressor. As the refrigerant travels from the evaporator outlet down the suction line to the compressor, more superheat is gained. Superheat ensures that no liquid refrigerant enters the compressor at low evaporator loadings when TXV valves are known to lose control of superheat settings. Because this point is superheated, no refrigerant pressure/temperature relationship exists.

CHAPTER TWO

Subcooling and Superheating

The concepts of subcooling and superheating are probably the two most important principles that the service technician must understand before attempting to systematically troubleshoot hvac/r systems. This chapter covers not only basic principles of subcooling and superheat, but also topics for the more advanced service technician. Examples dealing with superheat and subcooling amounts are also included for any clarifications that may be needed.

Liquid Subcooling

In today's competitive service market, every conscientious service technician should understand why a refrigeration or air conditioning system must have the proper amount of liquid subcooling. The amount of liquid subcooling affects system capacity, as well as the effectiveness and capacity of expansion type metering devices.

Subcooling is defined as the difference between the measured liquid temperature and the liquid saturation temperature at a given pressure. Any sensible heat taken away from the 100% saturated liquid point in the condenser can be defined as liquid subcooling. Liquid subcooling may occur from the start of the 100% saturated liquid point in the condenser to the metering device, Figure 2-1. The saturated liquid temperature can be obtained from a pressure/temperature relationship using the condensing pressure. This means that as soon as all of the saturated vapor in the condenser changes to saturated liquid, subcooling will start to occur if further heat is removed. This is a temperature change or sensible heat change, so any drop in temperature of the liquid below the saturation temperature for the pressure at that point is considered liquid subcooling.

Subcooling Categories

There are two subcooling categories: *condenser subcooling* and *total subcooling*.

Condenser Subcooling

Condenser subcooling is the liquid subcooling present leaving the condenser. It can be measured by subtracting the actual condenser liquid out temperature from the saturation temperature measured at the condenser outlet:

Condensing temperature (saturated) − Condenser liquid out temperature = Condenser subcooling

[Formula 2-1]

When subcooled, the refrigerant is not generating or losing any vapor pressure, so there is no pressure/temperature relationship and a pressure/temperature chart cannot be used. The condenser outlet temperature must be measured

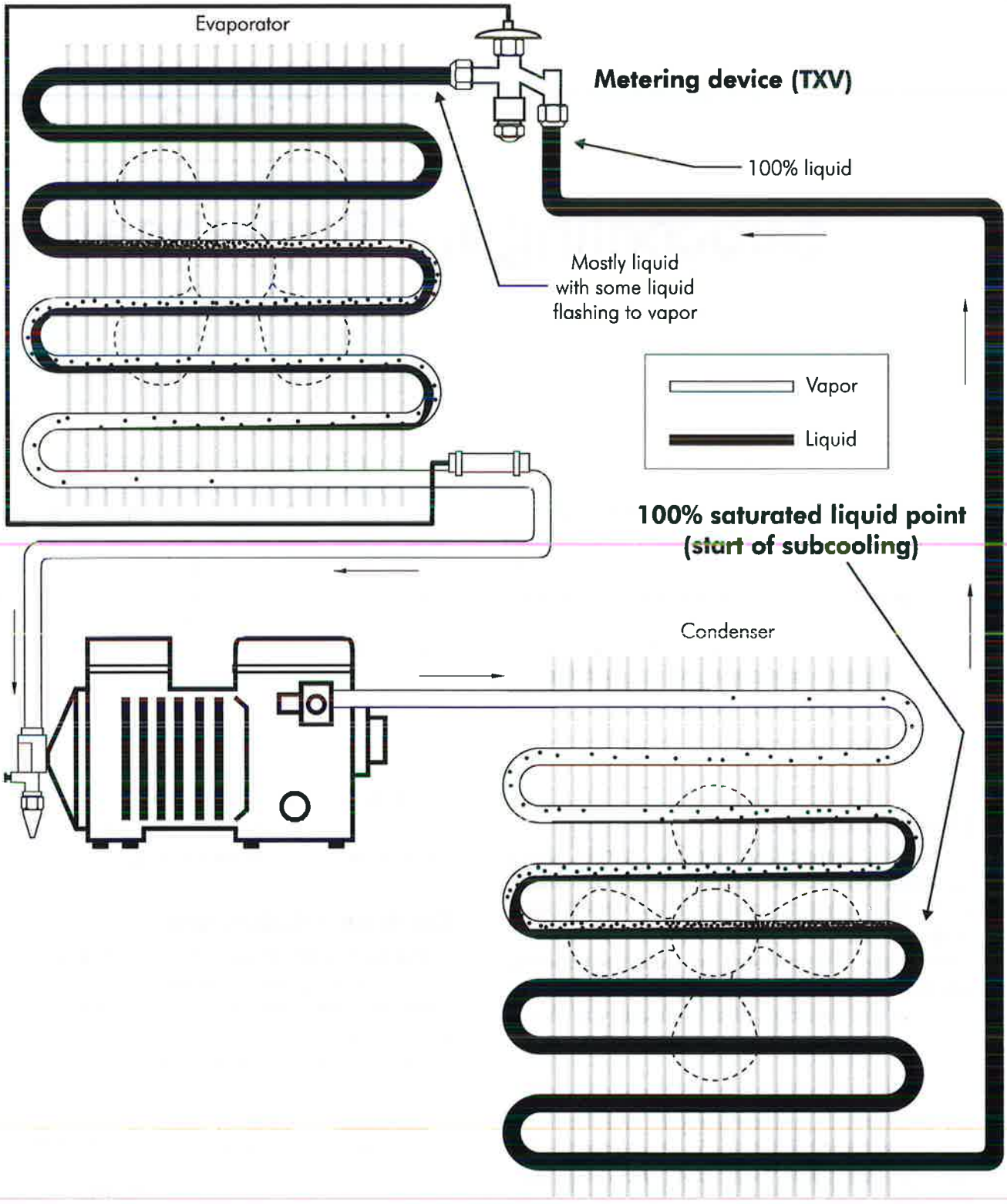


Figure 2-1. Start of liquid subcooling

with either a thermistor or a thermocouple fastened to the condenser outlet, Figure 2-2. The saturated temperature may be acquired from the condensing pressure read from a gauge on the condenser outlet. This is because a pressure/temperature relationship does exist in a saturated condition. If pressure drops exist in the system, the pressure must be measured where the temperature was taken to obtain an accurate liquid subcooling amount.

Example 2-1
What is the condenser subcooling if the head pressure on an R-22 system is 211 psig and a thermistor at the condenser outlet measures 95°F? (Assume no pressure drops through the condenser.) *Note: Thermistors or thermocouples must have good thermal contact onto the refrigerant line. The line must be straight and lightly sanded to ensure good thermal contact. Insulation must be placed around the line to insulate against outside ambient heat losses or gains.*

Solution 2-1
Convert 211 psig for R-22 to a 105°F condensing temperature using the pressure/temperature chart shown in Table 2-1:

105°F (condensing saturation temperature)	–	95°F (condenser out temperature)	=	10°F (condenser subcooling)

This means that there are 10°F of liquid subcooling leaving the condenser to feed the liquid line or receiver, depending on the type of system employed. However, if the liquid goes to a flow-through receiver, subcooling could be lost.

Example 2-2
This example incorporates pressure drop through the condenser into the equation for condenser subcooling. In this case, the pressure must be measured where the condenser outlet temperature is taken.

What is the condenser subcooling if the head pressure on an R-22 system is 211 psig, the thermistor at the condenser outlet measures 95°F, and the pressure reads 205 psig at that

Temperature (°F)	Pressure (psig)
65	111.23
66	113.22
67	115.24
68	117.28
69	119.34
70	121.43
71	123.54
72	125.67
73	127.83
74	130.01
75	132.22
76	134.45
77	136.71
78	138.99
79	141.30
80	143.63
81	145.99
82	148.37
83	150.78
84	153.22
85	155.68
86	158.17
87	160.69
88	163.23
89	165.80
90	168.40
91	171.02
92	173.67
93	176.35
94	179.06
95	181.80
96	184.56
97	187.36
98	190.18
99	193.03
100	195.91
101	198.82
102	201.76
103	204.72
104	207.72
105	210.75
106	213.81
107	216.90
108	220.02
109	223.17

Table 2-1. Pressure/temperature relationship for R-22

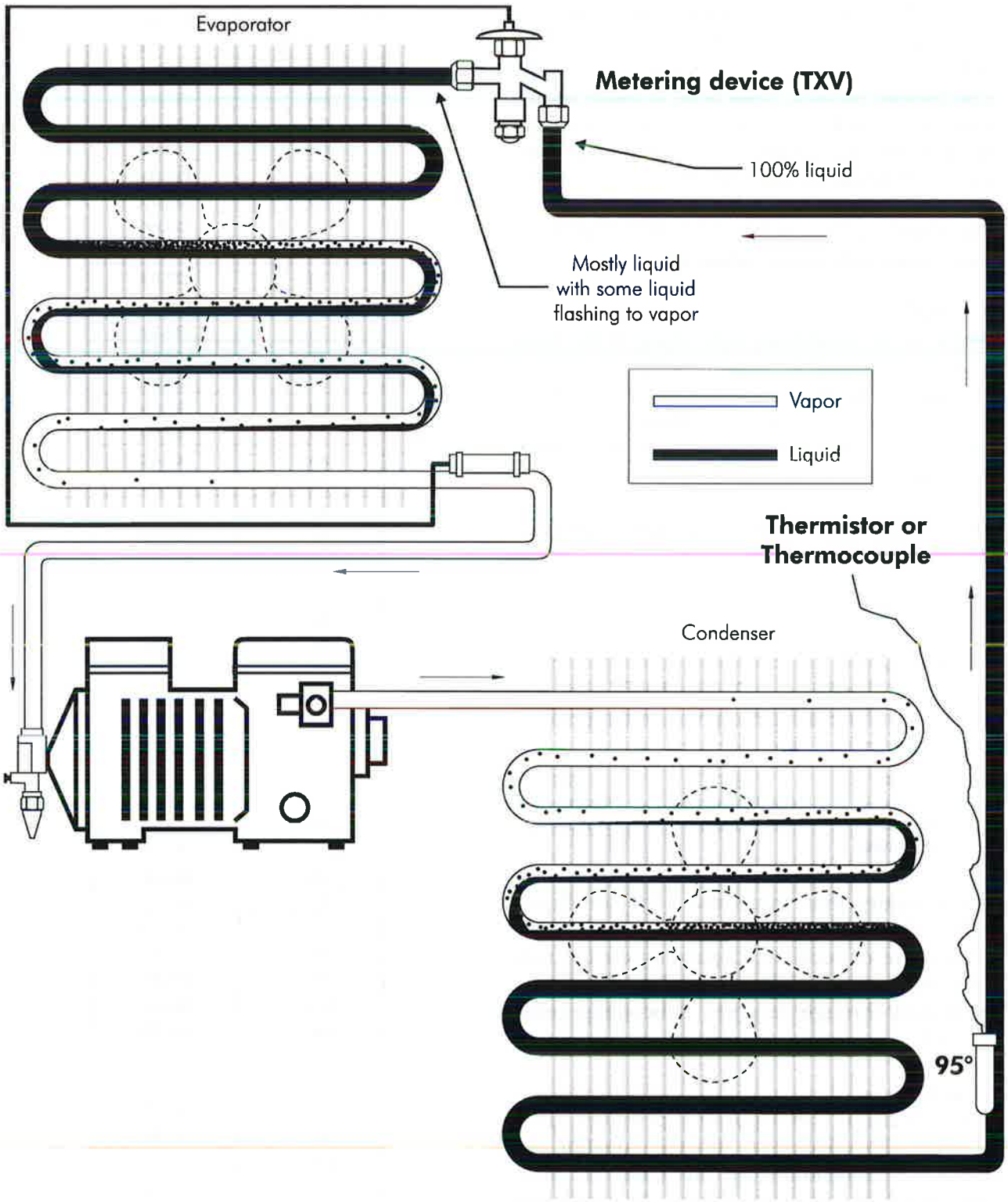


Figure 2-2. Thermistor or thermocouple at the condenser outlet

point? There is a 6 psi pressure drop through the condenser from line losses. The condenser outlet pressure can be read at the king valve, receiver charging valve, or simply by installing a line tap valve.

Solution 2-2
Convert 205 psig for R-22 to 103°F using Table 2-1:

103°F (saturation temperature at condenser outlet)	–	95°F (condenser out temperature)	=	8°F (condenser subcooling)

Total Subcooling

Total subcooling encompasses liquid subcooling from the 100% saturated liquid point to the expansion valve or metering device, Figure 2-3. This means that total subcooling includes condenser subcooling and any subcooling that takes place after the condenser. Liquid can lose heat and subcool in the receiver, filter drier, and liquid line before it gets to the metering device. However, if there is any refrigerant vapor in the receiver, the subcooled liquid will not subcool further. The vapor will re-condense to a liquid in the receiver, and both the liquid and vapor will reach a new saturation temperature. As the vapor re-condenses it rejects heat, which will be absorbed by the subcooled liquid. Subcooling can be lost in this situation if the liquid stays in the receiver too long.

Subcooling of liquid rarely occurs when a capillary tube metering device is used because of the severe pressure drop of the liquid as it travels the length of the capillary tube. Due to this reduction in pressure through the capillary tube, the liquid may actually be saturated with flash gas present. Because of the difficulty measuring the temperature at the capillary tube outlet in a service environment, assume that total subcooling ends at the capillary tube entrance.

Total subcooling can be calculated by subtracting the liquid line temperature at the entrance of the metering device from the saturation temperature at that point:

Saturation temperature from pressure at TXV inlet	–	TXV inlet temperature	=	Total subcooling

[Formula 2-2]

The concept of total subcooling is meaningless without incorporating pressure drops into the equation. Example 2-3 is a total subcooling calculation that does not take pressure drop into consideration.

Example 2-3
What is the total subcooling if the head pressure of an R-22 system is 223 psig, and a thermistor on the liquid line at the entrance of the metering device reads 90°F (assume no pressure drop)?

Solution 2-3
Convert 223 psig for R-22 to a 109°F condensing temperature using Table 2-1:

109°F (condensing saturation temperature)	–	90°F (liquid at metering device temperature)	=	19°F (total subcooling)

There are 19°F of total liquid subcooling before the liquid enters the metering device. *Note: When measuring the liquid line temperature at the entrance of a TXV or AXV metering device, place the thermistor or thermocouple on the liquid line about 3 to 4 inches in front of the expansion valve and insulate it from ambient heat gains. Measurement instruments placed too close to the metering device will measure colder temperatures due to heat conduction to the colder expansion device.*

Example 2-4
In this example, pressure drop is incorporated into the system as the liquid travels through the condenser to the metering device. This is typical of most medium and large systems.

What is the total subcooling if the head pressure of an R-22 system is 223 psig, a thermistor on the liquid line at the entrance of the metering device reads 90°F, and the pressure is 216 psig at that point? There is a 7 psi pressure

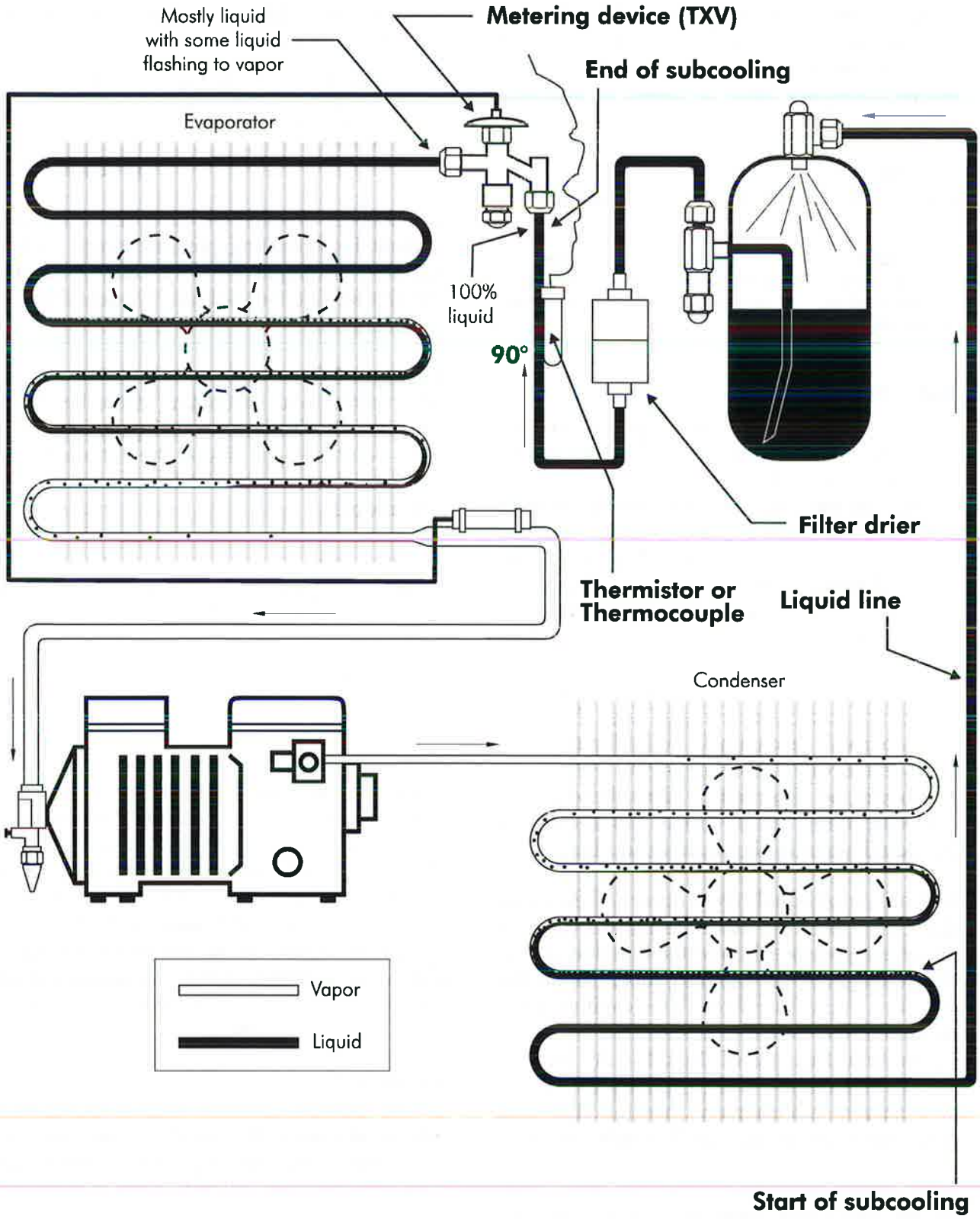


Figure 2-3. Thermistor or thermocouple at the TXV inlet

drop before the liquid reaches the metering device. This pressure drop could be static, friction, or both. The pressure of the liquid is measured just before the metering device where the 90°F temperature is taken, Figure 2-4. Line tap valves are often needed for this procedure.

Solution 2-4
Convert 216 psig for R-22 to a 107°F saturated temperature using Table 2-1:

107°F (saturated temperature for 216 psig at TXV inlet)	-	90°F (liquid at metering device temperature)	=	17°F (total subcooling)

Subcooling Needed to Prevent Liquid Line Flash

Condenser subcooling ensures there is a liquid seal at the condenser bottom so the liquid line or receiver is not fed with vapor. This condition prevents any refrigerant vapor or noncondensibles (e.g., air) from leaving the condenser. On receiverless systems, condenser subcooling can be an indicator of how much refrigerant charge is in a system. See manufacturer specifications for condenser subcooling amounts when charging a system using this criteria.

Another important reason why condenser subcooling must be present is to prevent liquid line flash gas (vapor bubbles) from forming in the liquid line. Liquid line flash gas is primarily caused from a pressure drop in the liquid line. Pressure drop in the liquid line can be caused by any restrictive device, including filter driers, sightglasses, solenoid valves, shutoff valves, or kinked or undersized liquid lines. There is also pressure drop associated with the length of run and the vertical lift of liquid in refrigerant lines. The longer the run and the more turns and bends associated with the liquid line, the larger the associated pressure drop. There is also more pressure drop with more vertical lift of liquid.

If there were no subcooling, saturated liquid from the condenser (while experiencing these pressure drops) would try to cool itself down to a lower saturation temperature to match the progressively lower pressure in the liquid line. The only way the saturated liquid could cool itself down is for some of its own liquid to flash into vapor. Whenever a liquid flashes into vapor, heat energy is needed for the vaporization process. Almost all of the heat supplied to the flashing process is from the heat in the saturated liquid itself. This cools the remaining liquid down to the new saturation temperature associated with the new lower pressure caused from the pressure drop. This type of cooling is referred to as *adiabatic cooling*, because there is no net loss or gain of energy in the liquid. Whenever saturated liquid experiences liquid line pressure drop, liquid line flash gas results and is detrimental to system performance. Metering devices will then experience a mixture of liquid/vapor instead of a solid column of liquid at their entrances. The liquid/vapor mixture decreases the capacity of the valves, because TXV capacity tables are based on vapor-free liquid at their inlets.

The balanced port TXV is designed to handle a certain amount of flash gas due to its oversized port. It is designed to maintain better control under such conditions and at light loads. The balanced port TXV can operate with its valve pin very close to the seat. This allows a large port to handle small loads. When liquid flashes in the liquid line, it also leaves less liquid to flash in the evaporator. This takes away from the refrigeration effect and adversely affects system capacity. Excessive liquid line flash gas may even erode the needle and seat areas of some TXVs. The erosion of needle valves usually occurs only with ammonia valves.

Pressure Drop

Total pressure losses that take place in the liquid line are the result of either friction pressure losses or static pressure losses.

Friction Pressure Losses

Friction pressure losses were discussed earlier. They are caused from refrigerant flowing

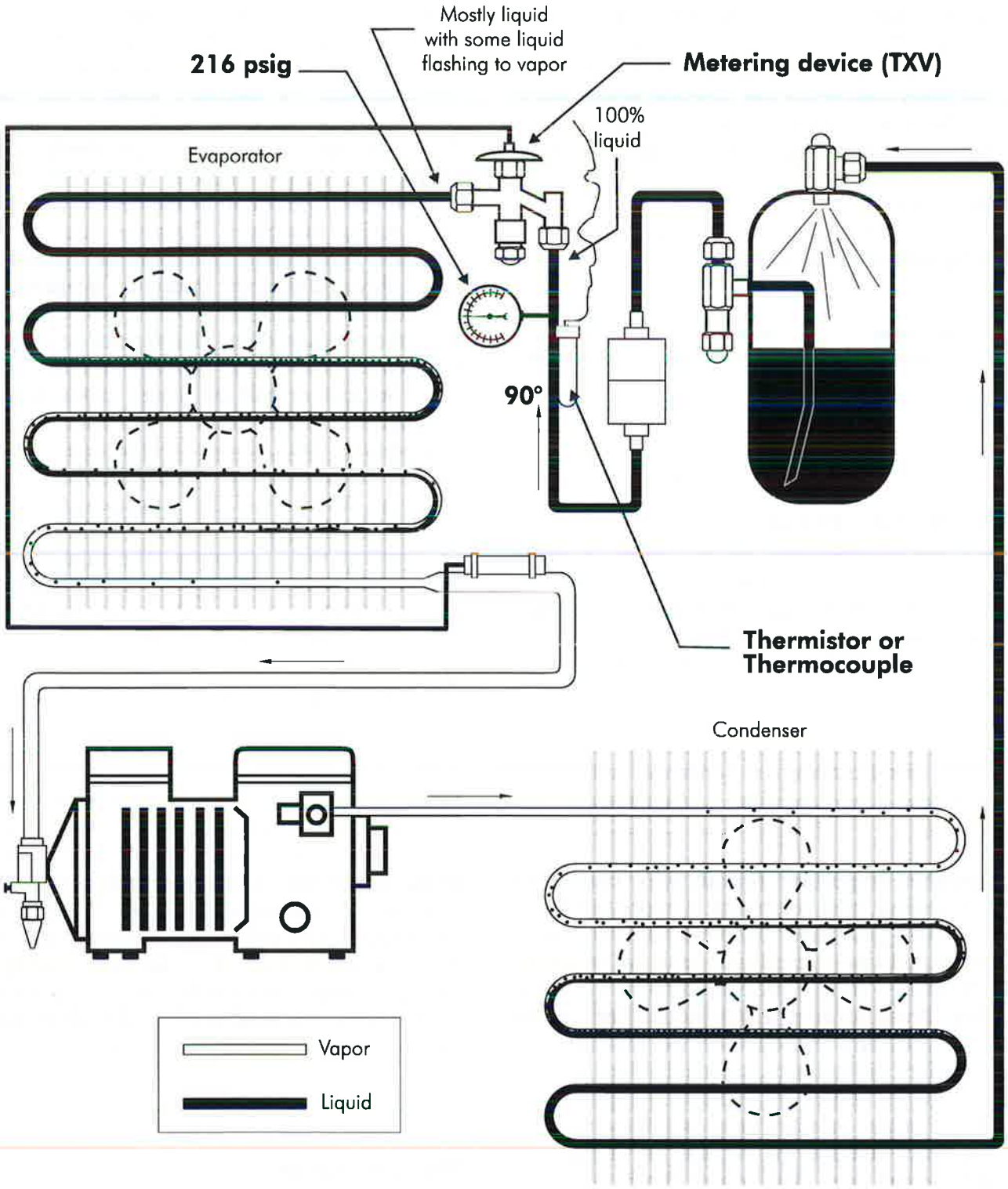


Figure 2-4. Thermistor or thermocouple and gauge at TXV inlet

either through tubing, driers, elbows, or even a sightglass on its way to the metering device. Some of the fluid energy is used to overcome friction from the side walls of the pipe or from an obstruction such as the filter drier. A velocity profile is set up in the pipe with the slowest moving fluid at the sidewalls and the fastest moving fluid in the center of the pipe, Figure 2-5. This lost energy in the fluid causes a loss in pressure, which is why liquids must be subcooled before experiencing these pressure drops to prevent liquid flashing. Proper line sizing, short runs, and unobstructed accessories like clean filter driers can prevent excessive friction pressure drops.

Static Pressure Losses

Static pressure losses occur in the liquid line whenever liquid has to flow uphill through vertical lifts. A vertical column of liquid naturally has more pressure at its bottom than its top due to the weight of the entire column of liquid at the bottom, Figure 2-6. Whenever liquid from a condenser or receiver must be piped uphill, there will be static pressure drops as the liquid climbs higher in the piping. This is in addition to the friction pressure losses from the pipe sidewalls as the liquid climbs. See Table 2-2 for static pressure losses corresponding to vertical lift for five different refrigerants. Refrigerant line sizing does not reduce or increase static pressure losses from vertical lift.

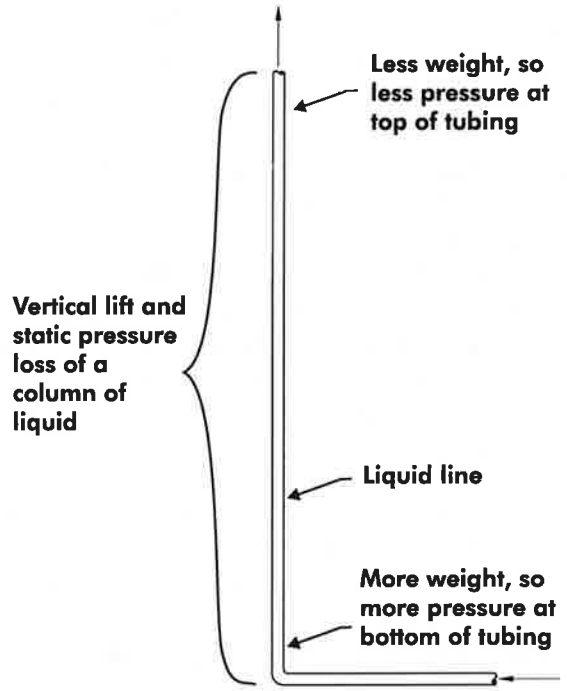


Figure 2-6. Vertical column of liquid showing static pressure losses

REFRIGERANT	VERTICAL LIFT — FEET				
	20	40	60	80	100
	STATIC PRESSURE LOSS — psi				
12	11	22	33	44	55
22	10	20	30	39	49
134a	10	20	30	40	50
502	10	20	30	40	50
717 (Ammonia)	5	10	15	20	25

Table 2-2. Static pressure losses as a function of vertical lift for five refrigerants (Courtesy, Sporlan Valve Company)

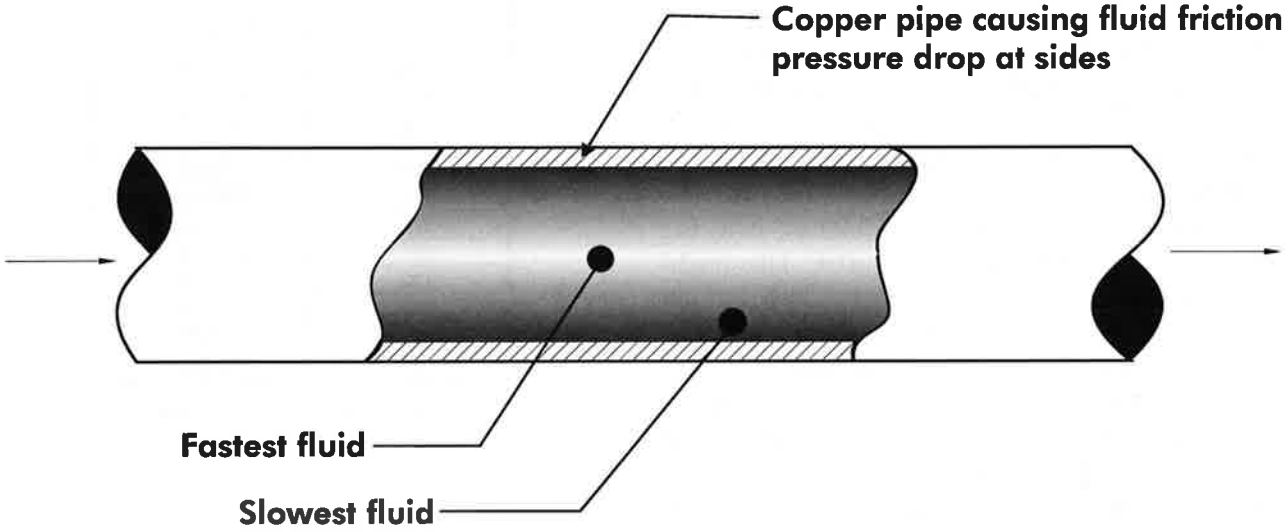


Figure 2-5. Velocity profile

How Much Subcooling Is Needed?

Subcooling can be accomplished by numerous methods. Common methods include liquid/suction line heat exchangers, ambient air, and direct expansion mechanical subcooler heat exchangers. On receiverless systems, simply adding a little more refrigerant to the system can increase subcooling, but this is usually not recommended because of added inefficiencies that may occur. Increasing the pressure of the liquid line is another popular method to subcool liquid and prevent flash gas. This method will be covered later in this chapter.

The amount of total subcooling needed is system dependent. The more pressure drops (friction and static) associated with the lines and accessories that carry the liquid in the system, the more need there is for liquid subcooling to prevent liquid line flash gas. For example, assume an R-12 refrigeration system is operating at a condensing pressure of 127 psig (105°F), Table 2-3. Also assume that all of the vapor in the condenser has turned to liquid and that it has reached the 100% saturated liquid point at 127 psig (105°F). For this example, assume no liquid subcooling. Now assume that this same saturated liquid experiences total pressure drops of 10 psig before it reaches the TXV. Because the liquid is saturated, it must eventually reach a new saturation pressure of 117 psig (127 psig - 10 psig). There is a saturated temperature associated with all saturated pressures, and for the saturated pressure of 117 psig, the new saturated temperature is 100°F (Table 2-3). However, for liquid to drop in temperature from 105° to 100°F, some heat must be removed from the liquid, which is where liquid line flash gas appears. As the liquid gradually travels through the 10 psig pressure drop, a portion of the liquid will continually flash and cool the column of liquid immediately behind it to the new saturation temperature of 100°F. This flashing cools the saturated liquid to 100°F, which corresponds to 117 psig at the entrance of the metering device. As mentioned earlier, premature liquid flashing that does not take place in the evaporator is lost refrigeration effect and can also derate the tonnage of expansion valves. The only sure way to safeguard against liquid

Temperature (°F)	Pressure (psig)
70	70.192
71	71.520
72	72.863
73	74.222
74	75.596
75	76.986
76	78.391
77	79.813
78	81.250
79	82.704
80	84.174
81	85.66
82	87.16
83	88.68
84	90.22
85	91.77
86	93.34
87	94.93
88	96.53
89	98.15
90	99.79
91	101.45
92	103.12
93	104.81
94	106.52
95	108.25
96	110.00
97	111.76
98	113.54
99	115.34
100	117.16
101	119.00
102	120.86
103	122.74
104	124.63
105	126.55
106	128.48
107	130.43
108	132.41
109	134.40
110	136.41
111	138.44
112	140.49
113	142.57
114	144.66
115	146.77
116	148.91
117	151.06
118	153.24
119	155.43
120	157.65
121	159.89
122	162.15
123	164.43
124	166.73
125	169.06

Table 2-3. Pressure/temperature relationship for R-12

line flash gas is to subcool the liquid before it experiences any pressure drop.

Example 2-5
Based on the R-12 example just discussed, figure out how much liquid subcooling needs to be present to prevent liquid line flash gas.

Solution 2-5
Refer to Table 2-3 to find the temperature span of 5°F that covers the pressure drop of 10 psig (127 psig - 117 psig). If there is at least 5°F of subcooling in the system, it acts as a sensible heat cushion to prevent liquid line flash gas. If the liquid is subcooled 5°F and experiences the gradual 10 psig pressure drop, flashing does not occur, because the liquid is already subcooled to accommodate the new lower pressures. When subcooled, the liquid is already at or below the saturation temperature for the new pressures. The *minimal amount* of subcooling required to accommodate the 10 psig pressure drop without liquid flashing is 5°F.

You can always use a pressure/temperature chart to figure the minimal amount of subcooling for any known pressure drop and refrigerant. However, different refrigerants require different amounts of liquid subcooling for the same pressure drops, Table 2-4. If charts such as Table 2-4 do not exist, calculations must be performed to find the amount of subcooling needed to handle whatever pressure drops exist. If the pressure drop is not known, make sure there is enough subcooling to handle an extreme pressure drop. Do not overcharge the system by adding too much refrigerant on receiverless systems. Overcharging increases condenser subcooling, but the head pressure will elevate and give unwanted inefficiencies from higher compression ratios. Always consult the system manufacturer for the proper amount of condenser subcooling if charging to this specification.

Example 2-6
Apply the principles just learned to an R-22 system piping layout, Figure 2-7. Assume the following conditions:

- Design pressure drop for friction losses in liquid line is 2 psig for copper pipes only
- Filter drier pressure drop is 1 psig (friction pressure drop)
- Vertical lift of liquid is 40 feet (static pressure drop)
- Static pressure drop is 20 psig (Table 2-2)
- System has a condensing pressure of 211 psig (105°F)

What is the *minimal* amount of liquid subcooling required to prevent liquid line flash gas in this R-22 refrigeration system?

Solution 2-6
The total pressure drop for this system is 23 psig (2 psig + 1 psig + 20 psig). The 23 psig pressure drop puts the new pressure at 188 psig (211 psig - 23 psig). According to Table 2-1, the saturation temperature that corresponds to 188 psig is about 97°F. Therefore, the liquid must be subcooled to at least 97°F to prevent liquid line flash gas. This means the system must have at least 8°F (105° - 97°F) of subcooling to prevent flash gas.

REFRIGERANT	100° F. Condensing					
	PRESSURE LOSS — psi					
	5	10	20	30	40	50
	REQUIRED SUBCOOLING — °F.					
12	3	6	12	18	25	33
22	2	4	8	11	15	19
500	3	5	10	15	21	27
502	2	3	7	10	14	18
717 (Ammonia)	2	4	7	10	14	17
REFRIGERANT	130° F. Condensing					
	PRESSURE LOSS — psi					
	5	10	20	30	40	50
	REQUIRED SUBCOOLING — °F.					
12	3	5	9	14	18	23
22	2	4	6	9	12	14
500	2	4	8	11	15	19
502	1	3	5	8	11	13
717 (Ammonia)	2	3	5	7	10	12

Table 2-4. Required subcooled liquid amounts to compensate for pressure drops (Courtesy, Sporlan Valve Company)

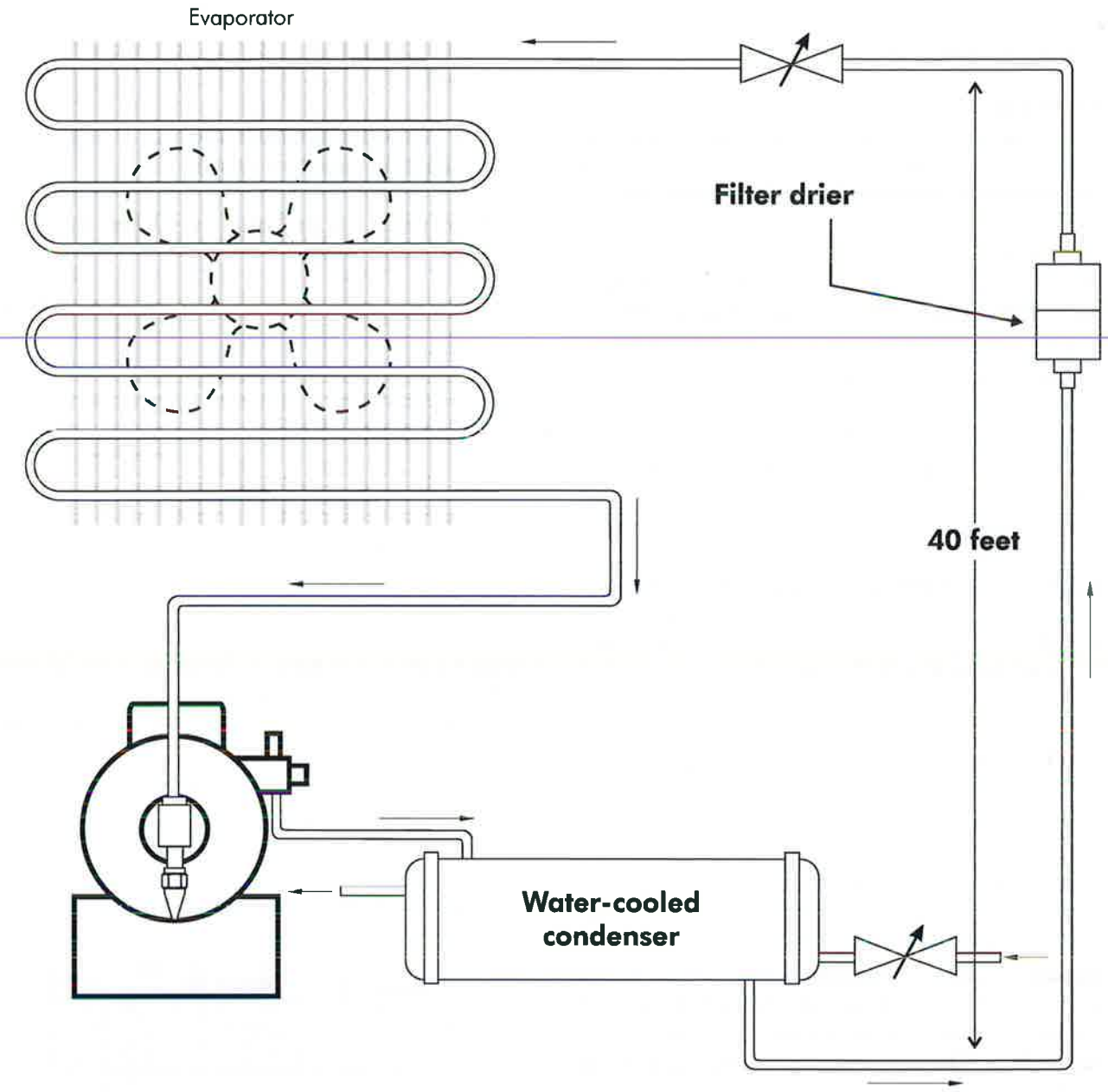


Figure 2-7. Refrigeration system with a 40-ft vertical rise in the liquid line

Liquid Pressure Amplification™ (LPA) and Superheat Suppression

Liquid pressure amplification technology[1] has given refrigeration systems capacity boosts while saving electrical energy. Using this technology, liquid refrigerant entering the liquid line is pressurized by a small centrifugal pump, Figure 2-8. The pressurized amount is equivalent to the pressure loss between the condenser outlet and the TXV inlet on receiverless systems, or the receiver and the TXV inlet on TXV/receiver systems. By increasing the pressure of the liquid refrigerant, the associated saturation temperature is raised, while the actual liquid temperature remains the same. The liquid becomes subcooled and will not flash if exposed to pressure drops in the liquid line. Because the liquid centrifugal pump motor is external to the refrigeration system and the impeller is driven by a revolving magnetic field, negligible energy and heat are added to the system. The

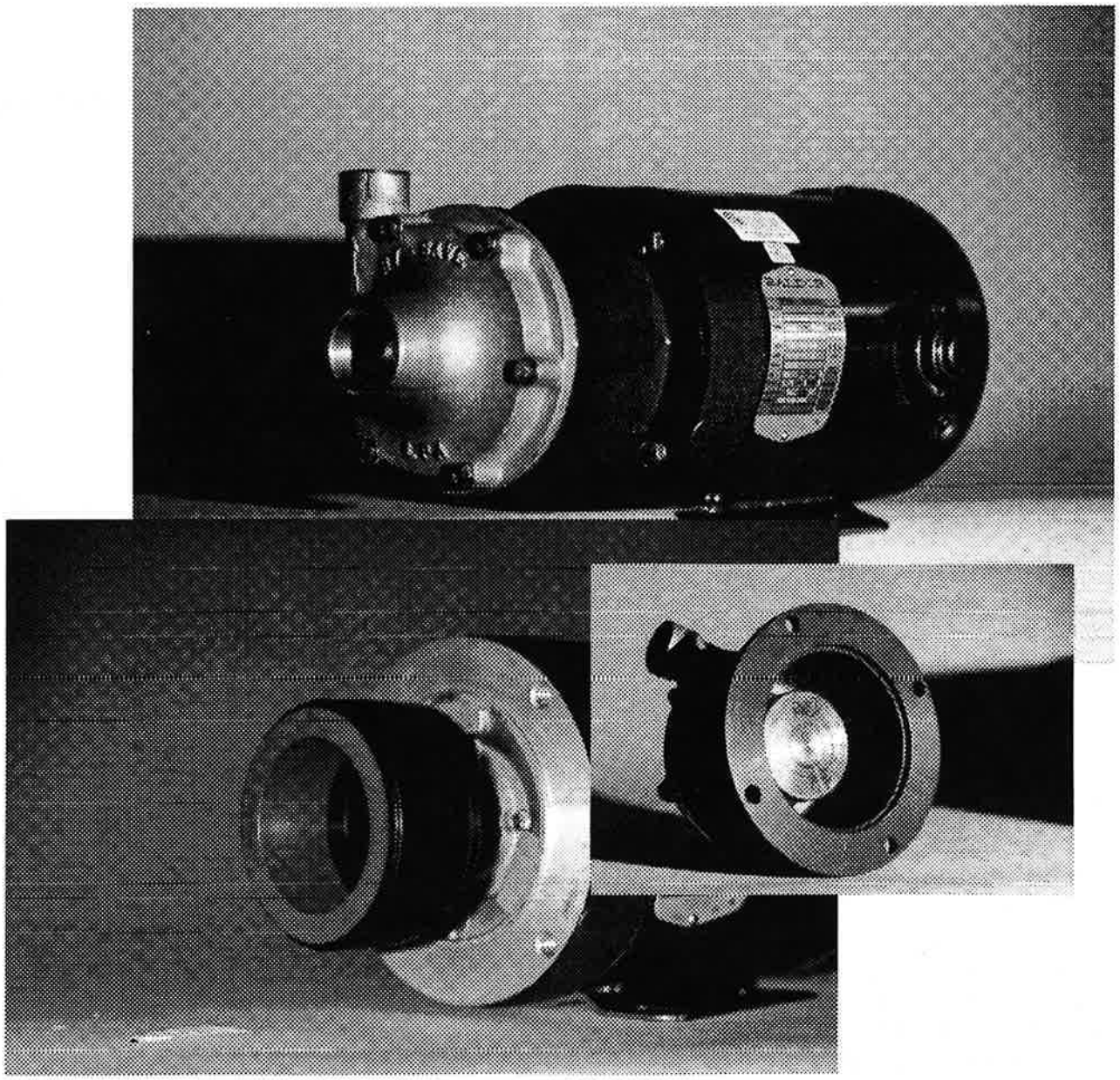


Figure 2-8. Centrifugal pump for liquid pressure amplification™ (Courtesy, Hy-Save, Inc.)

liquid is pressurized with a negligible addition of temperature or heat and allows for a completely sealed system, because there is no drive shaft protruding from the motor to the pump. The LPA centrifugal pump can increase the pressure of the liquid by approximately 8 to 20 psi. The liquid pressure amplification system is shown in Figure 2-9 and can be compared to a more common system shown in Figure 2-10.

Since subcooling exists in any liquid below its saturation temperature for a given pressure, there are really several ways to subcool liquid. One way is to sensible cool the liquid in the bottom of the condenser, giving it a sensible heat reduction to prevent flashing from liquid line pressure drops. However, subcooling this way will take up valuable condenser volume with subcooled liquid at its bottom, since condensing cannot occur in this area. This will cause higher head pressures and compression

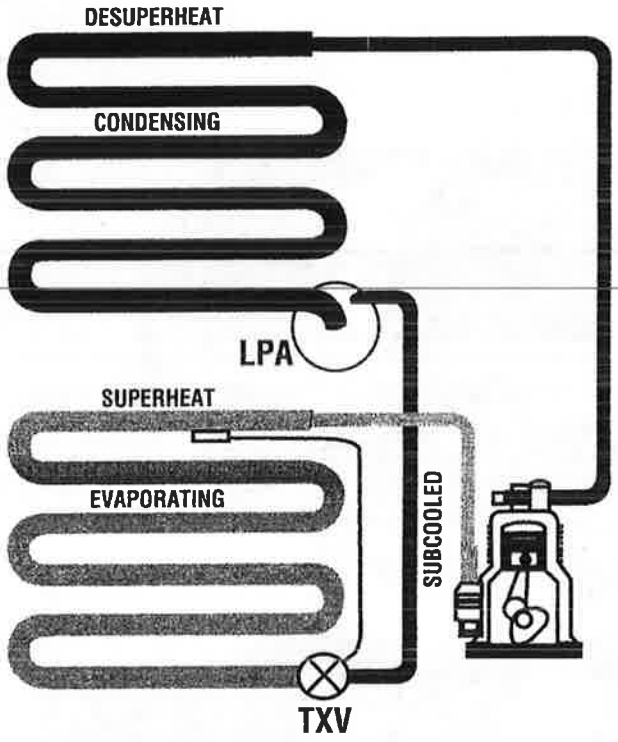


Figure 2-9. Normal refrigeration system with centrifugal pump for liquid pressure amplification (Courtesy, Hy-Save, Inc.)

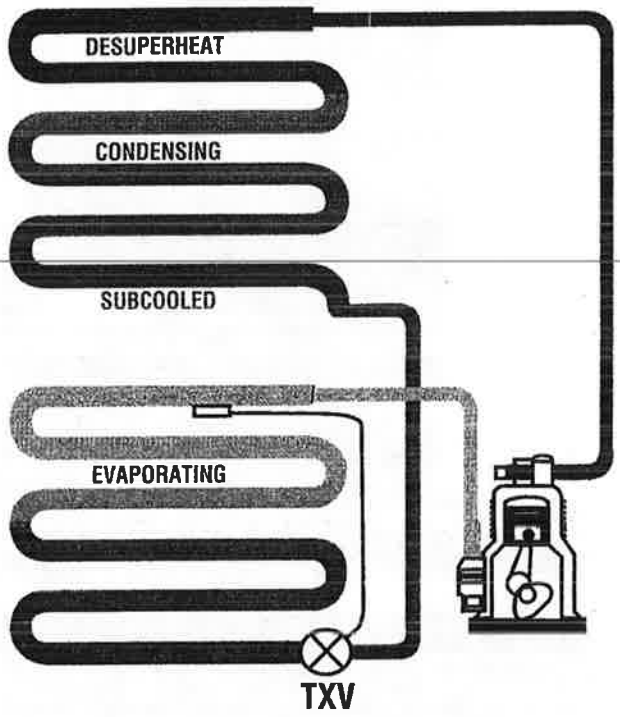


Figure 2-10. Typical refrigeration system (Courtesy, Hy-Save, Inc.)

ratios, thus lower efficiencies. To only consider the subcooling of the liquid without regard to decreasing condenser surface area, there will be a gain of 1/2% of capacity for every degree of liquid subcooling. Also, 7°F of ambient subcooling costs .167 brakehorsepower per ton because of elevated head pressures. However, if we consider the reduction of effective condensing surface area due to the condenser flooding necessary to achieve liquid subcooling, there is a net loss in capacity due to increased condensing pressures and temperatures.

This type of subcooling, often called ambient subcooling, has been practiced for years and was thought to be a "free" method of subcooling. This simply is not the case with receiverless systems having no separate subcooling coil. Ambient subcooling is usually accomplished at the cost of increased head pressures. It was used in refrigeration systems simply as a liquid seal in the condenser bottom and to prevent liquid line flash gas. This kept a solid column of liquid supplied to the metering device.

Ambient subcooling cannot be maintained at a given level with only air side controls. Condensing pressures are directly related to the temperature of the condenser cooling medium and the useful condensing area in the condenser. Useful or effective condensing area is the total condensing area minus the area used for desuperheating and the area used for subcooling:

Total condenser area	−	Condenser area used for desuperheating and subcooling	=	Useful or effective condensing area

[Formula 2-3]

As one can see, the more desuperheating and liquid subcooling that is done by the condenser, the less useful condenser area there is. This will raise condensing pressures and compres-

sion ratios and cause inefficiencies with higher power draws.

A more efficient way to subcool liquid usually is to increase the pressure of the liquid without raising the temperature. This puts the liquid at a higher pressure, thus it has a higher associated saturation temperature but does not change its actual temperature. As the subcooling definition states, this liquid is subcooled in an amount equal to the difference between the saturation temperature and the actual temperature, and the liquid is below its saturation temperature for that new pressure. By increasing the pressure of the subcooled liquid to overcome any pressure losses that occur in the liquid line, condensing pressures can be allowed to fall to their lowest pressures attainable without flash gas developing in the liquid line.

Another term for attaining the lowest possible head pressure is *floating the head pressure*. Condensing temperatures of 20°F are not uncommon in low temperature systems incorporating LPA. However, if one tries to float the head pressure with the ambient, these lower head pressures will require more subcooling for the same pressure drops in the liquid line in order to prevent flashing. This phenomenon happens because the refrigerant pressure/temperature graph is non-linear. The pressure/temperature graph is much flatter at the lower pressures, meaning that the same amount of liquid subcooling is needed to overcome less of a pressure drop at these lower pressures and temperatures, Figure 2-11. This is one of the reasons why LPA is incorporated in the system when the head pressure is floated with the ambient. It subcools the liquid by increasing the pressure of the liquid and forces the liquid to have a new higher saturation temperature. Thus, flash gas is prevented when head pressures are allowed to float, because the LPA ensures that the liquid line pressure (and saturation temperature) are always higher than the actual liquid temperature.

If head pressure is reduced without the LPA and the liquid experiences the same pressure drop through the liquid line as it did at the higher condensing pressures, the flash gas will occupy more volume in the liquid line because of the higher specific volume of the flashed vapors. The TXV will begin to hunt, alternately overfeeding and underfeeding the evaporator. The system capacity will be reduced, and damage to the compressor may result.

For example, consider the curve for a CFC refrigerant, Figure 2-12. As the pressure in the liquid line drops, more liquid progressively flashes into vapor to cool the remaining liquid to the saturation temperature corresponding to the progressively lower pressure. With an 8 psig pressure drop, the flash gas by weight is 2% with a 100°F condensing temperature (214 psig). The vapor bubbles in the liquid line are very compressed and occupy only 20% of the volume in the liquid line. However, reduce the pressure to 97 psig (50°F), and the flashing vapor will occupy 38% of the liquid line volume. This vapor reduces the flow through the expansion valve, has little refrigeration effect, and must be recompressed after doing work. Again, system capacity will suffer, the evaporator will starve, and the TXV will begin to hunt. This is the primary value of the LPA system - to ensure a solid column of liquid to the TXV, so the TXV can supply adequate liquid to the evaporator.

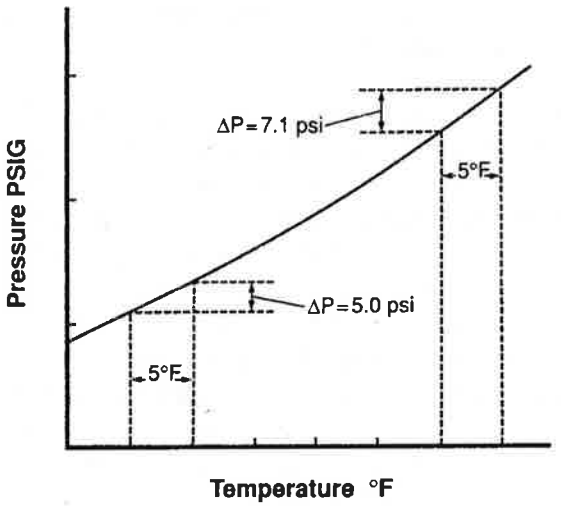


Figure 2-11. R-22 pressure/temperature curve (Courtesy, Sporlan Valve Company)

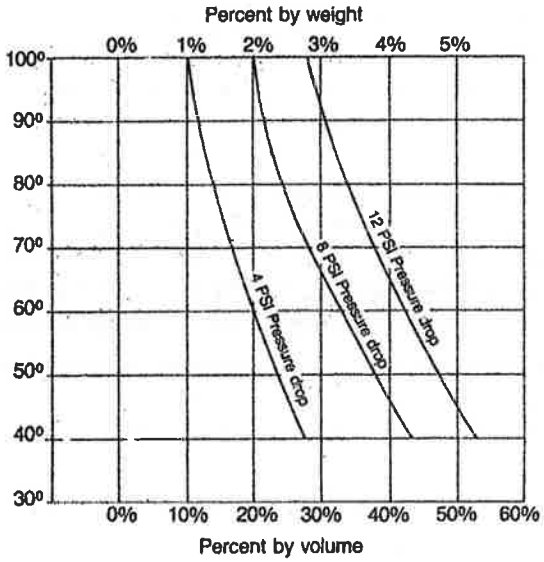


Figure 2-12. Effects of condensing pressure on flash gas (Courtesy, Hy-Save, Inc.)

Floating Head Pressures and LPA Advantages

In the past, designers of air conditioning and refrigeration systems picked an outdoor design condition for the system. This outdoor design condition typically was a temperature that would not be reached any more than 2% of the time in the life of the system. This design condition would also occur only a couple of hours at a time when reached. The selection of the condenser was then made based on this seldom-reached condition.

This was acceptable several years ago when energy was much cheaper, and designers could select condensing temperatures at 20° to 30°F above the ambient. This was done because it was thought the higher condensing temperatures and pressures would enhance the flow through the metering device to outweigh any inefficiencies from the high compression ratios. This would force condensing temperatures and pressures higher, causing high compression ratios and lower efficiencies. With today's escalating energy costs, designers are specifying larger condensers with condensing temperatures 10° to 15°F above the ambient. The significant energy savings from lower compression ratios and possibly increased subcooling of liquid negate the higher costs of the larger condenser.

After much research with metering valve suppliers, it was discovered that TXVs would work with much less pressure drop across them than expected in the past, as long as pure liquid was supplied to them. The balanced port TXV design today is noted for its low pressure drop performance. With this new knowledge, condensing pressures and temperatures were allowed to "float" downward with the ambient temperature. In fact, a majority of the outdoor temperatures in the USA are below 70°F more than they are above. The compressor capacity increases about 6% for every 10°F drop in condensing temperature.

Pressure drops of less than 30 psig across the expansion valve should be avoided for proper evaporator feeding of liquid. For an evaporator to operate at peak efficiency, it must have a high percent of liquid to vapor ratio entering the evaporator. To accomplish this, the expansion valve must allow liquid refrigerant to enter the evaporator at the same rate that it evaporates. With an LPA pump, subcooling and pressure can be maintained at a constant at the metering device. Overfeeding and underfeeding by the expansion valve, which dramatically affect the efficiency of the evaporator, will then be minimized.

Historically, high head pressures and temperatures were artificially maintained in a refrigeration system so it would function well at low outdoor temperatures. These higher pressures were considered mandatory in order for the TXV to feed the evaporator adequately. Power costs 50 years ago were not the primary consideration, so the added cost of lower efficiency did not matter. At today's power costs, inefficiency is an unacceptable part of a company's overhead. LPA allows lowering of the head pressure and reduced power consumption along with higher efficiencies. Here are four advantages to having an LPA system:

- Eliminates liquid line flashing by overcoming line pressure losses

- Reduces energy costs because pumping liquid refrigerant is up to 40 times more efficient than using head pressure from the compressor to do the same work
- Increases evaporator capacity along with the net refrigeration effect
- Lowers compression ratios and reduces stress on compressors, resulting in longer compressor life

The days of the fixed elevated head pressures are fading. No longer are consumers willing to pay for inefficiencies. Customers are looking at life cycle costs, which are the equipment costs plus the operational costs for the life of the equipment.

Superheat Suppression

Superheat suppression can be used in conjunction with LPA. The superheat suppression process injects liquid refrigerant into the compressor discharge line or inlet of the condenser, Figure 2-13. This liquid usually comes from the same centrifugal pump used in the LPA process (see Figure 2-9). This liquid flashes to vapor while cooling the superheated discharge gas closer to its condensing temperature. As a result, less surface area is required for desuperheating. This leaves a more efficient condenser because of the increase in useful condensing surface area, which increases the overall performance of the system. Savings from 6% to 12% can be realized with superheat suppression.

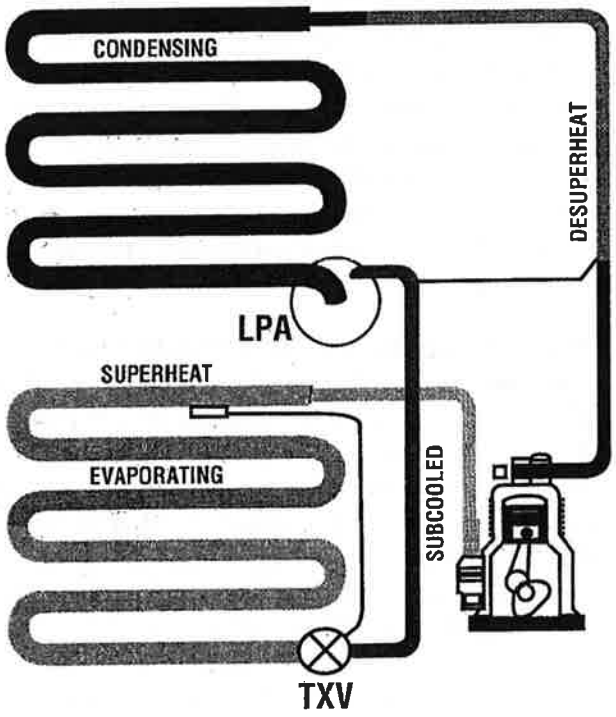


Figure 2-13. System with LPA and superheat suppression (Courtesy, Hy-Save, Inc.)

Superheat suppression processes have been used in large ammonia plants for years. However, the process was not usually feasible on smaller systems. This spurred the use of LPA and superheat suppression on the same system. A small portion of the pressurized liquid refrigerant provided by LPA (centrifugal pump) is diverted to the compressor discharge line to cool the superheated vapors coming from the compressor. Reduced superheat of the gas entering the condenser results in higher condenser efficiency, a lower condensing temperature, and greater compressor efficiency. Although some efficiency gains are seen at low ambients, the greatest gain with superheat suppression is realized at the higher ambient temperatures.

Suppression or cooling of superheated vapors in a refrigeration system has four main advantages:

- Reduced superheated vapor heat intensity (temperature), so the pressure and volume of these superheated vapors will decrease
- Faster saturation of superheated vapors, which condense more quickly
- Condensing of the vapors occurs closer to the inlet of the condenser, resulting in lower condensing temperatures and possibly more ambient subcooling
- Higher overall condenser heat transfer because of the increased liquid/vapor mixture heat transfer and increased effective condensing area

LPA/Superheat Suppression with Subcooling/Reheat Coil

In an air conditioning system, the evaporator removes moisture from the air. Therefore, it would seem reasonable that if the efficiency of the evaporator were increased, more moisture could be removed. Part of dehumidification is reheating the air that comes out of the evaporator to a lower relative humidity and raising the temperature to an acceptable delivery level. Accomplishing both of these functions greatly enhances the desired operation. Figure 2-14 shows a combination LPA/superheat suppression system incorporating a subcooling/reheat coil at the evaporator air outlet.[2] The coil allows subcooled liquid to be further subcooled, while at the same time providing reheat to the air leaving the evaporator. The relative humidity of the air is thus lowered. Depending on the size of the coil, the liquid should be able to be subcooled to within 8°F of the air temperature leaving the evaporator. With 60°F leaving air, the liquid should be subcooled to approximately 68°F. Remember, subcooling reduces liquid flash loss in the evaporator creating a greater net refrigeration effect.

With the rapid transition to alternative refrigerants and escalating energy costs facing the refrigeration industry today, more attention is being paid to operating costs and system efficiencies. Inattention to system pressures and liquid subcooling can make systems very inefficient. Because of this, it is very important that every technician understands the principles behind subcooling and system pressures.

ENVIROGUARD™ REFRIGERANT CONTROL SYSTEM

Enviroguard is a refrigerant control system that allows head pressures to float with the ambient temperature, while reducing system refrigerant charge up to one-third when compared to conventional systems.[3] Key features and benefits of this system include the following:

- Reduces system refrigerant charge up to one-third
- Reduces refrigerant leak liabilities compared to conventional systems
- Provides early warning of dirty condenser or failed condenser fans
- Provides early warning of a system refrigerant leak
- Increases compressor life because of lowered compression ratio and discharge temperatures due to lower condensing pressures when operated in cooler ambients
- Increases volumetric efficiencies due to lowered compression ratios

Principles of Operation

The amount of liquid refrigerant used in the Enviroguard system is controlled by a system pressure regulator (SPR), Figure 2-15. The system refrigerant charge is reduced by taking the receiver out of the active refrigerant circuit and allowing liquid refrigerant to return directly to the liquid manifold, which feeds the branch refrigeration circuits. With the receiver out of the active refrigeration circuit, a minimum receiver charge is no longer required as with conventional system design. The receiver is now only used as a storage vessel to store the condenser charge variations between summer and winter operations.

The SPR is controlled by a pilot pressure from a remote mounted, refrigerant filled, ambient air sensor connected to the SPR by a pilot line. The SPR's remote sensing bulb is charged with the same refrigerant as the system. The remote mounted ambient air sensor is located under the air-cooled condenser to sense the ambient air temperature entering the condenser, Figure 2-16. Figure 2-17 illustrates the location of the remote sensor in the evaporative condenser sump if an evaporative condenser is used.

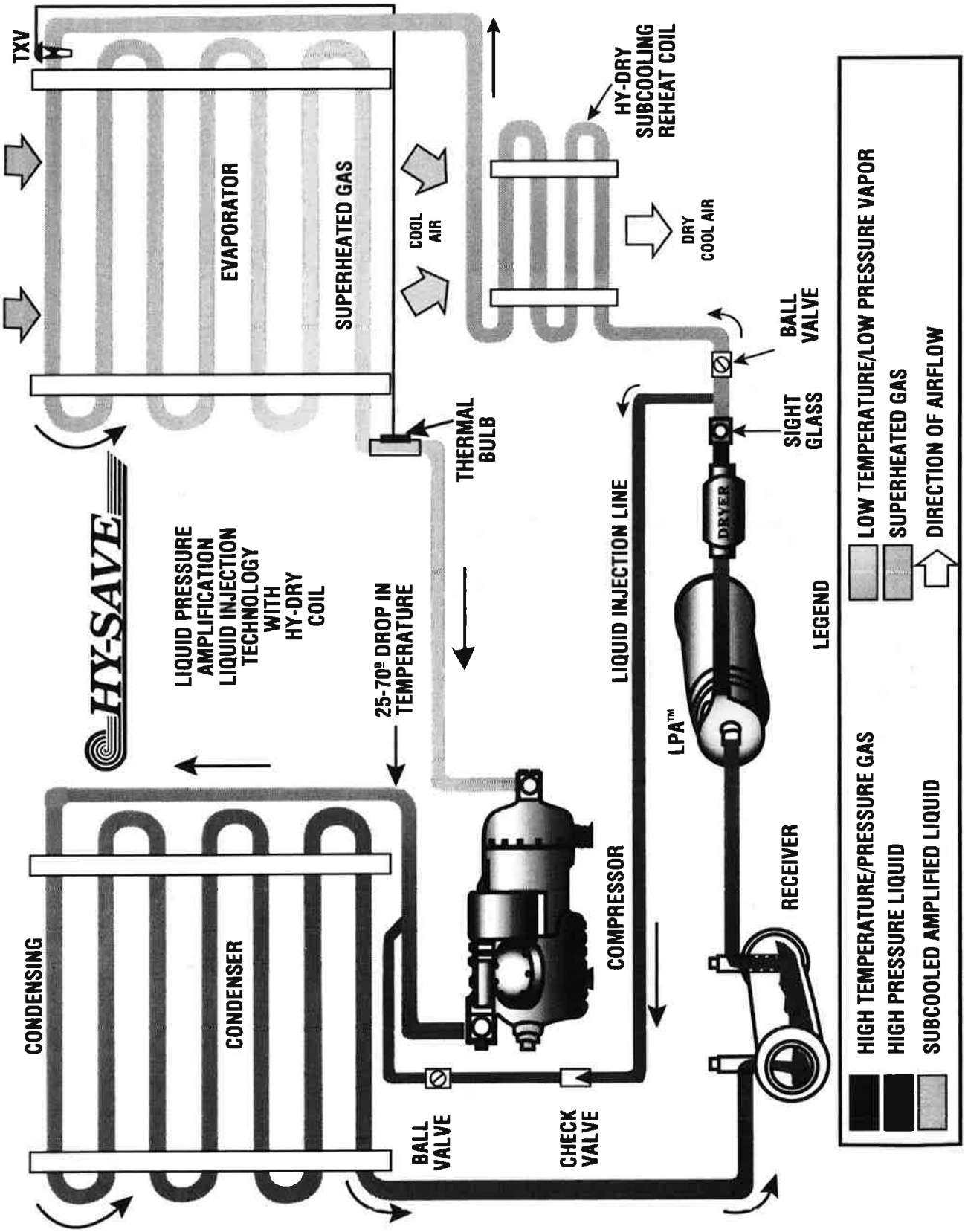


Figure 2-14. Combination LPA/superheat suppression system with a subcooling/reheat coil (Courtesy, Hy-Save, Inc.)

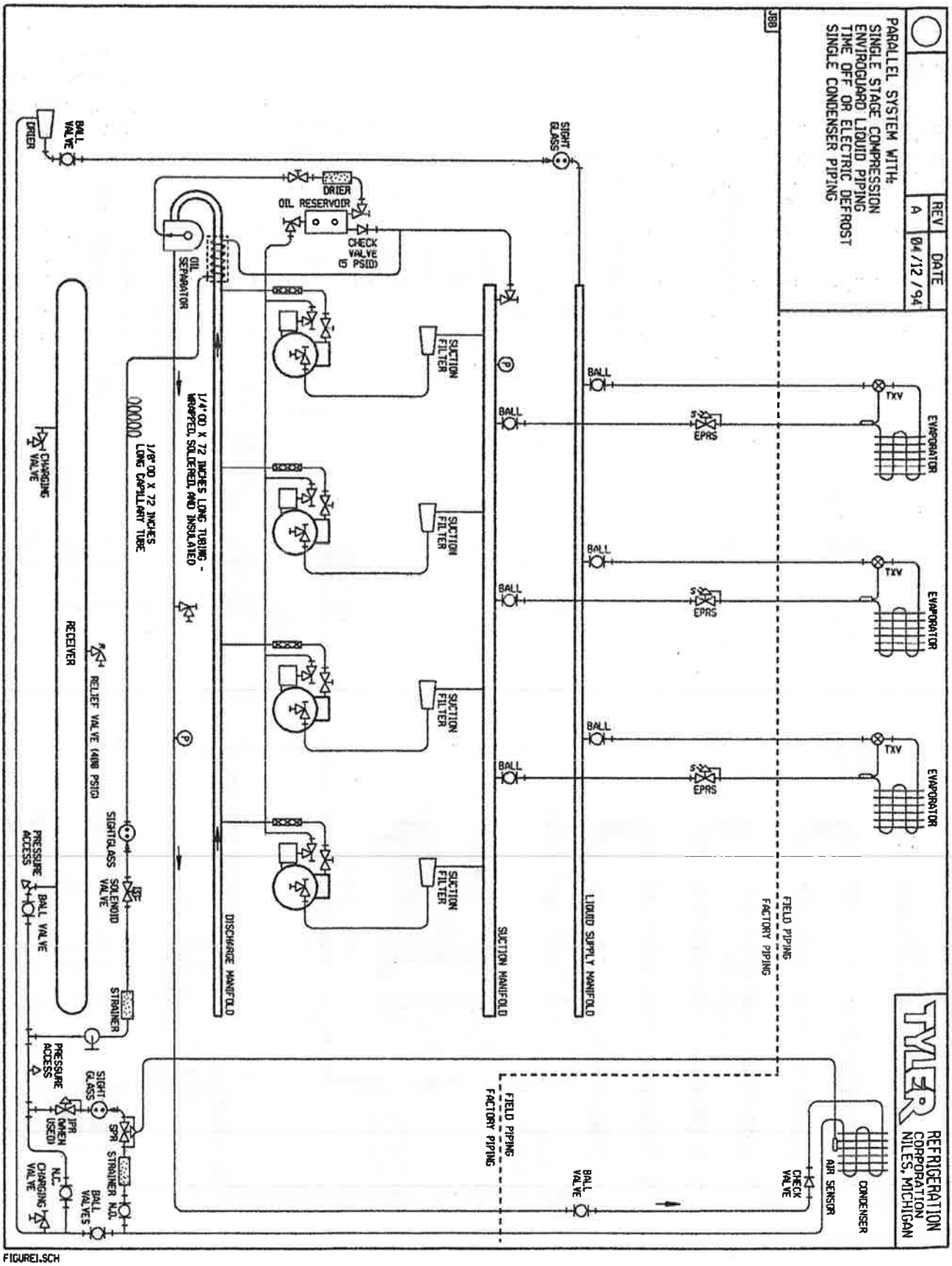


Figure 2-15. Enviroguard piping diagram (Courtesy, Tyler Refrigeration Corporation)

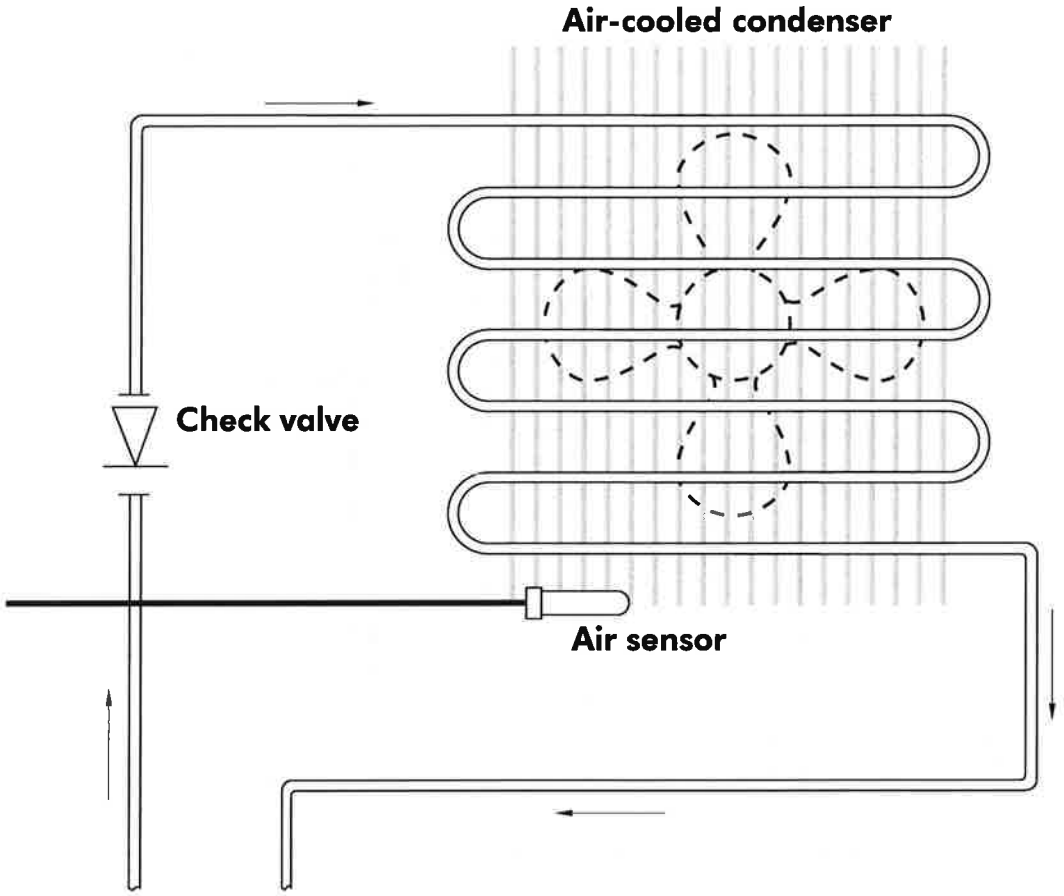


Figure 2-16. Condenser with air sensor for SPR valve (Courtesy, Tyler Refrigeration Corporation)

When the ambient air temperature rises, the pressure inside the sensor and pilot line rises, exerting a higher pressure in the pilot line and diaphragm of the SPR valve. A spring with an adjustable resistance (differential) counteracts the SPR's diaphragm pressure. Diaphragm force against spring force determines the position of the SPR valve, and therefore, how much refrigerant will be bypassed to the receiver. When the ambient air temperature falls, the pressure inside the sensor and pilot line also falls, exerting a lower pressure on the pilot and diaphragm of the SPR. The SPR setting is adjusted to achieve a differential of approximately 45 psi for R-22 low temperature air-cooled applications and approximately 61 psi for R-22 medium temperature air-cooled applications. Since the SPR pilot pressure is equivalent to the saturated refrigerant pressure at the ambient temperature, the pressure at which the SPR begins to bypass refrigerant into the receiver on an R-22 low temperature system is the sum of the saturated refrigerant pressure corresponding to the actual ambient air temperature plus the appropriate differential pressure setting of either 45 or 61 psi.

Whenever the ambient air temperature drops, the pressure setting at which the SPR bypasses liquid refrigerant to the receiver also drops relative to the ambient air temperature. Anytime the condensing pressure rises 45 or 61 psi above the corresponding ambient air sensor pressure, the SPR begins to bypass refrigerant into the receiver. Condensing pressure changes occur at the same rate and time relative to changes in the ambient air temperature so liquid feed is always constant.

If the condenser should become damaged, fouled internally or externally, or should it lose a fan motor, an elevated condensing pressure

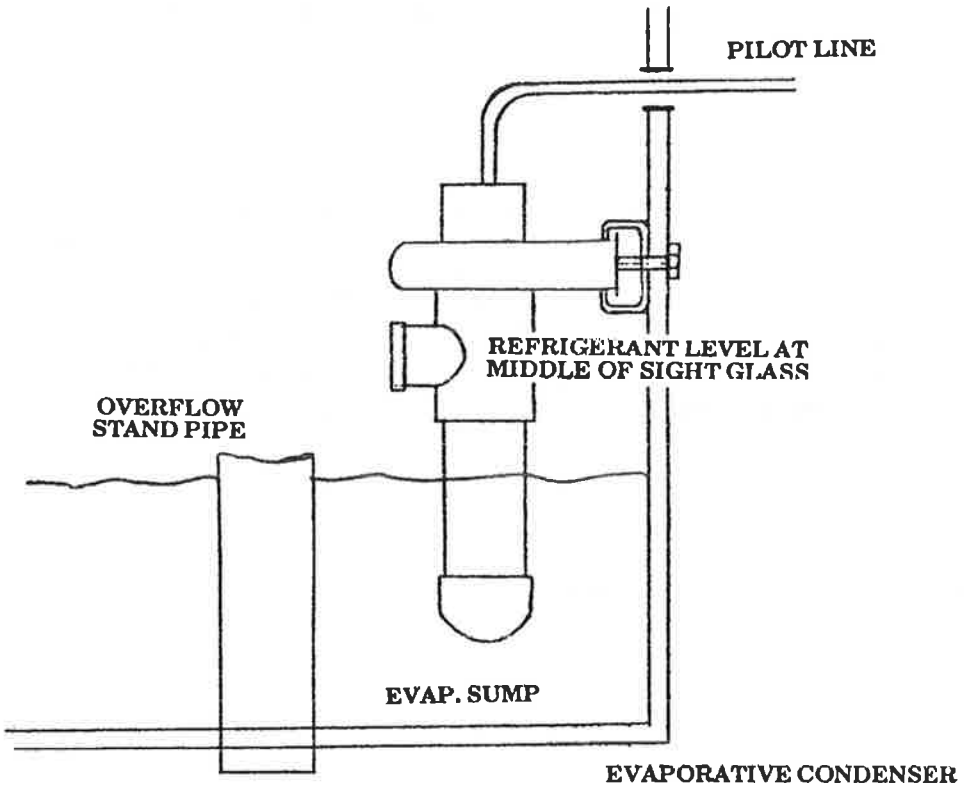


Figure 2-17. Remote sensor in the evaporative condenser sump (Courtesy, Tyler Refrigeration Corporation)

will occur, resulting in refrigerant being bypassed into the receiver by the SPR valve. This will prevent high compression ratios and high discharge temperatures with lower volumetric efficiencies. These occurrences often go unnoticed for long periods of time and stress compressor systems. Eventually, branch circuit evaporator temperatures will rise, because refrigerant is being bypassed out of the working part of the system into the receiver, simulating a refrigerant starved system. Rising evaporator and product temperatures will be noticed sooner, and the problem can be remedied. If a refrigerant leak should occur in the system, it will also be noticed earlier because of higher evaporator temperatures. Overall, less refrigerant is lost to the atmosphere than with conventional system design, because system problems are noticed much faster.

Whenever any or all of the compressors are running, there is a bleed circuit that is opened to the suction manifold, which bleeds from the receiver back to the system for use. This allows the refrigerant working charge in the system to seek its own level of equilibrium relative to the ambient temperature. Because of this bleed circuit, certain conditions may allow the receiver to be void of refrigerant and at suction pressure.

As refrigerant bleeds from the receiver to the suction manifold, a capillary restrictor tube and discharge line heat exchanger vaporize any liquid (see Figure 2-15). This prevents liquid from entering the suction manifold. This same refrigerant is also strained and filtered before entering the suction manifold. For example, on most systems, when the ambient air temperature is below 70°F, the receiver is empty because refrigerant is flooding the condenser. This is because less condenser surface is required during winter operation due to lower ambient air temperatures. During typical system operation when the ambient air temperature is above 70°F, part of the refrigerant charge normally flooding the condenser is stored in the receiver since more condenser surface is required to reject the total heat of rejection at the higher ambient temperatures.

Determining the SPR Valve Setting

For a forced air condenser, the proper steps to be taken in determining the proper SPR valve setting for a low temperature R-22 system application are as follows:

1.	Choose a design ambient air temperature for the geographical region	95°F
2.	Choose the condenser temperature difference (TD)	10°F
3.	Design in a TD safety factor	5°F
4.	Add the first three together to obtain the adjusted condensing temperature	**110°F**
5.	Corresponding saturated refrigerant pressure equal to the adjusted condensing temperature (110°F)	227 psig
6.	Corresponding saturated refrigerant pressure equal to design ambient air temperature of 95°F (SPR bulb and pilot line pressure at design ambient)	182 psig
7.	Subtract No. 6 from No. 5 for target SPR differential pressure	**45 psig**
8.	Actual ambient air temperature at time of system start-up	60°F
9.	Corresponding saturated refrigerant pressure equal to actual ambient air temperature of 60°F	102 psig
10.	Target SPR differential pressure	45 psig
11.	Add No. 9 and No. 10 for the target SPR bypass pressure	**147 psig**

The actual pressure at which the SPR valve will be set to bypass refrigerant to the receiver for this low temperature R-22 example will be 147 psig with the ambient air temperature measured at 60°F.

For an evaporative condenser, the proper steps to be taken in determining the proper SPR valve setting for a low temperature R-22 system application are as follows:[4]

1. Determine the area design ambient wet bulb temperature.

2. Determine the design wet bulb temperature to refrigerant condensing temperature differences. Typical condensing temperatures for low temperature systems are 90° to 95°F with a 20°F TD. The design condensing temperature for medium temperature applications is

from 95° to 100°F with a 25°F TD. Always refer to manufacturer guidelines.

a)	Design wet bulb temperature for the geographical area	75°F
b)	TD wet bulb to refrigerant	20°F
c)	Add a) and b) for saturated condensing temperature	**95°F**
d)	Corresponding saturation pressure at condensing temperature (95°F)	182 psig
e)	Corresponding saturation pressure at design wet bulb (75°F)	132 psig
f)	Subtract e) from d) to obtain SPR setting required	**50 psig**

SUPERHEAT

Superheat is any heat added to completely saturated vapor that results in a rise in temperature (sensible heat change) of the gas. Superheat is measured as the difference between the actual temperature of refrigerant vapor and the saturation temperature of the refrigerant at that same point. Superheat on the system's low side can be divided into two types: evaporator superheat and total (compressor) superheat.

Evaporator Superheat

Evaporator superheat starts at the 100% saturated vapor point in the evaporator and ends at the outlet of the evaporator, Figure 2-18. Technicians usually put a thermistor or thermocouple at the evaporator outlet to obtain the evaporator outlet temperature. A pressure gauge at the same point as the temperature reading will give the technician the saturated vapor temperature.

A Schrader tee can be used to obtain evaporator outlet pressure. Simply install the Schrader tee into the external equalizer access fitting on the side of the TXV once the system is pumped down, Figure 2-19. Once the pressure gauge and external equalizer line are reconnected to the TXV, the technician should wait from 10 to 15 minutes to let the system reach a running equilibrium again. The evaporator outlet pressure can then be read.

To determine evaporator superheat, consider an R-134a refrigeration system. In this system, the low side gauge reading at the evaporator outlet is 20 psig, or 23°F (see Table 1-1), and the evaporator outlet temperature (thermistor reading) is 30°F. The evaporator superheat calculation is as follows:

30°F (evaporator outlet temperature) − 23°F (saturation temperature at evaporator outlet) = 7°F (evaporator superheat)

If a technician were to measure the pressure at the compressor instead of the evaporator outlet, a higher, fictitious superheat value would be read. As the refrigerant travels the length of the suction line, there is associated pressure drop from friction and/or restrictions. This causes the pressure at the compressor to be lower than the pressure at the evaporator outlet. This higher, fictitious superheat reading may lead the technician to open the TXV more to compensate for the fictitious reading. This could cause compressor damage from liquid flooding or slugging from a superheat setting that is too low.

To illustrate this point, consider the same R-134a refrigeration system just discussed with a 5 psi pressure drop from the evaporator outlet to the compressor. In this system, the low side gauge reading at the compressor inlet is 15 psig or 15°F, and the evaporator outlet temperature (thermistor reading) is 30°F. The evaporator superheat calculation is as follows:

30°F (evaporator outlet temperature) − 15°F (saturation temperature at compressor inlet) = 15°F (superheat)

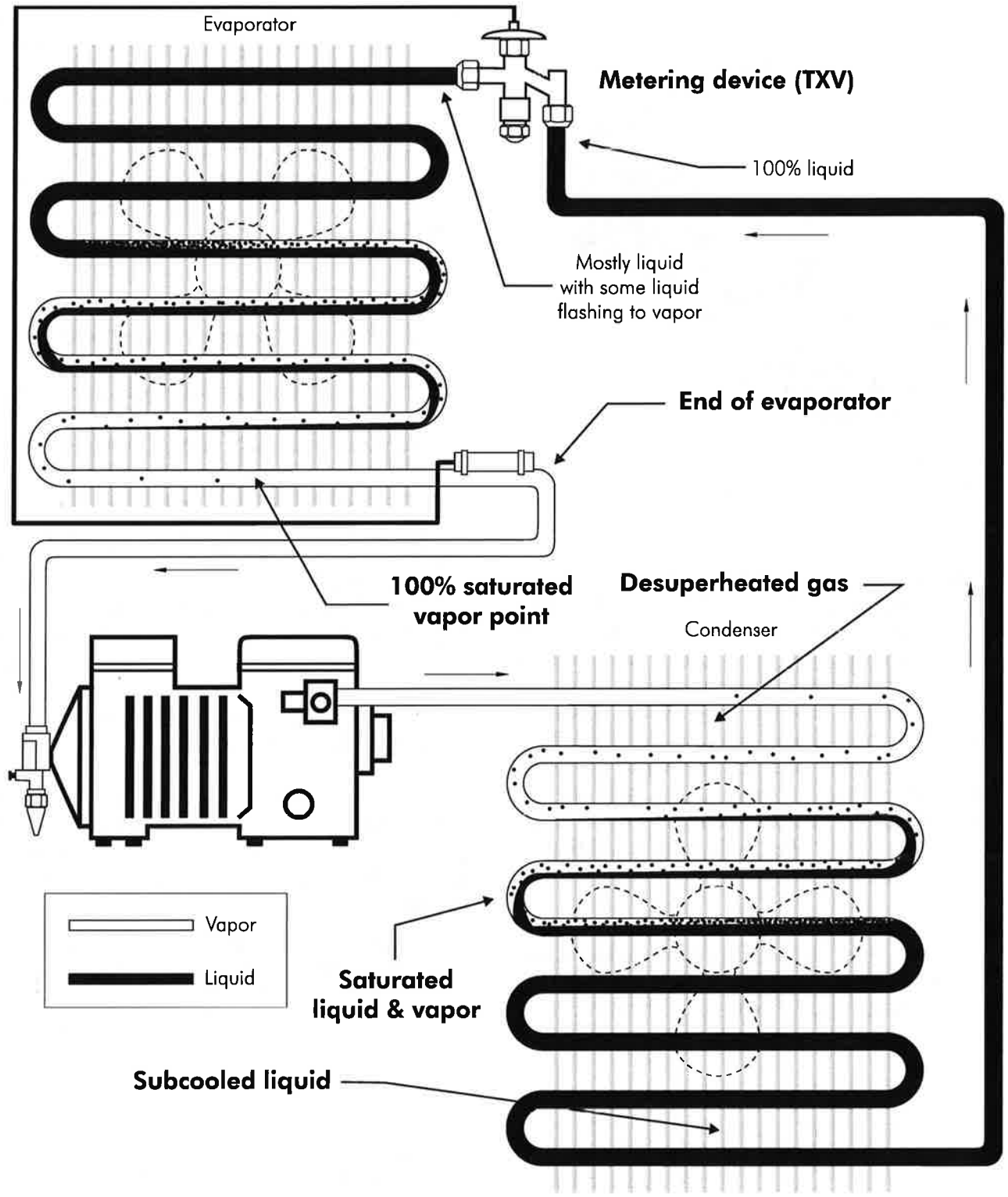


Figure 2-18. 100% saturated vapor point in evaporator and start of superheat

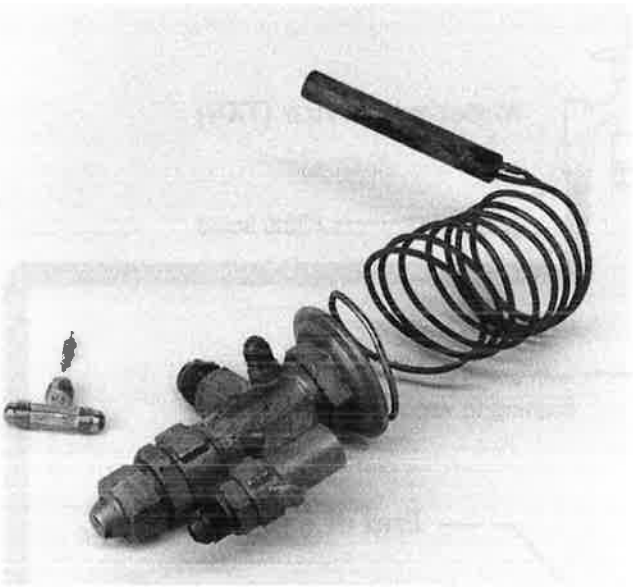


Figure 2-19. Externally equalized TXV with Schrader tee (Photo by Bill Bitzinger, Office of University Communication Services, Ferris State University)

The superheat changed from 7° to 15°F simply by reading the pressure at the compressor inlet instead of the evaporator outlet. The correct evaporator superheat is 7°F. It is best to measure the pressure at the same location as the temperature to exclude any system pressure drops. As an alternate, read the pressure at the compressor inlet and add estimated losses in the suction line to arrive at the approximate pressure at the evaporator outlet. (This alternate method is to be used on air conditioning systems only.)

The amount of evaporator superheat that is required for certain applications varies. Commercial ice makers call for 5° to 6°F of evaporator superheat to fill out their ice sheets; however, suction line accumulators are often employed on these systems for added protection. This helps ensure that all of the refrigerant entering the compressor is liquid free. This also helps keep a fully active evaporator. Lower temperature applications generally utilize lower evaporator superheat. Consult the case manufacturer if in doubt. In the absence of manufacturer data, Table 2-5 illustrates guidelines for superheat settings.

Total Superheat

There will always be times when the evaporator has a light load and the TXV may lose control of its evaporator superheat due to limitations of the valve and system instability. This is when total superheat comes into play. Total superheat, also called compressor superheat, is all the superheat in the low side of the refrigeration system. It starts at the 100% saturated vapor point in the evaporator and ends at the compressor inlet, Figure 2-20. Total superheat consists of evaporator superheat and suction line superheat. Total superheat can be measured by a technician by placing a thermistor or thermocouple at the compressor inlet and taking the temperature. A pressure reading will also be needed at this same location.

For example, consider an R-134a system with a low side pressure at the compressor of 20 psig or 23°F and a compressor inlet temperature of 50°F. The total superheat calculation is as follows:

50°F (compressor inlet temperature)	–	23°F (saturation temperature)	=	27°F (total superheat)

Applications	Air Conditioning and Heat Pump	Commercial Refrigeration	Low Temperature Refrigeration
Evaporator temperature	40° to 50°F	0° to 40°F	-40° to 0°F
Suggested superheat setting	8° to 12°F	6° to 8°F	4° to 6°F

Table 2-5. Evaporator superheat guidelines

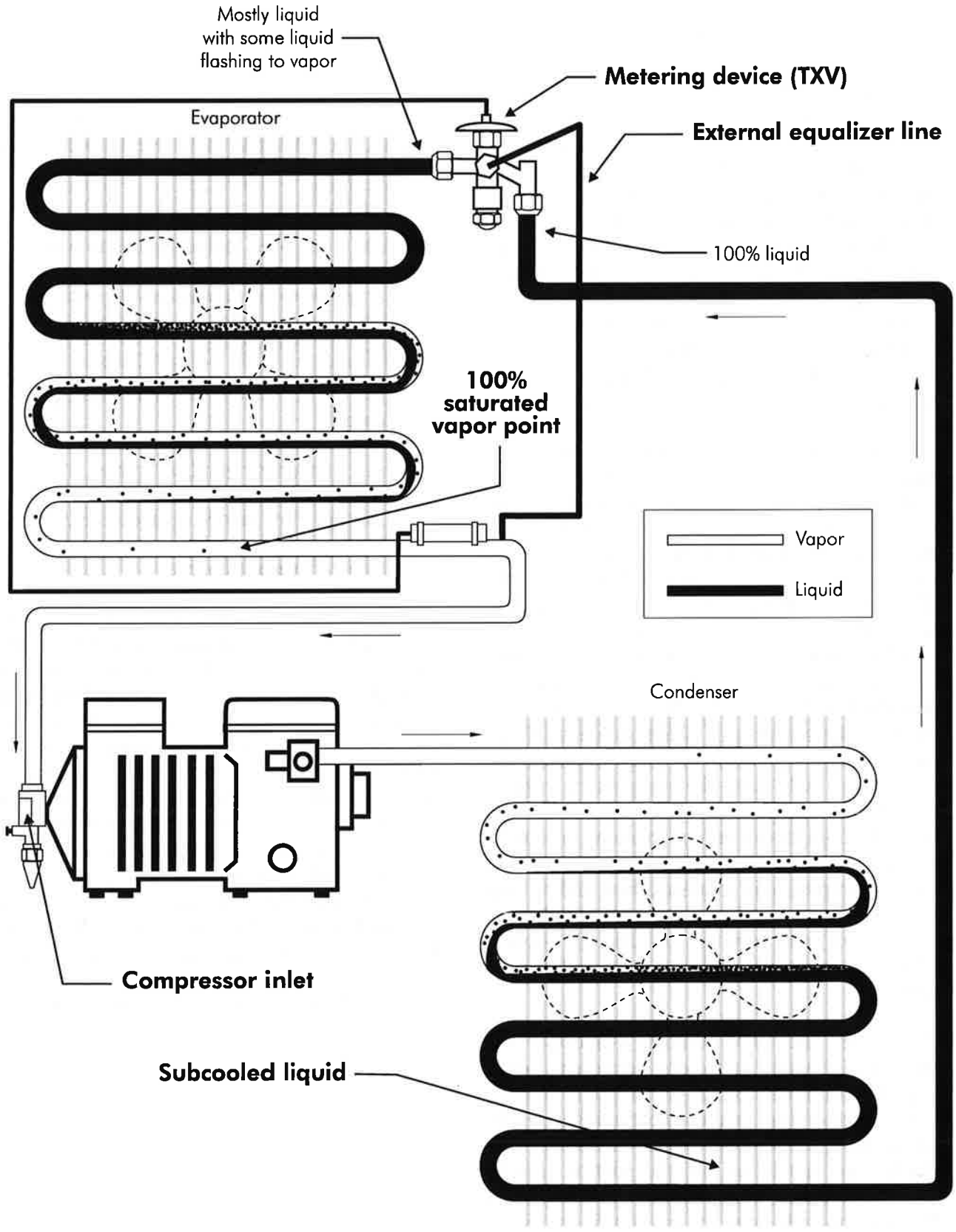


Figure 2-20. Refrigeration system showing total superheat

In this example, the total superheat is 27°F. It is possible to have a TXV that is adjusted to control superheat at the coil (evaporator superheat) and still return liquid refrigerant to the compressor at certain low load conditions. If so, the conditions causing the floodback should be found and corrected. It is recommended that all TXV controlled refrigeration systems have at least 20°F of compressor superheat to ensure that the compressor does not see liquid refrigerant (flood or slug) at low evaporator loads. Total superheats from 20° to 30°F are recommended to ensure adequate compressor cooling and preventive liquid control to the compressor. The TXV, however, should be set to maintain proper superheat for the evaporator.

Air-cooled compressors are more vulnerable to slugging and valve damage, because the suction gases are not heated by the motor windings. The gases enter the sidewall of the compressor and go directly to the valves. The 20°F of compressor superheat is a buffer in case the TXV loses control of superheat at these low loads. However, the evaporator superheat must still be maintained by the guidelines in Table 2-5. A buffer of 20° to 30°F of compressor superheat makes sure that the refrigerant vapor entering the compressor is not too dense. Density of vapors that are too high entering the compressor will cause the compressor to have a higher than normal amp draw. This will overload the compressor motor in many instances and open thermal overloads. On the other hand, excess compressor or total superheat and/or long periods of low mass flow rate (e.g., an unloaded compressor) can result in insufficient cooling of the stator and opening of the internal protectors.

Remember, the TXV controls evaporator superheat. To obtain more total superheat, add a liquid/suction heat exchanger or even run a bit longer suction line to allow heat gains from the surrounding temperature to heat the suction line. Taking the insulation off the suction line to increase total superheat is not recommended, because this will cause the suction line to sweat from water vapor in the air reaching its dew point on the suction line. Freezing of this condensation may also occur if suction line temperatures are below 32°F. If at all possible, do not sacrifice (raise) evaporator superheat to obtain the amount of total superheat needed. This will not maintain an active evaporator and system capacity will suffer.

Low evaporator loads can be caused by many different situations, including the following:

- Evaporator fan motor not operating
- Iced up or dirty coil
- Defrost circuit malfunction causing coil icing
- End of the refrigeration cycle
- Low air flow across evaporator coil
- Low refrigerant charge

Any time the evaporator coil sees a reduced heat load, a TXV can lose control and hunt. Hunting is when the valve overfeeds then underfeeds while trying to find a controlling point. Hunting occurs during periods of system unbalance (e.g., low loads) when temperatures and pressures become unstable. The TXV tends to overfeed and underfeed in response to these rapidly changing values until the system conditions settle out and the TXV can stabilize. It is this overfeeding condition that may damage compressors. Evaporator superheat settings that are too low may also cause the TXV to hunt. Remember, a total superheat setting of at least 20°F can prevent the compressor from seeing any liquid refrigerant at most conditions.

CHAPTER THREE

Metering Devices

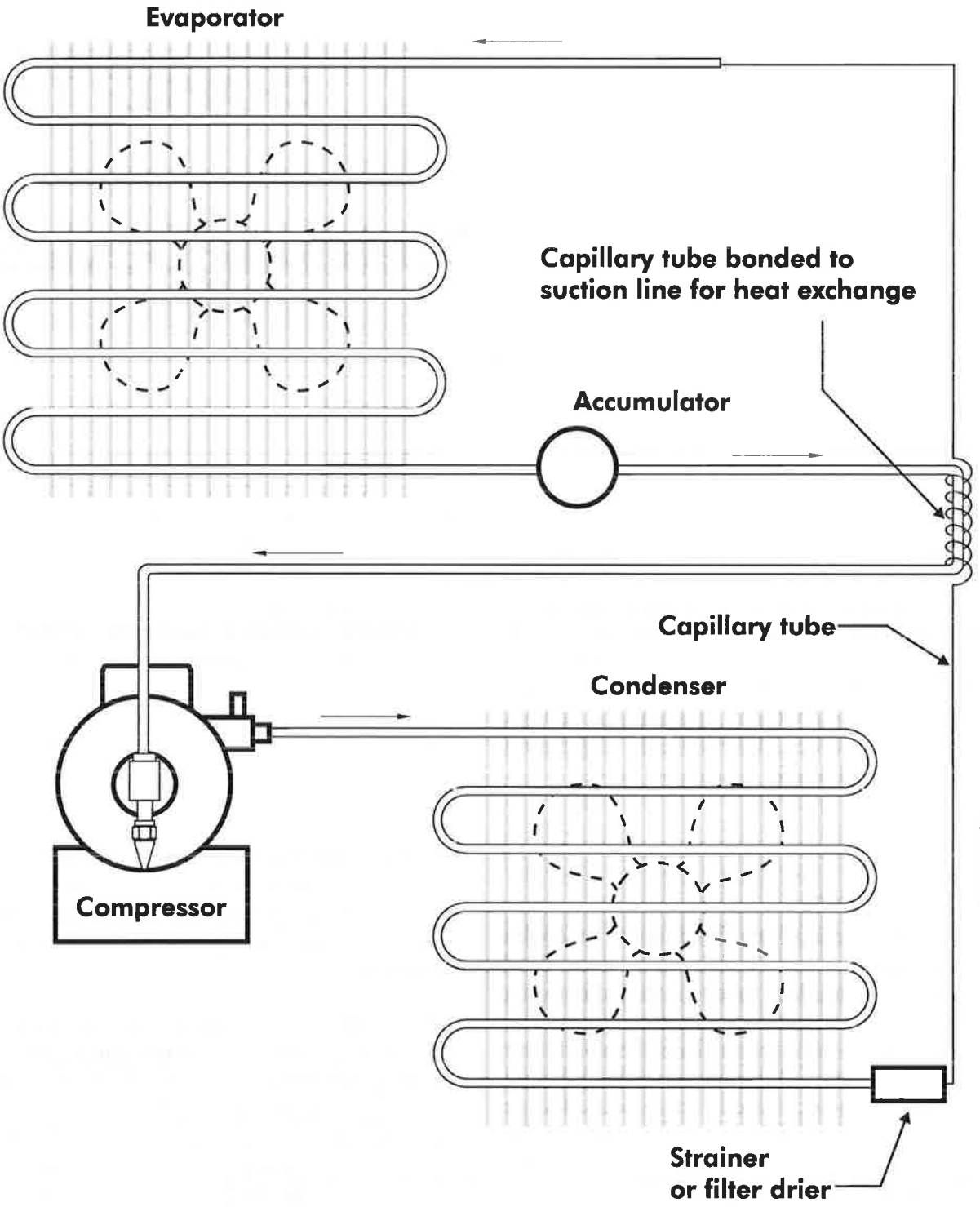
Metering devices separate the high and low side pressures of a refrigeration and/or air conditioning system by adding restriction to the system. Metering devices meter liquid refrigerant from the liquid line to the evaporator, and some metering devices control evaporator superheat, and/or pressures, and even prevent evaporator pressures from exceeding maximum set pressures. The most common types of metering devices are capillary tubes, thermostatic expansion valves, and automatic expansion valves.

Capillary Tubes

A capillary tube is a long fixed-length tube with a very small diameter, which varies from .025" to .060". The capillary tube is installed between the condenser and the evaporator, Figure 4-1, and meters the refrigerant from the condenser to the evaporator. Because of its long length and small diameter, there is an associated large friction loss as refrigerant flows through the tube. As subcooled liquid travels through the capillary tube, it may flash because pressure drops may bring it lower than the saturation pressure for its temperature. This flashing is caused by the expansion of the liquid as it experiences pressure drop. Liquid flashing amounts, if any, depend on the amounts of liquid subcooling coming from the condenser and in the capillary tube itself. If liquid flashing does occur, it is desirable to keep the flashing as close to the evaporator as possible to ensure better system performance. To prevent liquid flashing, capillary tubes are usually twisted around, placed inside, or soldered to suction lines, Figure 4-2. Because the capillary tube restricts and meters the flow of liquid to the evaporator, it helps maintain the needed pressure difference for proper system operation.

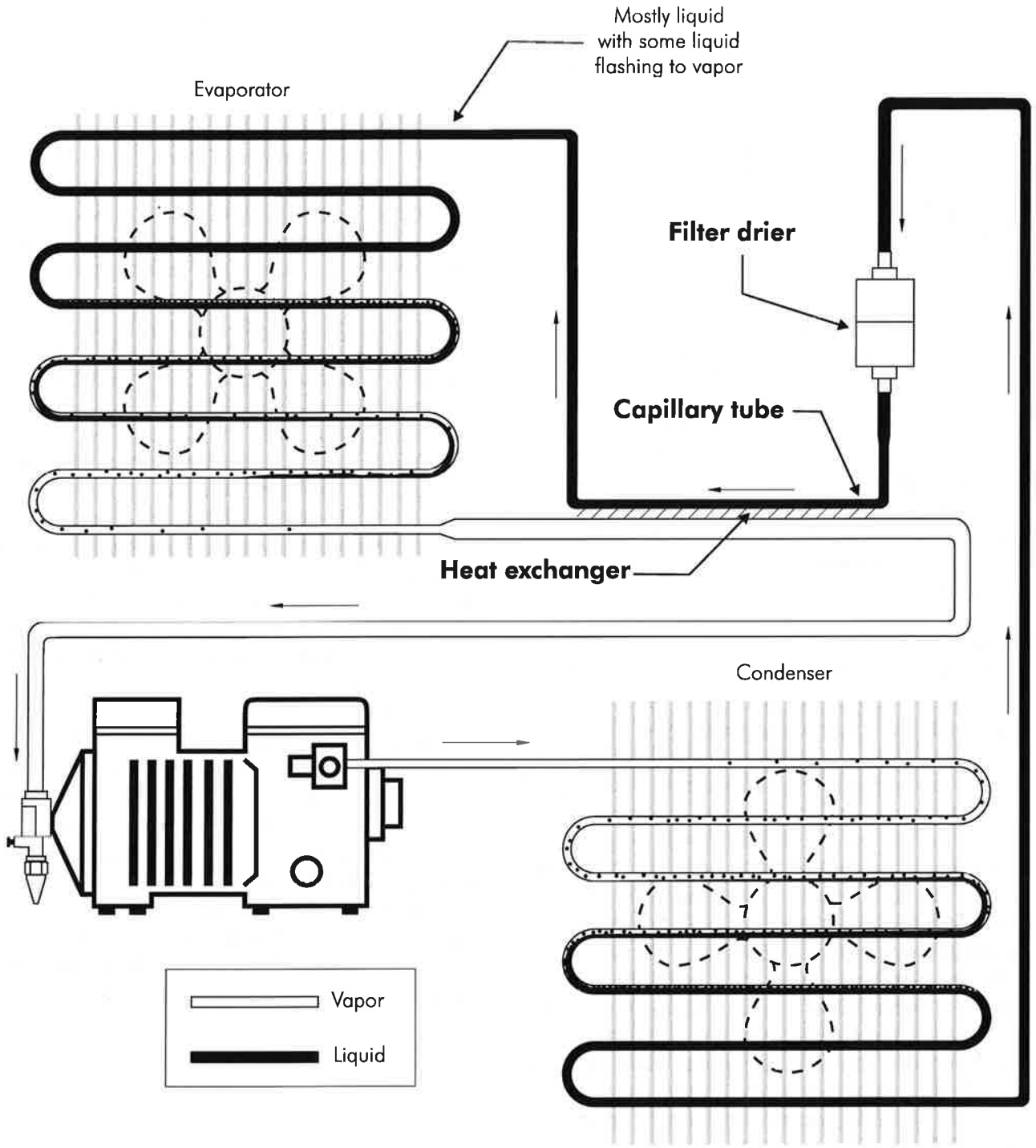
Manufacturers use capillary tubes as metering devices because of their simplicity and low cost. Systems employing capillary tubes as metering devices also do not require receivers, which is another cost savings. Capillary tube systems are found mainly in domestic and small commercial applications in which hermetic compressors are employed, and somewhat constant loads and small refrigerant flow rates are experienced.

The capillary tube has no moving parts and does not control evaporator superheat at all loading conditions, Figure 4-3. Even with no moving parts, the capillary tube varies the flow rate as system pressures change in the evaporator and condenser, or both. In fact, it can only reach its best efficiency at one set of high and low side pressures. This is because the capillary tube works off of the pressure difference between the high and low sides of the refrigeration system. As the pressure difference between the high and low side of the system becomes greater, the flow rate of refrigerant increases. The capillary tube operates satisfactorily over quite a large range of pressure differences but not very efficiently.

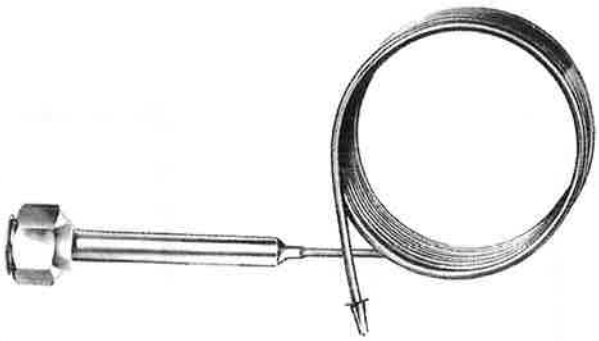


Sizing

Since the capillary tube, evaporator, compressor, and condenser are in series, the flow rate of the capillary tube must be equal to the compressor pumping rate. This is why the designed length and diameter of the capillary tube at designed vaporizing and condensing pressures is critical and must be equal to the compressor pumping capacity at these same design conditions. Too many turns in the capillary tube will affect resistance of flow and system balance.



If the resistance of the capillary tube is too great because it is too long, a partial restriction exists, the diameter is too small, or there are too many turns as it is coiled, the capacity of the tube will be less than that of the compressor. The evaporator will then be starved, causing low suction pressure and high superheats. At the same time, subcooled liquid will back up in the condenser, causing higher head pressure, because there is no receiver in the system to hold the refrigerant. With the higher head pressure and lower evaporator pressure, the refrigerant flow rate will be increased because of a greater pressure difference across the capil-

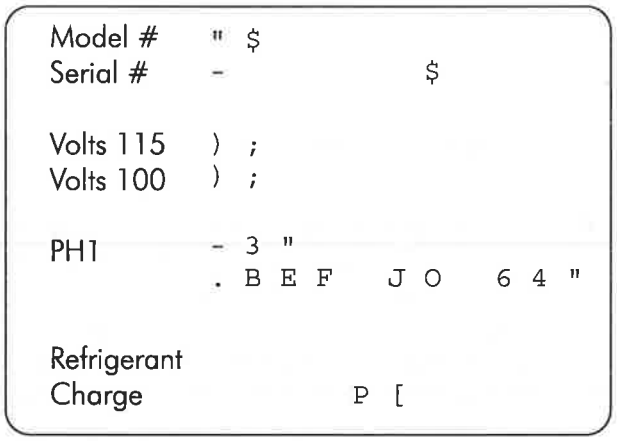


lary tube. At the same time, the compressor capacity will decrease because of higher compression ratios and lower volumetric efficiencies. This will cause the system to establish a balance but at higher head pressures and lower evaporating pressures, causing unwanted inefficiencies.

If the resistance of the capillary tube is less than called for because it is too short or its diameter is too large, refrigerant flow will be greater than the pumping capacity of the compressors. This will cause high evaporator pressures, low superheats, and possible flooding of the compressor due to overfeeding of the evaporator. In addition, subcooling will drop in the condenser, causing low head pressures and even a loss of liquid seal at the condenser bottom. This low head pressure and higher than normal evaporator pressures will decrease the compression ratio of the compressor, causing high volumetric efficiencies. This will increase the pumping rate of the compressor, and a balance may be reached if the compressor can keep up with the high refrigerant flow in the evaporator. Usually flooding occurs and the compressor fails to keep up. It is for these reasons that the capillary tube should be sized properly by calling the manufacturer. It is also necessary that capillary tube systems have an exact (critical) charge of refrigerant in their systems. Too much or too little refrigerant will cause a severe imbalance to occur and seriously damage compressors from slugging or flooding.

If a capillary tube system is overcharged, it will back up the excess liquid in the condenser, causing high head pressures. This is because there is no receiver in the system. The pressure difference between the low and high side of the system will also increase, causing an increased flow rate to the evaporator and overfeeding of the evaporator, which may result in low superheats, flooding, or slugging of the compressor. This is another reason why a capillary tube system must be critically charged with a specified amount of refrigerant.

Capillary tube systems do not stop the flow of refrigerant after the compressor shuts off. Refrigerant will continue to flow from the condenser to the evaporator until high and low pressures equalize. In fact, most of the subcooled liquid in the condenser will flow through the capillary tube to the evaporator during the off cycle, which is why the capillary tube system must be critically charged with refrigerant. Too much liquid in the evaporator during the off cycle can cause compressor damage at start-ups. The refrigerant charge should satisfy the requirements of the evaporator and maintain a subcooled liquid seal at the condenser bottom and capillary tube entrance. If any excess refrigerant is added past the critical charge, subcooled liquid will back up in the condenser, causing high head pressures and unwanted inefficiencies. Because of the pressure equalization during off cycles, capillary tube systems can employ low starting torque motors. These motors usually use current starting relays sometimes without starting capacitors. The critical charge is usually stamped on the manufacturer nameplate, Figure 4-4.



It is important that larger commercial capillary tube systems have accumulators at their evaporator outlets to catch any liquid that may escape vaporization in the evaporator during start-ups. Liquid coming from the evaporator enters the bottom of the accumulator, and the vapors are drawn from the top into the compressor.

High Heat Loads

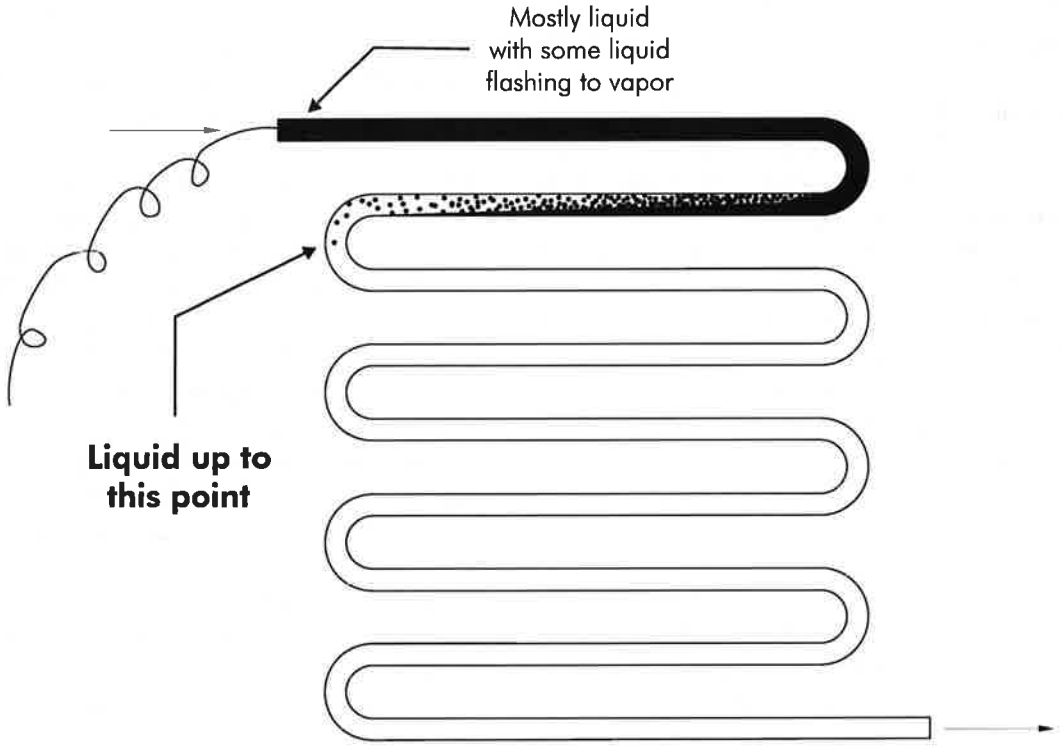
Under high loads, capillary tube systems will often cause high superheat in the evaporator of 40° or 50°F. This is because the refrigerant in the evaporator will vaporize rapidly and drive the 100% saturated vapor point back up in the evaporator to give the system a high superheat reading, Figure 4-5. The capillary tube does not have a feedback mechanism to tell it there is high superheat, which makes the system run inefficiently at higher loads. This is one of the main disadvantages of a capillary tube system. Many technicians have the urge to add more refrigerant to the system at these high loads because of the high superheat readings. Adding refrigerant at this point in time will only overcharge the system. Before adding refrigerant, check the superheat reading at the low evaporator load and see if it is normal. If unsure, recover the refrigerant, evacuate the system, and add the specified nameplate critical charge of refrigerant.

Low Heat Loads

Once the high heat load on the evaporator drops off and the system is under a low load, the 100% saturated vapor point in the evaporator will climb down the last passes of the evaporator. This is caused by the decreased rate of vaporization of refrigerant in the evaporator from the low heat load. The system will then have a normal evaporator superheat of about 10° to 12°F, Figure 4-6. These normal evaporator superheat readings only occur with low heat loads on the evaporator.

Service

One of the major problems with capillary tube systems is that they can become restricted or completely blocked due to foreign materials, freeze-up if water exists in the system, or kink-



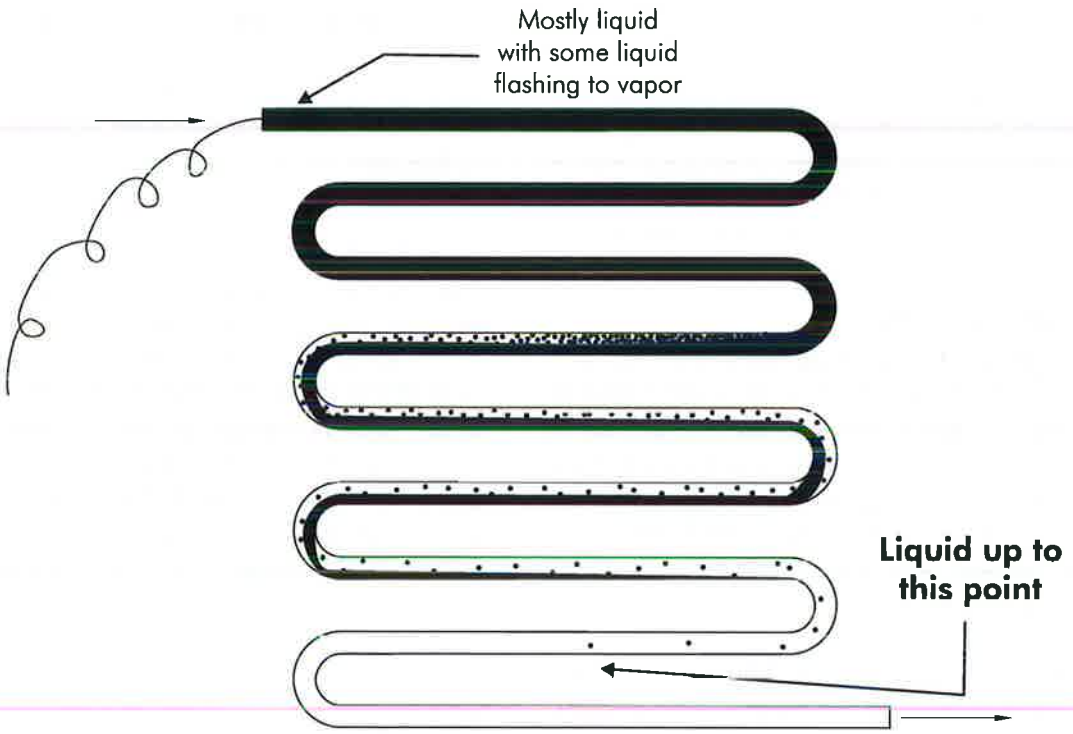


Figure 4-6. Capillary tube feeding evaporator with proper amount of superheat for low heat load

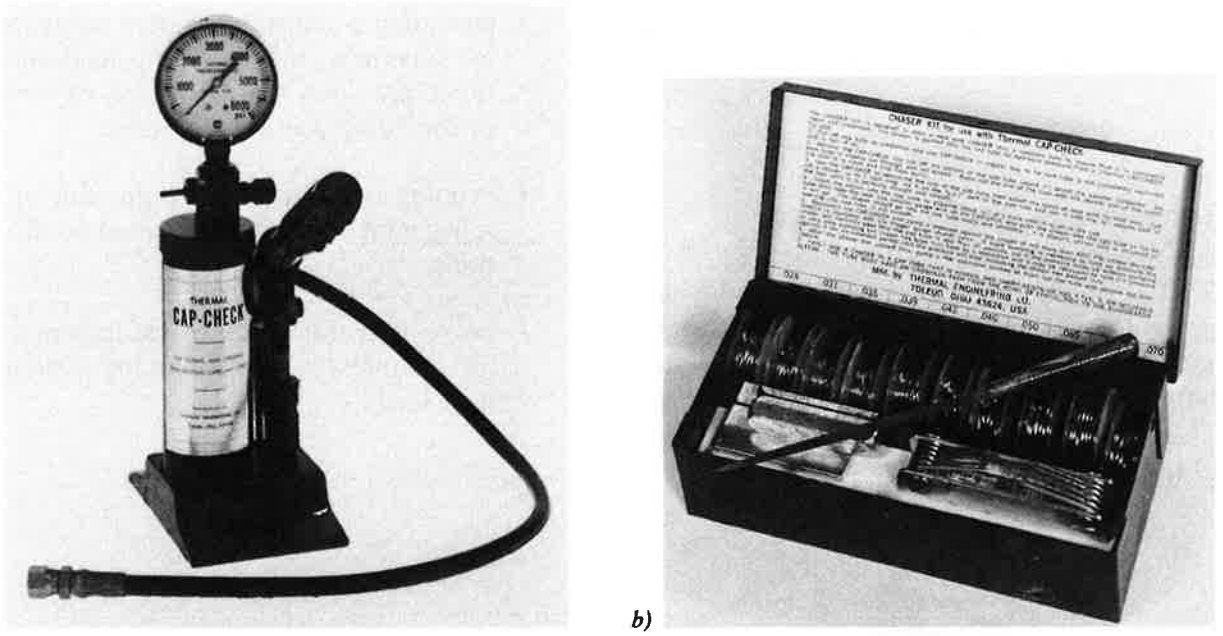
ing. Dirt, sandpaper grit, steel wool, and sand can all restrict or completely block capillary tubes.

It is extremely important to install a filter drier before the capillary tube to catch any foreign objects. Filter driers also help catch moisture so freeze-up will not occur at the evaporator entrance. High head pressure from backed-up subcooled liquid in the condenser, which will starve the evaporator and cause low pressures in the low side of the system, is one symptom of a partially blocked capillary tube. The symptoms will vary depending on how severe the capillary tube is restricted. A completely plugged capillary tube system will have dangerously high head pressures with evaporator pressures in the deep vacuum range.

Moisture in the system will pass through the condenser and the capillary tube. Once the moisture experiences the sudden pressure drop of the evaporator, it will gradually freeze. This gradual freezing around the inside diameter of the capillary tube will cause a slight restriction and hinder system performance. Head pressures will elevate and evaporator pressures will drop from the partial restriction. If the moisture problem is severe, this frozen layer will soon cause an entire restriction and shut down the compressor by thermal overload. Once the system is off, the frozen layers will melt and system pressures will equalize and wait for the thermal overload to reset. The system will have to be dehydrated and recharged.

There are hydraulic capillary tube cleaners on the market that can be very successful in removing foreign objects from capillary tubes if used properly, Figure 4-7a. Hydraulic pressure is applied to one end of the capillary tube while the other end of the tube is removed from the system. Some capillary tubes may require a small piece of wire to be hydraulically blown through them in order to clean foreign objects and sludge from their internal diameter. There is a kit available that comes with coils of small wire pistons to clear capillary tubes. This kit also includes a hydraulic cleaner, a file, and a capillary tube diameter gauge, Figure 4-7b. The wire is just undersized enough to act as a piston to clear the capillary tube when used with the hydraulic cleaner.

Some technicians may replace a plugged or restricted capillary tube with a new one. The replacement capillary tube must have the same diameter, length, and number of coils or turns as the old one. There are capillary tube gauges on the market for measuring the right diameter of capillary tube, Figure 4-8. When cutting a capillary tube, filing a small V-notch in the side of the capillary tube with an appropriate file and bending the tube back and forth where the V-notch is cut will make for a nice clean break. Burrs at the ends of capillary tubes will hinder refrigerant flow and cause inefficient system performance.



a) b)

Figure 4-7. a) Hydraulic capillary tube cleaner; b) accessories included in capillary tube cleaner kit (Courtesy, Thermal Engineering Company)

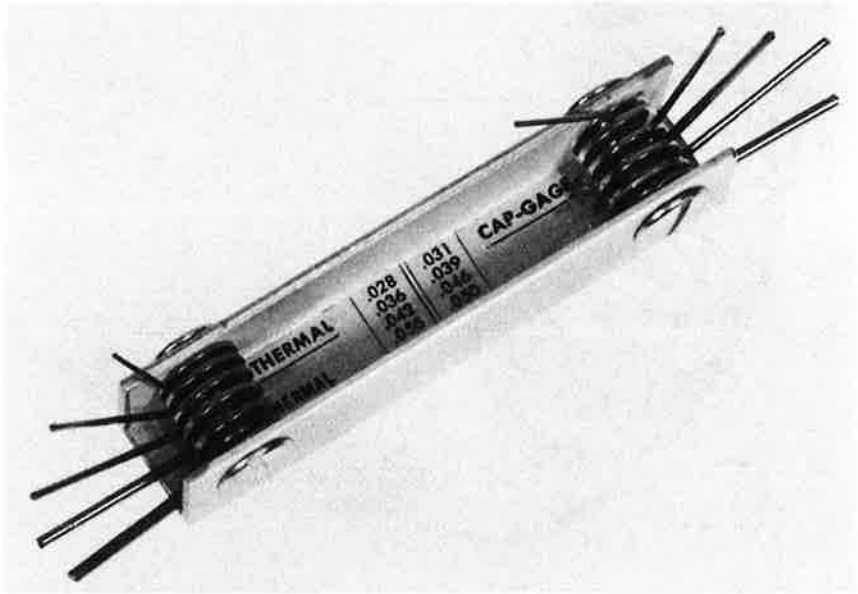


Figure 4-8. Capillary tube feeler gauges (Courtesy, Thermal Engineering Company)

Soldering the capillary tube in place requires certain skills. The capillary tube must be inserted about one or two inches past where the solder joint is to be made, Figure 4-9. This ensures solder will not creep up or down by capillary action and block the inlet or outlet of the capillary tube. A little bit of refrigeration oil smeared on the end of the capillary tube may help stop solder from bonding to the capillary tube end. When crimping the line where the capillary tube is inserted, care must be taken not to crimp or harm the capillary tube itself.

Thermostatic Expansion Valve

The thermostatic expansion valve (TXV or TEV), is probably the most popular and widely used valve in the hvac/r industry today, Figure 4-10. Chapters of material could be written about TXV applications, types, and functions, but this section will cover TXV basics. Figure 4-11 shows an exploded view of a TXV.

The TXV is a metering device located between the liquid line and the evaporator, Figure 4-12. The TXV is a dynamic valve that constantly modulates and meters just enough refrigerant into the evaporator to satisfy all heat loads on the evaporator. The basic TXV controls a set amount of evaporator superheat under varying heat loads. The three main functions of a TXV include:

- providing a constant amount of evaporator superheat under varying load conditions, provided the range and capacity of the valve are not exceeded.
- keeping the entire evaporator full of refrigerant under all heat load conditions.
- preventing floodback of refrigerant to the compressor under varying evaporator heat loads.

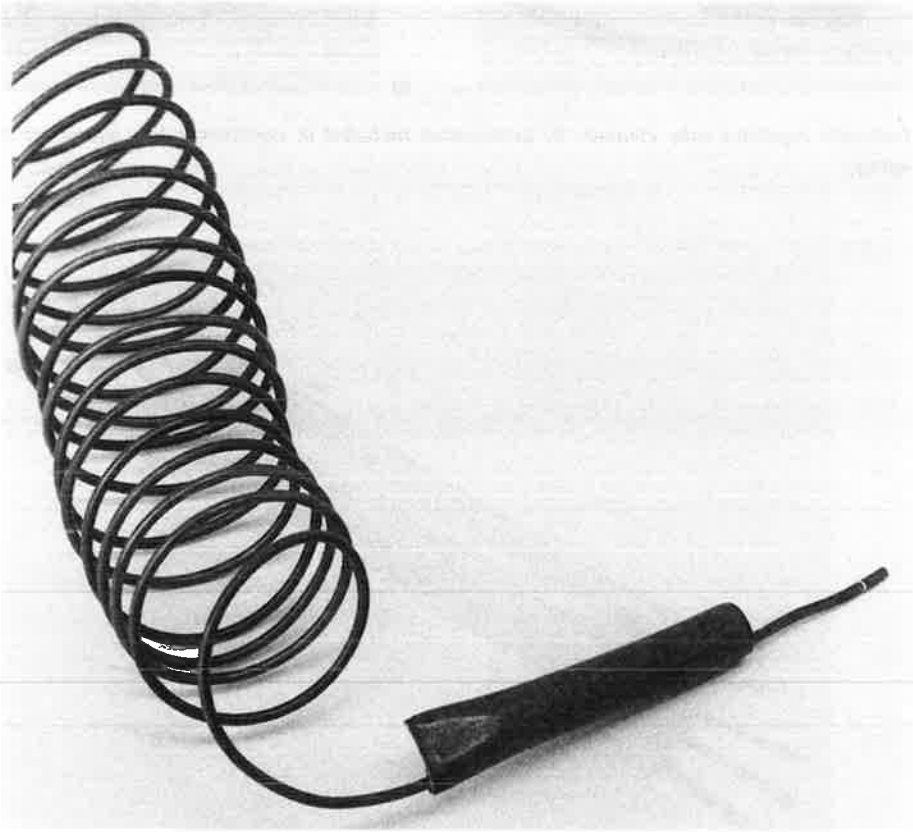


Figure 4-9. Capillary tube inserted two to three inches past solder joint to prevent solder from plugging capillary tube (Photo by Bill Bitzinger, Office of University Communication Services, Ferris State University)

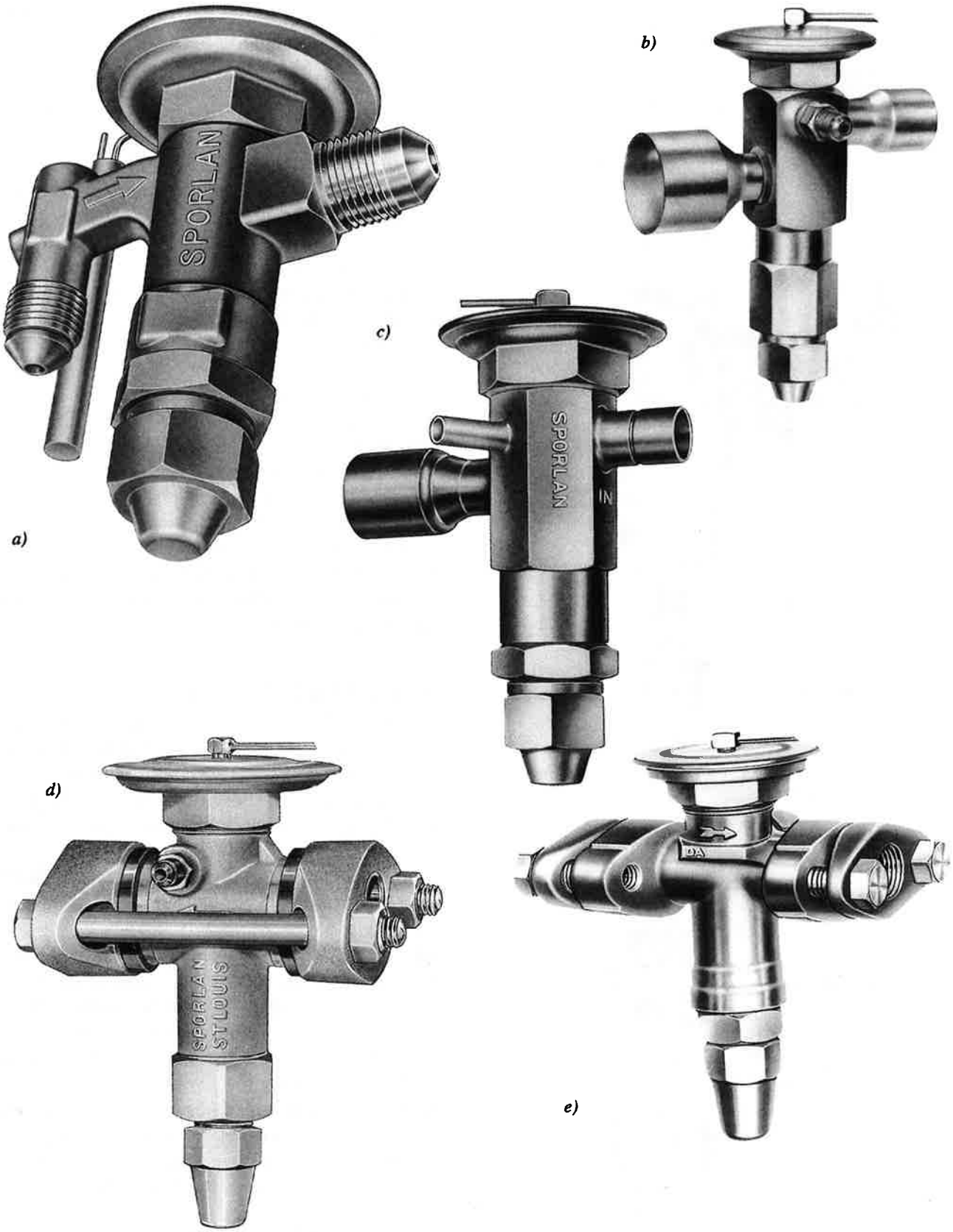


Figure 4-10. Examples of different TXVs (Courtesy, Sporlan Valve Company)

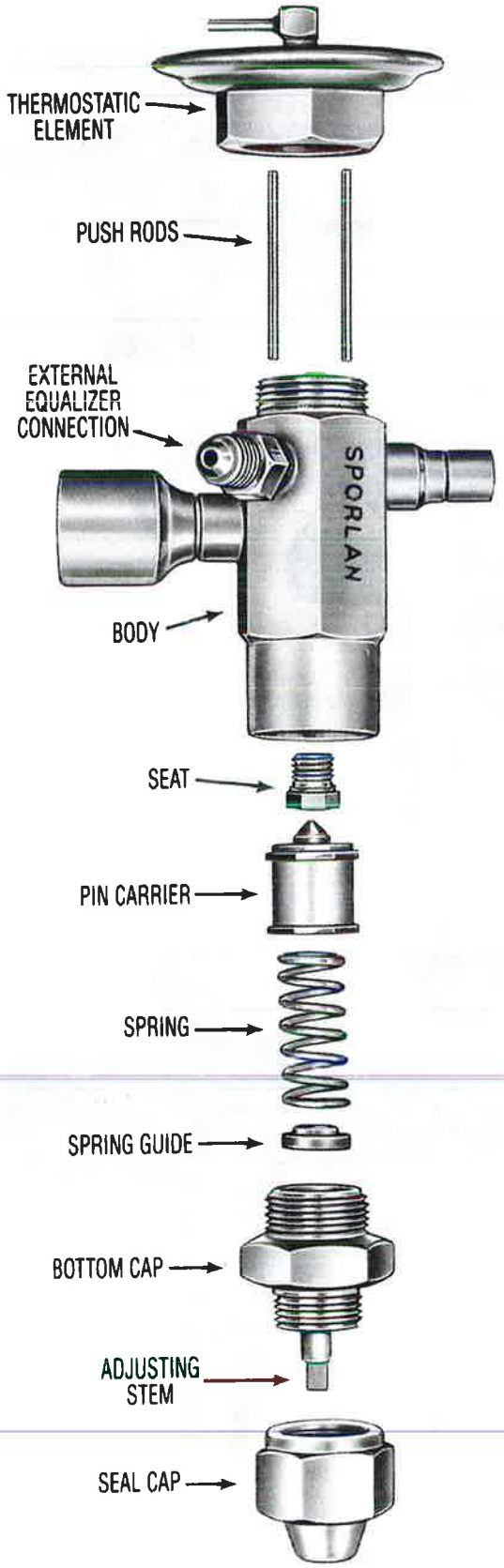


Figure 4-11. Exploded diagram of a TXV (Courtesy, Sporlan Valve Company)

The TXV does not control evaporator pressure, cycle the compressor, control running time, or control box temperature. The main function of the TXV is to meter the right amount of refrigerant into the evaporator coil under all load conditions. By doing this, the valve controls a set amount of evaporator superheat to maintain active evaporators and keep refrigeration and air conditioning systems running safely and efficiently.

One of the main differences between a TXV system and a capillary tube system is the TXV has the ability to control the right amount of evaporator superheat under varying evaporator heat loads. A capillary tube cannot do this and will run very high evaporator superheat at a high evaporator load situation. The TXV will fill the evaporator coil with refrigerant at all loads but will not cause overflow into the suction line if adjusted properly. The action of this dynamic valve is from the interaction of three separate pressures acting on the valve: *remote bulb pressure* (opening force), *spring pressure* (closing force), and *evaporator pressure* (closing force), Figure 4-13.

Remote Bulb Pressure

In Figure 4-13, notice that the pressure (P_1) from the volatile fluid in the remote bulb acts on the top side of the flexible bellows through the connecting capillary tube, causing the valve to open as pressure increases. This changing pressure in the remote bulb is caused by the evaporator outlet temperature becoming hotter or colder from varying heat loads on the evaporator, which momentarily affect evaporator superheat. Higher superheat causes more fluid to vaporize in the remote bulb, producing higher pressures. Less superheat causes more vapor to condense in the remote bulb, decreasing its pressure and closing the valve. As soon as the TXV realizes that the evaporator superheat is changing it either opens or closes, putting more or less refrigerant into the evaporator from changing remote bulb pressures. This maintains a set amount of evaporator superheat.

The remote bulb is clamped firmly to the suction line for good thermal contact. The remote

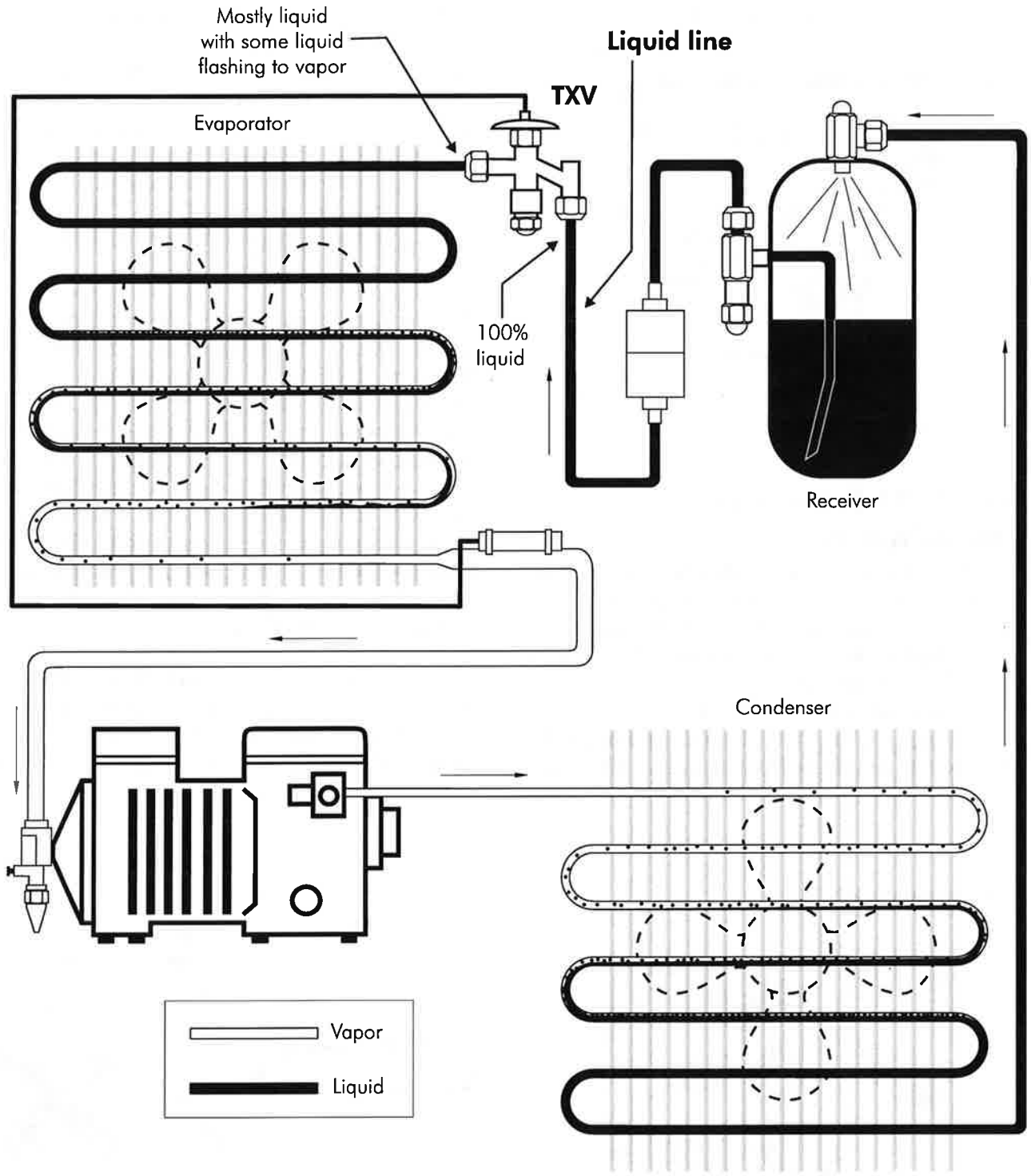


Figure 4-12. TXV location

bulb is the TXV's feedback mechanism that gives the valve an indication of current evaporator superheat. Remote bulbs come in a variety of charges for spccial applications. Consult with the TXV manufacturer for specific information. Figure 4-14 illustrates the do's and don'ts for mounting a remote bulb onto the evaporator outlet.

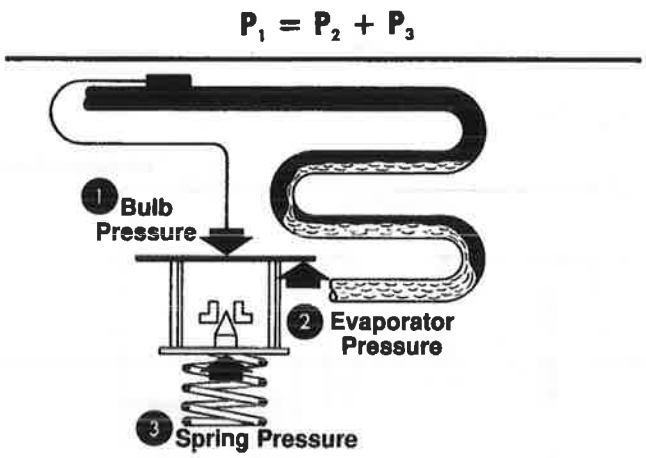


Figure 4-13. Forces acting on a TXV (Courtesy, Sporlan Valve Company)

Evaporator Pressure and Spring Pressure

As heat loads on the evaporator change, the rate of vaporization of the refrigerant in the evaporator also changes. Higher heat loads cause higher rates of vaporization of the refrigerant in the evaporator, which increases evaporator pressure. Lower heat loads on the evaporator decrease the rate of vaporization of the refrigerant and decrease evaporator pressure.

Notice in Figure 4-13 that evaporator pressures act on the underside of the flexible bellows, causing the valve to close as the evaporator pressure increases. Through a drilled passageway called the internal equalizer, evaporator pressure from the coil entrance acts on the underside of the bellows. In Figure 4-13, evaporator pressure is designated as P_2. This evaporator pressure opposes the remote bulb pressure.

The spring pressure acts in the same direction on the bellows as the evaporator pressure. It is a closing force and is designated as P_3 in Figure 4-13. When the spring pressure (P_3) and the evaporator pressure (P_2) equal the remote bulb pressure (P_1), or $P_1 = P_2 + P_3$, the valve is said to be in dynamic balance and will neither open nor close until the evaporator heat load changes. If the suction pressure rises or falls, the valve tends to open or close. If the remote bulb pressures change from changing superheat, the valve will also open or close.

Spring pressure in most valves is set for a predetermined amount of evaporator superheat (usually between 7° and 10°F) by the manufacturer. This spring is often referred to as the

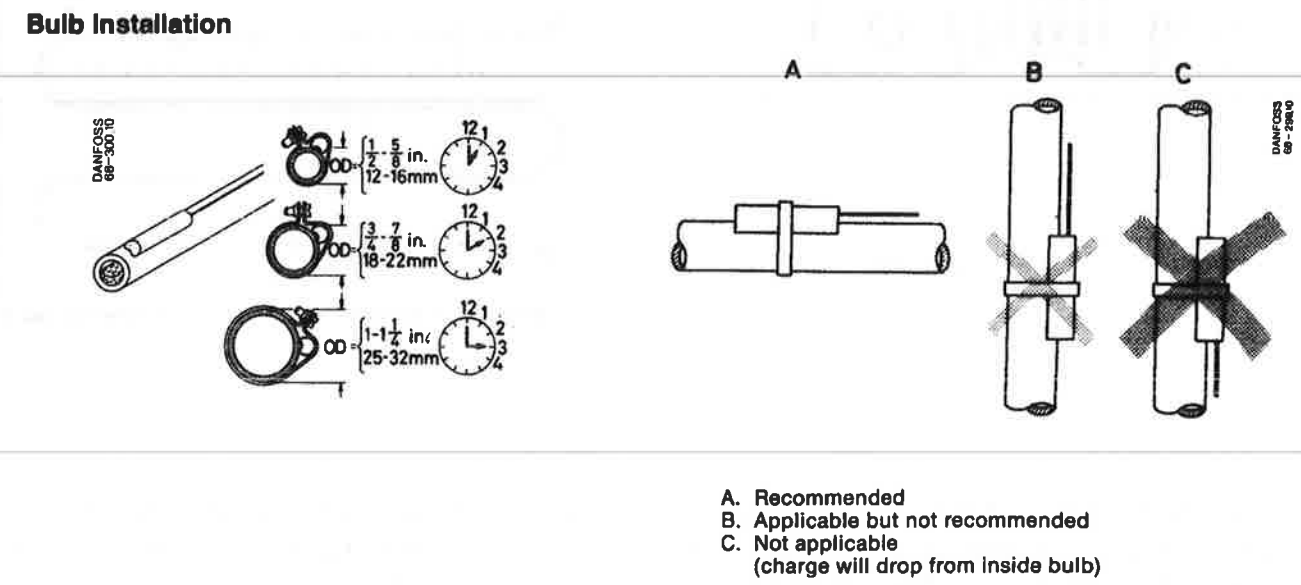


Figure 4-14. Do's and don'ts for connecting remote bulbs to evaporator outlet pipes (Courtesy, Danfoss Automatic Controls, Division of Danfoss, Inc.)

superheat spring, since it is the only way a technician can change the amount of superheat in a TXV system. Increasing the spring tension by turning the spring adjustment clockwise tightens the spring. This, in turn, gives the TXV more closing force and causes the superheat setting to increase, giving the system more operating evaporator superheat. There are some TXVs on the market that come with a set amount of superheat that cannot be changed by a technician. These TXVs are usually used in applications in which superheat settings are critical for proper performance, such as in some ice makers.

Consider a TXV whose superheat spring is set to control 10°F of evaporator superheat for a walk-in cooler. The valve is in a static equilibrium with forces ($P_1 = P_2 + P_3$). As long as there is no change in the evaporator load, meaning no change in evaporator superheat, the valve will remain static with a constant refrigerant flow rate controlling 10°F of evaporator superheat. The degree of evaporator superheat will be 10°F, which is exactly needed to offset the pressure exerted by the superheat spring. The evaporator pressure will also remain constant.

Now assume that someone opens the door to load the cooler. The added heat load on the evaporator will instantaneously cause the 100% saturated vapor point in the evaporator to climb. This happens because of the added rate of vaporization of the refrigerant, which causes the evaporator superheat to rise over the set 10°F. The remote bulb will then experience a higher evaporator superheat and temperature, and the pressure in the remote bulb will exceed the combined evaporator and spring pressure. This will cause the valve to open and let more refrigerant into the evaporator, which will have a tendency to reduce the amount of superheat until the required 10°F is again established. The evaporator will try to fill with refrigerant to gain maximum efficiency and capacity. The valve will again be in a state of static equilibrium controlling 10°F of evaporator superheat but at a higher evaporator pressure.

Additional Pressure

An additional pressure affecting TXV performance arises from the actual pressure drop across the valve. This pressure is a product of the pressure drop across the valve and the ratio of the port area to the effective area of the diaphragm:

Pressure = (Pressure drop) (Port area/effective diaphragm area)

This pressure is an opening force, and the liquid flow through the valve tends to move the TXV in an opening direction. It is sometimes referred to as liquid pressure, but this nomenclature is often misleading. This pressure only becomes significant when the pressure drop across the valve becomes large and/or the ratio of port area to effective diaphragm area becomes large. Because of this pressure, the balanced port TXV was introduced, which will be discussed later in this chapter.

Color Codes

Many times manufacturers of TXVs will color code the top of the valve. This helps the service technician quickly distinguish what refrigerant is in the system. The type of refrigerant is often stamped on the valve top along with a color-coding decal. Figure 4-15 is an example of one manufacturer's color-coding system.

Refrigerant Color Code Used on decals				
R-11	— Blue	R-114	— Blue
R-12	— Yellow	R-134a	— Blue
R-13	— Blue	R-500	— Orange
R-13B1	— Blue	R-502	— Purple
R-22	— Green	R-503	— Blue
R-113	— Blue	R-717	— White

Figure 4-15. Color-coding chart for TXVs (Courtesy, Sporlan Valve Company)

Special Application TXVs

There are special application TXVs available that should be considered when selecting the correct valve for an application. These special application TXVs include the *externally equalized valve*, the *maximum operating pressure valve*, and the *balanced port valve*.

Externally Equalized Valves

The externally equalized valve senses evaporator pressure at the evaporator outlet, not the evaporator inlet, Figure 4-16. This compensates for any friction pressure drops throughout the length of the evaporator. By sensing evaporator outlet pressure, there is less pressure on the bottom of the TXV bellows than if evaporator inlet pressure were sensed. This lower pressure causes a smaller closing force, so the valve remains more open and fills out more of the evaporator by compensating for the pressure drop through the evaporator. Evaporator outlet pressure is sensed through a small tube at the tailpipe of the evaporator leading underneath the TXV bellows, Figure 4-17. Externally equalized TXVs were shown in Figure 4-10 (b), (c), and (d).

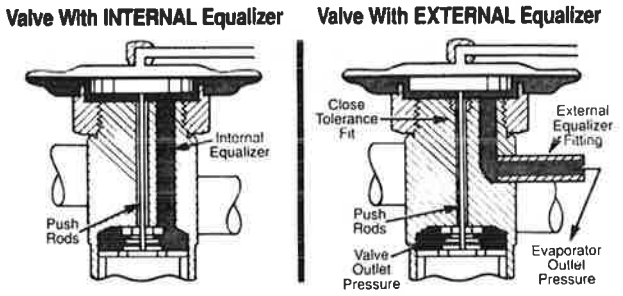


Figure 4-16. Cutaway view of an externally equalized TXV (Courtesy, Sporlan Valve Company)

An externally equalized TXV is used under the following conditions:

- When evaporator pressure drops cause a 2°F drop in temperature in the evaporator. This is for evaporator temperature applications above 0°F.
- When evaporator pressure drops cause a 1°F drop in temperature in the evaporator. This is for evaporator temperature applications below 0°F.
- Anytime distributors feed an evaporator.

Maximum Operating Pressure (MOP) Valves

MOP valves limit evaporator pressure to a maximum but not a minimum. MOP valves prevent high amp draws and overloads of compressors on high heat loads or pulldowns. They also prevent overfeeding at start-ups. This type of TXV has a limited amount of liquid in its remote bulb and will run out of liquid after reaching a certain temperature or maximum operating pressure, Figure 4-18. This prevents the further opening of the TXV and protects the compressor from high evaporator pressures during high loads or pulldowns, because the

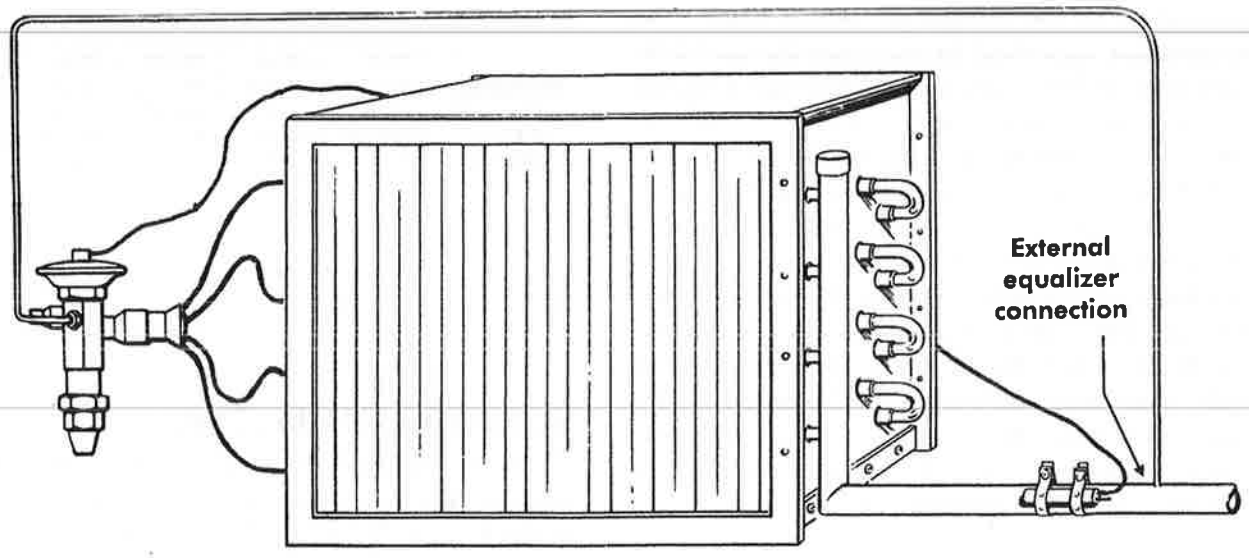


Figure 4-17. External equalizer connection for a TXV (Courtesy, Sporlan Valve Company)

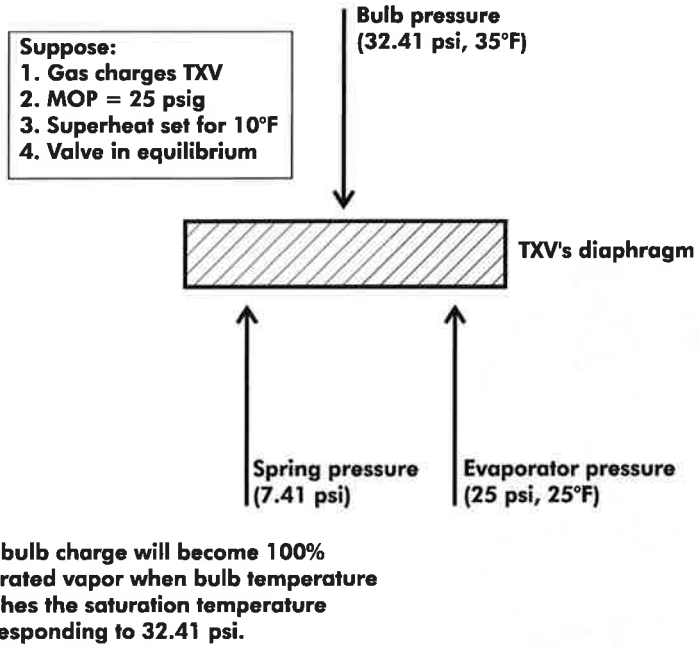


Figure 4-18. Forces on a maximum operating pressure (MOP) TXV

evaporator pressure will never exceed the MOP setting of the valve. In some valves, turning the superheat spring will set a new MOP set point. The MOP TXV may have a mechanical means of preventing a maximum pressure from occurring. The MOP of the valve is usually stamped on the valve in an obvious location.

Balanced Port TXVs

Balanced port TXVs have balanced pressures across the valve to negate the effects of varying head pressures or pressure drops across the valves. The liquid pressure is actually canceled out because it acts on equal areas but in opposite directions. The design uses a double seating piston operated by a single pushrod. The two port construction divides the refrigerant flow into opposite directions, providing a balanced pressure differential across the piston, Figure 4-19. Balanced port TXVs have a large port compared to conventional TXVs, and liquid pressure can affect a large ported valve if not balanced. This pressure only becomes significant when the pressure drop across the valve becomes large and/or the ratio of port area to effective diaphragm area becomes large. Because of the balanced port design, the valve can operate satisfactorily down to approximately 25% of its rated capacity.

The balanced port valve operates with the valve pin controlling very close to its seat. This provides a very stable control of superheat at minimum valve stroke movements, which allows a large port to handle both large and small loads. This type of design helps refrigeration equipment pull down faster. The larger port size can handle some liquid line flash gas if it exists.

Balanced port TXVs should be used if any of the following conditions exist:

- Liquid line flash gas
- Large varying head pressures
- Large varying pressure drops across the TXV affecting capacity of the valve
- Widely varying evaporator loads
- Very low liquid line temperatures

Selection Procedures

When selecting a TXV, always follow the manufacturer's selection procedures and nomenclature. The procedures used in selecting a TXV in this section are provided by Sporlan Valve

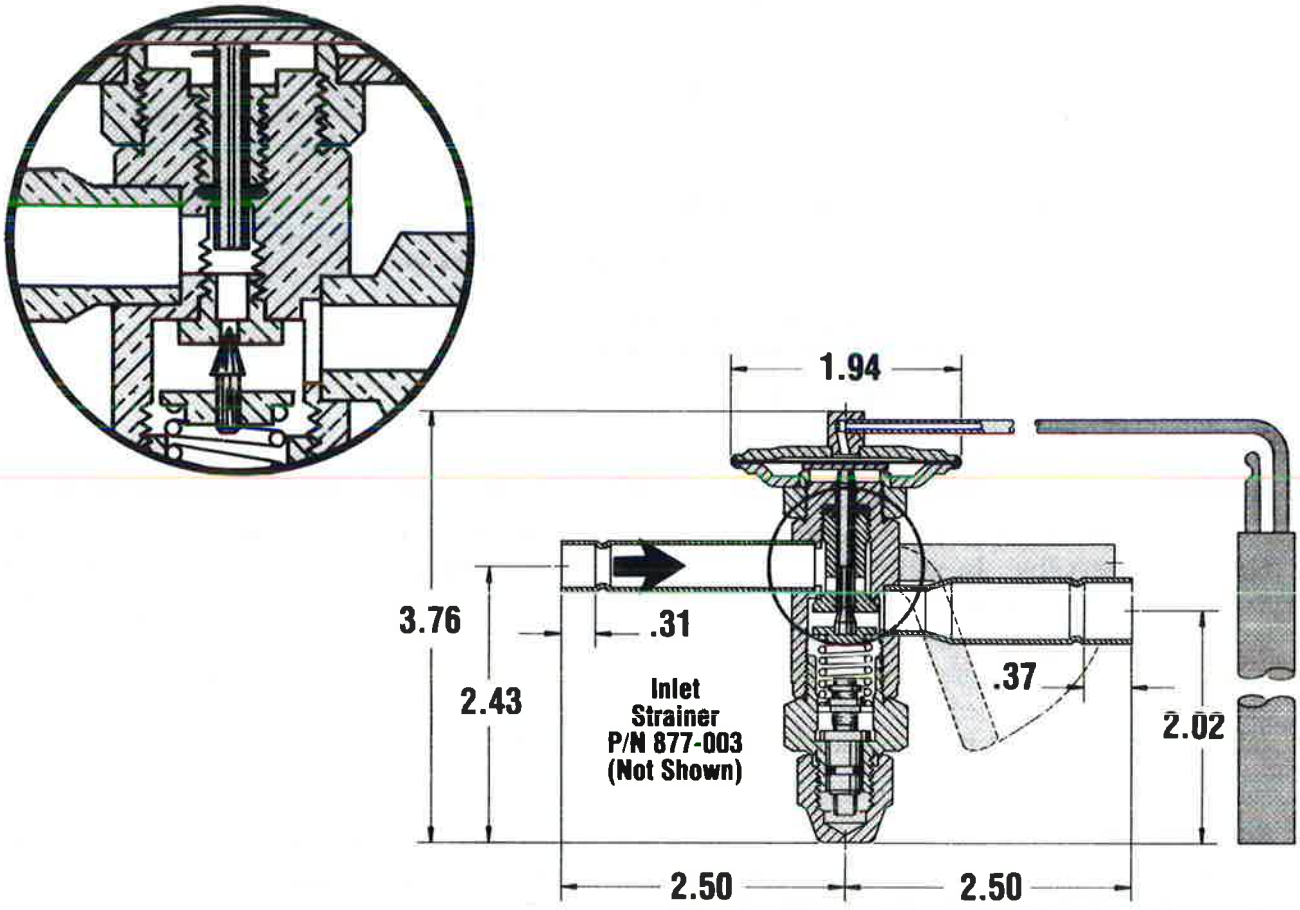


Figure 4-19. Cutaway of balanced port TXV (Courtesy, Sporlan Valve Company)

Company. Similar procedures are used when selecting TXVs from other manufacturers:

1. Determine the pressure drop across the valve by subtracting the evaporating pressure from the condensing pressure. The condensing pressure should be the minimum operating condensing pressure of the system, which usually occurs at lower loads and lower ambients. From this value, subtract all other pressure losses to obtain the net pressure drop across the valve. Pressure drop across the valve is an important measurement, because that pressure is the driving force for the refrigerant to flow across the valve. Without a pressure drop, there will not be any flow. As the pressure drop increases, more refrigerant will flow across the valve. Consider all sources of pressure drop, including:

   a. friction losses through refrigeration lines including the evaporator, suction line, and condenser.

   b. pressure drop across liquid line and suction line accessories such as solenoid valves, strainers, filters, and filter driers (see manufacturer catalogs).

c. static pressure losses or gains due to the vertical lift or drop of the liquid line, Figure 4-20.

d. pressure drop across a refrigerant distributor (if used).

REFRIG-ERANT	VERTICAL LIFT — FEET				
	20	40	60	80	100
	Static Pressure Loss — psi				
12	11	22	33	44	55
22	10	20	30	40	50
500	10	19	29	39	49
502	10	21	31	41	52
717 (Ammonia)	5	10	15	20	25

Figure 4-20. Static pressure losses vs vertical lift for different refrigerants (Courtesy, Sporlan Valve Company)

2. Determine the liquid temperature of the refrigerant entering the valve. The closer the liquid temperature is to the evaporating temperature, the less flash gas that will occur at the entrance of the valve. The less flash gas, the more net refrigeration effect. The TXV capacity table in Table 4-1 is based on a liquid temperature of 100°F for R-22. If the liquid temperature is higher or lower than 100°F, correction factors must be applied, which are also listed at the bottom of Table 4-1. Correction factors must be multiplied by the appropriate capacity in the selection table to either increase or decrease capacity (see example in bottom right hand corner of Table 4-1). Notice that if the liquid temperature falls below 100°F, a higher correction factor is applied. Higher correction factors increase the capacity of the TXV, because the liquid is colder and closer to the evaporating temperature, which causes less flash gas at the evaporator entrance and more net refrigeration effect.

3. Select the valve from the capacity table. **Select a valve based on the design evaporating temperature and the available pressure drop across the valve from Table 4-1**. If possible, the valve capacity should equal or slightly exceed the design rating of the system, Figure 4-21. Be sure to multiply the appropriate liquid temperature correction factor to the valve capacity rating shown in the tables. Once the desired valve capacity is located, determine the nominal capacity of the valve from the second column of the tables. On multiple evaporator systems, select each valve on the basis of individual evaporator capacity.

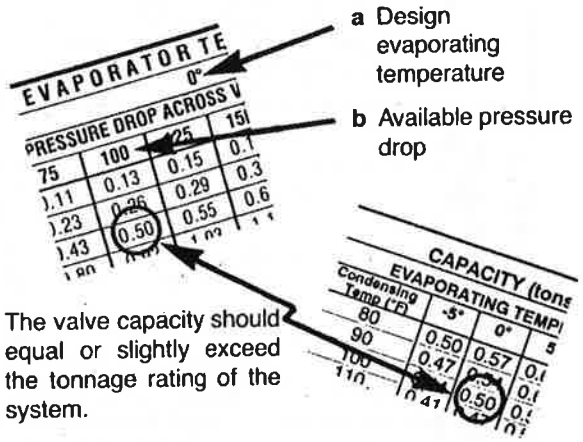


Figure 4-21. Valve capacity should equal or slightly exceed design rating (Courtesy, Sporlan Valve Company)

In Table 4-1, notice that for a constant pressure drop across the valve, as the evaporator temperature increases, the capacity of the TXV increases. This is caused by a lower compression ratio from a higher evaporator temperature and pressure, which causes a higher volumetric efficiency. In turn, the mass flow rate of refrigerant increases through the entire system including the TXV. Less flash gas is also experienced at the evaporator entrance, because the liquid in the liquid line is closer to the higher evaporator temperature. This causes a higher capacity and more net refrigeration effect.

4. Determine if an external equalizer is required. The amount of pressure drop between the valve outlet and remote bulb location (evaporator pressure drop)

R-22

THERMOSTATIC EXPANSION VALVE CAPACITIES for REFRIGERANT-22

TONS OF REFRIGERATION

AIR CONDITIONING, HEAT PUMP and COMMERCIAL REFRIGERATION APPLICATIONS

VGA, VCP100 AND VC THERMOSTATIC CHARGES

VALVE TYPE	NOMINAL CAPACITY	EVAPORATOR TEMPERATURE (°F)																							
		40°						20°						0°											
		PRESSURE DROP ACROSS VALVE (PSI)																							
		75	100	125	150	175	200	75	100	125	150	175	200	75	100	125	150	175	200	75	100	125	150	175	200
G-EG	1/3	0.17	0.20	0.22	0.24	0.26	0.28	0.17	0.19	0.22	0.24	0.26	0.28	0.15	0.17	0.19	0.21	0.23	0.24						
NI-F	1/4	0.22	0.25	0.28	0.31	0.33	0.35	0.21	0.24	0.27	0.30	0.32	0.34	0.20	0.23	0.25	0.28	0.30	0.32						
G-EG	1/3	0.30	0.35	0.39	0.43	0.46	0.49	0.30	0.34	0.38	0.42	0.45	0.48	0.26	0.30	0.33	0.37	0.39	0.42						
NI-F-G-EG	1/2	0.39	0.45	0.50	0.55	0.60	0.64	0.38	0.44	0.49	0.54	0.58	0.62	0.33	0.38	0.43	0.47	0.51	0.54						
G-EG	3/4	0.65	0.75	0.84	0.92	0.99	1.06	0.63	0.73	0.82	0.90	0.97	1.03	0.55	0.64	0.71	0.78	0.85	0.90						
NI-F-G-EG	1	0.87	1.00	1.12	1.22	1.32	1.41	0.84	0.97	1.09	1.19	1.29	1.38	0.74	0.85	0.95	1.04	1.13	1.21						
F-G-EG	1½	1.39	1.60	1.79	1.96	2.12	2.26	1.35	1.56	1.74	1.91	2.06	2.21	1.18	1.36	1.52	1.67	1.80	1.93						
F(Ext)-G & EG(Ext)-S	2	1.73	2.00	2.24	2.45	2.65	2.83	1.69	1.95	2.18	2.39	2.58	2.76	1.48	1.70	1.91	2.09	2.26	2.41						
F(Int)-G(Int)	2½	2.17	2.50	2.80	3.06	3.31	3.54	2.11	2.44	2.72	2.98	3.22	3.45	1.85	2.13	2.38	2.61	2.82	3.01						
F(Ext)-G & EG(Ext)-S	3	2.77	3.20	3.58	3.92	4.23	4 53	2.70	3.12	3.49	3.82	4.13	4 41	2.36	2.73	3.05	3.34	3.61	3.86						
C-S	4	3.90	4.50	5.03	5.51	5.95	6.36	3.80	4.39	4.90	5.37	5 80	6.20	3.32	3.84	4.29	4.70	5.07	5.42						
C-S	5	4.50	5.20	5.81	6.37	6.88	7 35	4.39	5.07	5.67	6.21	6.70	7.17	3.84	4 43	4.96	5.43	5.86	6.27						
C&S(Ext)	6	6.93	8.00	8.94	9.80	10.6	11.3	6.75	7.80	8.72	9.55	10 3	11.0	5.45	6.29	7 03	7.70	8.32	8.89						
S(Ext)	10	8.66	10.0	11.2	12.2	13.2	14.1	8.44	9.75	10.9	11 9	12.9	13.8	6.81	7.86	8.79	9.63	10.4	11.1						
H	16	13.2	15.2	17.0	18.6	20 1	21.5	12.6	14.5	16.2	17.8	19 2	20.5	11.2	13.0	14.5	15.9	17 1	18.3						
H	20	19.2	22.2	24.8	27.2	29.4	31.4	18 4	21.2	23.7	26.0	28 1	30.0	16.4	18 9	21.2	23.2	25.0	26 8						
M	21	18.6	21.5	24.0	26.3	28.4	30.4	18 1	21.0	23.4	25.7	27 7	29 6	17.5	20.2	22.5	24.7	26.7	28.5						
M	26	22.9	26.5	29.6	32.5	35.1	37.5	22.4	25 8	28.9	31.6	34 2	36.5	21.5	24.8	27.8	30.4	32.9	35.1						
M	34	29.4	34.0	38.0	41.6	45.0	48.1	28.7	33 1	37.1	40.6	43 8	46.9	27 6	31.9	35.6	39.0	42.2	45 1						
M	42	36.4	42.0	47.0	51.4	55 6	59.4	35.5	40 9	45.8	50 1	54 2	57.9	34 1	39.4	44 0	48.2	52 1	55.7						

BALANCED PORT THERMOSTATIC EXPANSION VALVES

BF-EBF	AA	0.65	0.75	0.84	0.92	0.99	1.06	0.63	0.73	0.82	0.90	0.97	1.03	0.55	0.64	0.71	0.78	0.85	0.90						
BF-EBF	A	1.39	1.60	1.79	1.96	2.12	2.26	1.35	1.56	1.74	1.91	2.06	2.21	1.18	1.36	1.52	1.67	1.80	1.93						
BF-EBF	B	2.42	2.80	3.13	3.43	3.70	3.96	2.36	2.73	3.05	3.34	3.61	3.86	2.07	2.39	2.67	2.92	3.16	3.38						
BF-EBF	C	4.50	5.20	5.81	6.37	6.88	7.35	4.39	5.07	5.67	6.21	6.70	7.17	3.84	4.43	4 96	5.43	5 86	6.27						
EBS	11	10.0	11.5	12.9	14.1	15.2	15.3	9.22	10.6	11.9	13.0	14 1	15.1	7.64	8.82	9.86	10.8	11.7	12.5						
O	15	13.0	15.0	16.8	18.4	19.8	21.2	12.0	13 9	15.5	17.0	18 4	19 6	10.1	11.7	13 0	14.3	15.4	16.5						
O	20	19.2	22.2	24.8	27.2	29.4	31.4	17.8	20.6	23.0	25.2	27.2	29.1	14.9	17.2	19.3	21.1	22.8	24 4						
O	30	26.4	30.5	34.1	37.4	40.3	43 1	24.5	28 2	31.6	34.6	37 4	39 9	20.5	23.7	26.5	29.0	31.3	33.5						
O	40	34.9	40.3	45.1	49.4	53.3	57.0	33.7	38 9	43.5	47.6	51 4	55.0	24.8	28.6	32.0	35.1	37.9	40.5						
O	55	47.6	55.0	61.5	67.4	72.8	77.8	46.0	53 1	59.3	65.0	70.2	75 1	33.8	39.1	43.7	47.9	51.7	55.3						
O	70	63.2	73.0	81.6	89.4	96.6	103	61.0	70 4	78.8	86.3	93.2	99.6	44.9	51.9	58.0	63.5	68.6	73.3						
V	52	45.0	52.0	58.1	63.7	68.8	73.5	43.5	50 2	56.1	61.5	66.4	71 0	41.8	48 3	54 0	59 1	63.9	68 3						
V	70	63.2	73.0	81.6	89.4	96.6	103	61.0	70 4	78.8	86.3	93.2	99.6	58.7	67.8	75 8	83.0	89.6	95.8						
V	100	86.6	100	112	122	132	141	83.6	96.5	108	118	128	136	80.4	92.8	104	114	123	131						
W	135	124	143	160	175	189	202	119	138	154	169	183	195	115	133	148	163	176	188						
W	180	156	180	201	220	238	255																		

REFRIGERANT LIQUID TEMPERATURE CORRECTION FACTORS	Refrigerant Liquid Temperature °F.	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°	110°	120°	130°	140°
	Correction Factor	1.56	1.51	1.45	1.40	1.34	1.29	1.23	1.17	1.12	1.06	1.00	0.94	0.88	0.82	0.76

These factors include corrections for liquid refrigerant density and net refrigerating effect and are based on an average evaporator temperature of 0°F. However they may be used for any evaporator temperature from – 40°F to 40°F since the variation in the actual factors across this range is insignificant.

EXAMPLE: Actual capacity of nominal 5 ton valve at 0°F evaporator, 175 psi pressure drop and 120°F liquid temperature = 5.86 tons × 0.88 = 5.16 tons.

Table 4-1. R-22 TXV capacity with liquid subcooled correction factors (Courtesy, Sporlan Valve Company)

will determine if an external equalizer is required (see section on externally equalized TXVs earlier in this chapter).

5. Select the body type according to the style connections desired. These include flare, solder, and flange connections. Brass, bronze, and gray cast iron are some other options. Different connection angles are also available (see Figure 4-10).

6. Select the selective thermostatic charge according to the design evaporating temperature from Table 4-2.

Example 4-1
Select a TXV for an R-22 air conditioning system with the following criteria:

- Design evaporator temperature = 40°F
- Design condensing temperature = 105°F
- Refrigerant liquid temperature = 90°F
- Design system capacity = 2 tons
- Condensing pressure = 211 psig
- Evaporating pressure = 69 psig
- Liquid line, suction line, and accessory losses = 7 psi
- Distributor and tube losses = 35 psi
- Refrigerant liquid correction factor = 1.06 (90°F liquid at TXV)

Solution 4-1
It is important to note that since there is a distributor used on the evaporator, an external equalizer valve must be used due to the large pressure drop created by the distributor.

First determine the available pressure drop across the TXV, which is the condensing pressure minus the evaporating pressure:

211 psig - 69 psig = 142 psig

From that total, subtract the liquid line, suction line, and accessory losses and the distributor and tube losses:

142 psig - 7 psi - 35 psi = 100 psi
(100 psi is the actual pressure drop across the TXV)

Using Figure 4-22, the valve capacity is as follows:

At a 100 psi pressure drop across the valve and a 40°F design evaporating temperature, the rated capacity of 2 tons exactly meets the design system capacity of 2 tons. However, we must take into consideration the 90°F liquid correction factor, which will increase the capacity rating of the 2 ton rated TXV to 2.12 tons (see below).

(2.00 rated tons)(1.06 liquid correction factor) = 2.12 tons

REFRIGERANT	AIR CONDITIONING OR HEAT PUMP	COMMERCIAL REFRIGERANT +50°F to −10°F	LOW TEMPERATURE REFRIGERANT 0°F to −40°F	EXTREME LOW TEMPERATURE REFRIGERANT −40°F to −100°F
12	FCP60	FC	FZ FZP	—
22	VCP100 and VGA	VC	VZ VZP40	VX
134a	JCP60	JC	—	—
502	RCP115	RC	RZ RZP	RX
717	REFER TO THE SECTION ON AMMONIA REFRIGERATION IN BULLETIN 10-9			

Table 4-2. Selective charges (Courtesy, Sporlan Valve Company)

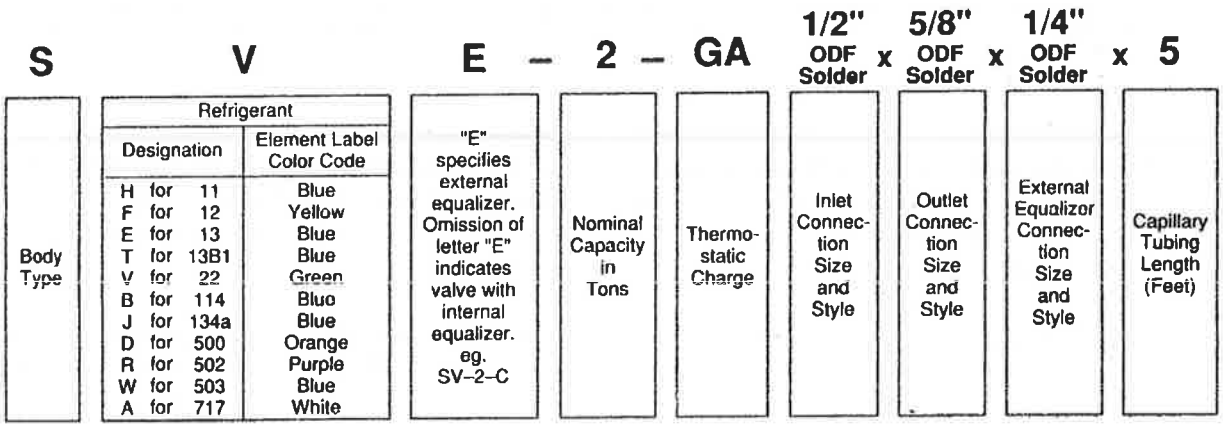


Figure 4-22. TXV selection (Courtesy, Sporlan Valve Company)

Remember, we want to select a valve with a rated valve capacity that equals or slightly exceeds the design tonnage rating of the system, which is 2 tons.

The valve that is rated for 2.12 tons, after figuring in the liquid correction factor, does slightly exceed the design tonnage of 2.0 tons. The choice would then be a 2.12 ton rated capacity valve, which would be a 2.00 ton nominal capacity valve in Table 4-1.

All that is left to do is to specify a body type; type of refrigerant; external or internal equalizer; thermostatic element charge type; inlet, outlet, and external equalizer connection sizes; and capillary tube length for the thermostatic remote bulb (see Figure 4-20).

Automatic Expansion Valves (AXVs)

The AXV is located between the liquid line and the evaporator. Its main function is to maintain a constant pressure in the evaporator under all evaporator loading conditions without regard to evaporator superheat; in fact, there is no remote bulb or feedback mechanism to measure evaporator superheat. The AXV either starves or fills out the evaporator coil with refrigerant in response to different heat loads on the evaporator. Because of the AXV's constant pressure characteristic, it cannot be used with a low pressure motor control.

The AXV is made of a valve casing, pressure bellows or diaphragm, needle and seat, spring, adjustment stem or screw, and a liquid inlet strainer. The two forces that throttle the AXV open and closed are the spring and the evaporator pressure, Figure 4-23. Constant evaporator pressure is maintained by the interaction of the spring and evaporator pressures. The evaporator pressure is a closing force, and the spring pressure is an opening force. During the compressor on cycle, the valve will hold the evaporator pressure in equilibrium with the spring pressure. The valve stem or adjustment screw should be adjusted for the desired evaporator pressure. As the compressor cycles off, the valve will close due to increasing evaporator pressure from residual refrigerant in the evaporator boiling off, and refrigerant flow to the evaporator will cease.

AXV efficiency is considered poor when compared to other metering devices. This is because in order to keep a constant pressure in the evaporator, the rate of vaporization in the evaporator must be kept constant under all evaporator loads. At heavy evaporator loads, the refrigerant delivered to the evaporator must be throttled back because of the increased rate of refrigerant vaporization in the evaporator, momentarily causing a higher evaporator pressure. This throttling effect causes an inactive evaporator with an abundance of superheat, Figure 4-24; however, a constant evaporator pressure is still maintained.

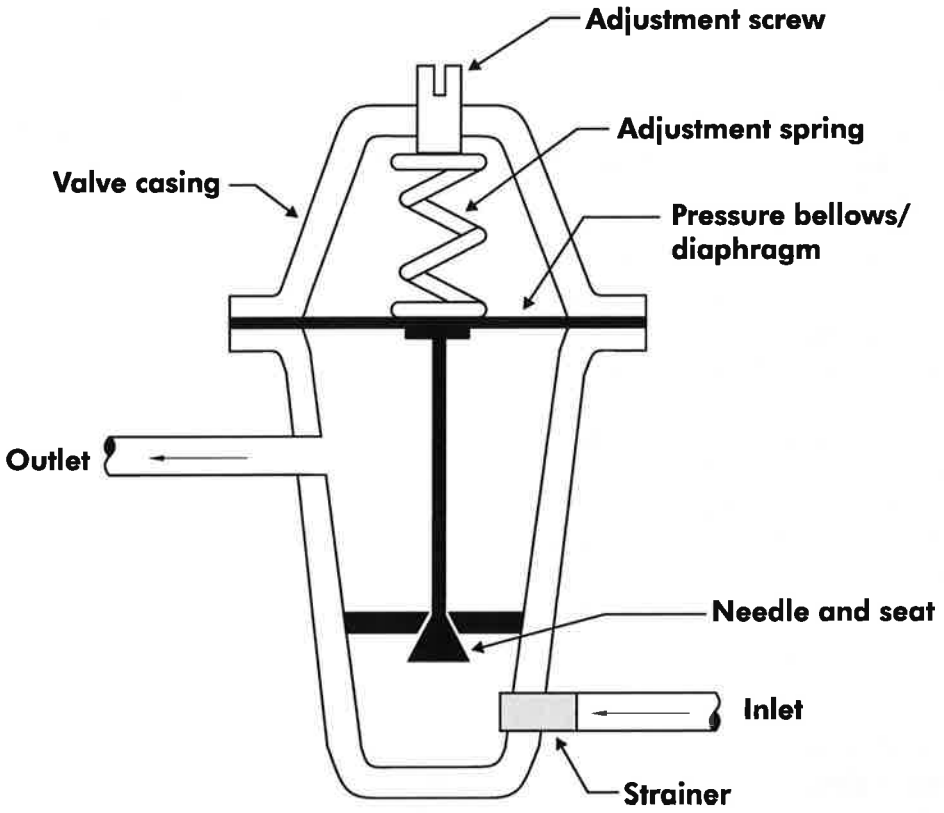


Figure 4-23. Spring and evaporator pressure on an AXV

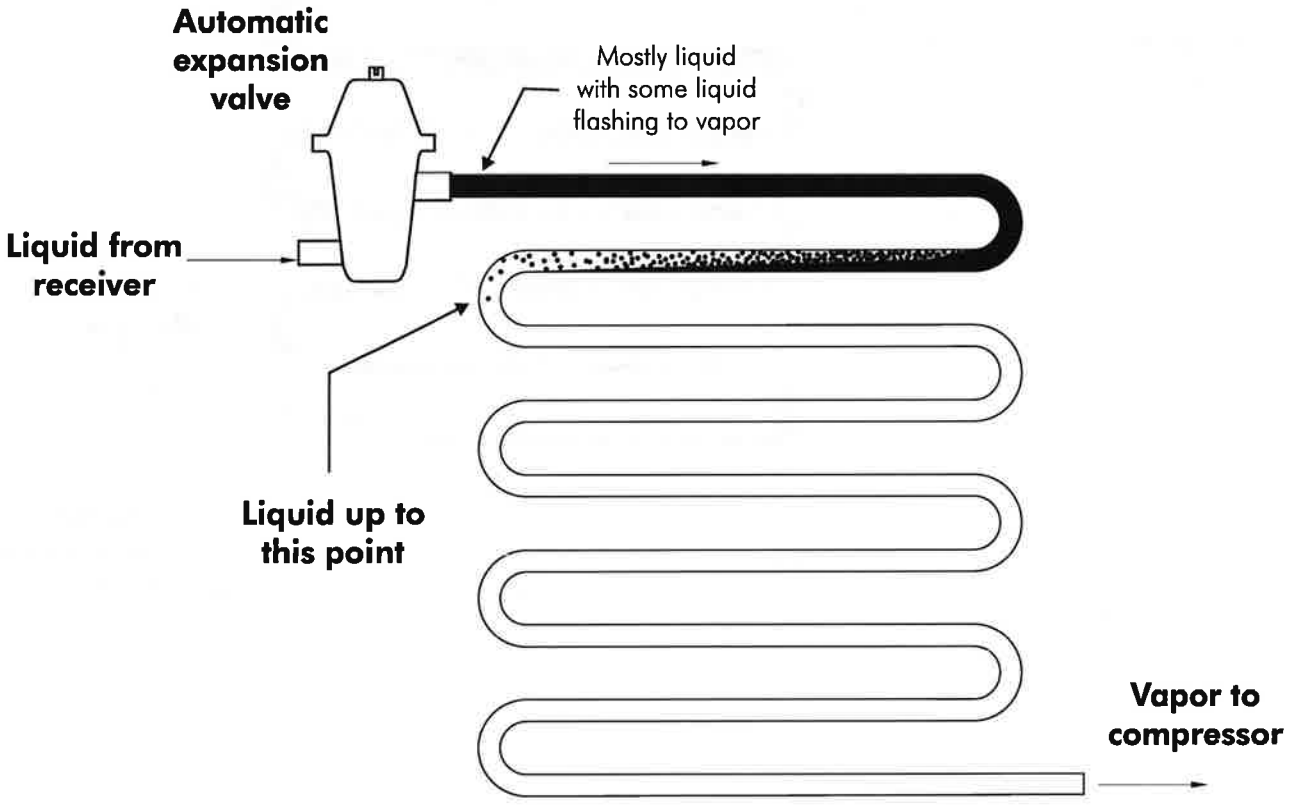


Figure 4-24. Inactive evaporator with an abundance of superheat

At low loads, the evaporator pressure momentarily decreases, causing the spring force to throttle the valve more open. This maintains the valve's constant pressure characteristics. At a low evaporator load, more of the evaporator surface must be wetted with refrigerant in order to maintain a constant rate of vaporization, Figure 4-25. This lowers the evaporator superheat. If the load is permitted to be lowered too far without shutting the compressor off, the valve could open too far and flood the compressor in order to try and maintain a constant evaporator pressure. The thermostat should be adjusted to shut the compressor off before flooding of the entire evaporator occurs. The AXV should be adjusted to maintain an evaporator pressure that corresponds to the lowest evaporator temperature desired through the entire running cycle of the compressor.

AXVs are usually found on smaller equipment with constant evaporator loads, including ice cream freezers and makers and drinking fountains. AXVs are often used to prevent products or water from freezing.

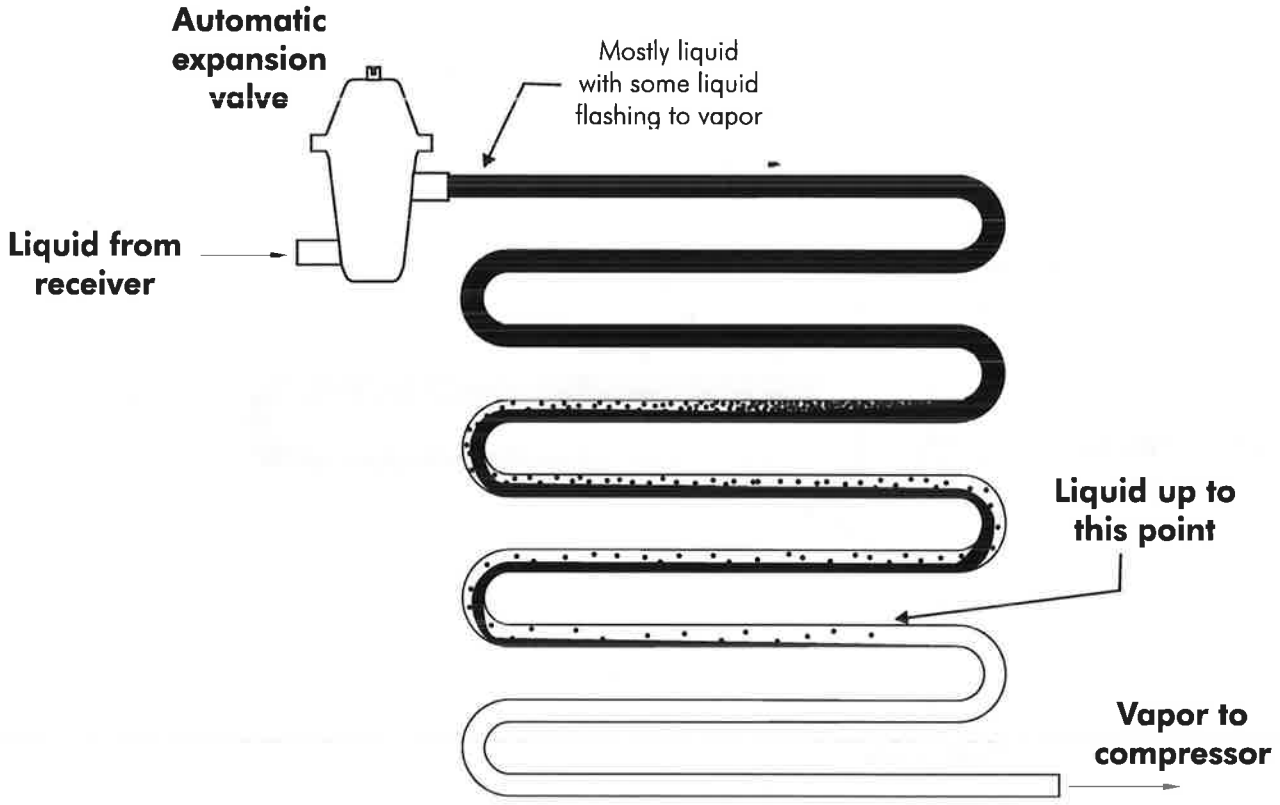


Figure 4-25. AXV maintaining constant rate of vaporization

CHAPTER FOUR

System Charge

One of the most puzzling questions a technician will ask is how much refrigerant should be put in a refrigeration system when charging. There is really no straightforward answer, because all refrigeration systems differ in the amount of charge they hold. However, there are guidelines, charts, and techniques all technicians should follow when charging a system. The techniques discussed in this chapter do not apply to automotive or any transportation air conditioning system charging.

Charging TXV/Receiver/Sightglass Refrigeration Systems

Most large commercial refrigeration systems employ TXVs as metering devices. Because of the throttling action of the TXV metering device to satisfy refrigeration loads, a liquid receiver must be employed to act as a liquid refrigerant reservoir. Liquid receivers also act as storage vessels to store the condenser charge variations between summer and winter operations. When the TXV is throttled down at lower loads, the receiver will contain more liquid. The TXV throttles open more at higher heat loads and draws liquid refrigerant from the receiver. The remote bulb sends more of an opening pressure to the TXV diaphragm, which opens the valve more. This is why it is of utmost importance to put the refrigeration equipment under a high load when charging with refrigerant. High loads ensure that the TXV metering device is fully open and delivering the maximum amount of refrigerant to the evaporator. At these higher loads, the receiver will be at its lowest level. High loads can be achieved by placing a false heat load on the evaporator.

At higher loads, a sightglass located in the liquid line will bubble if undercharged. The sightglass in the liquid line may not bubble when a system is undercharged if the TXV is throttled partly closed, the receiver has some liquid, and the system is at a lower heat load. Unless severely undercharged, a sightglass will only bubble under higher loads. The service technician must make sure the sightglass does not bubble under high and low loads in order for the system to be properly charged.

Service technicians often confuse a bubbling sightglass with a low flow rate sightglass. If undercharged and at a high load, the receiver will be at its lowest level and the sightglass will have entrained refrigerant gas bubbles in it, Figure 5-1. If the system is charged correctly but experiences a very low heat load on the evaporator, a low flow rate sightglass, not a bubbling sightglass, may be seen. Low flow rate sightglasses do not have refrigerant gas bubbles entrained in the flowing liquid. A low flow rate sightglass only partially fills the volume of the sightglass and does not indicate an undercharge, Figure 5-2. The low flow rate of refrigerant through the system is caused from

the lower loads on the evaporator. Lower loads mean higher compression ratios and lower volumetric efficiencies. The density of the return gas to the compressor is also decreased at lower loads and lower suction pressures, causing low mass flow rates of refrigerant. Horizontal liquid lines with sightglasses encounter this phenomenon much more often than vertical liquid lines with sightglasses. Gravity will force the liquid in the liquid line to settle at the lowest point, and this can entirely fill a sightglass volume even at low flow rates.

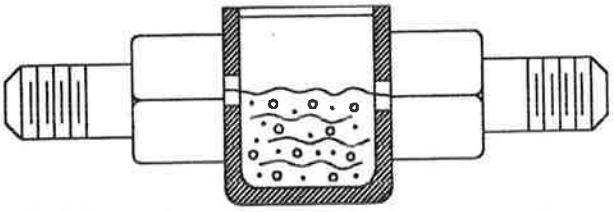


Figure 5-1. Bubbling sightglass (Courtesy, Todd Rose, Ferris State University)

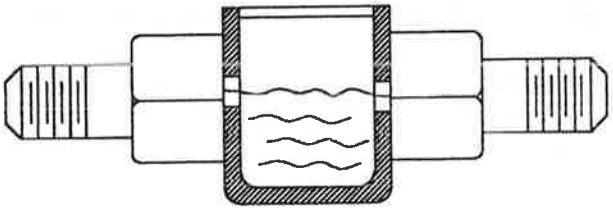


Figure 5-2. Low flow rate sightglass (Courtesy, Todd Rose, Ferris State University)

In these cases, the technician must make sure the liquid line is full of liquid before and after the sightglass. This can be done with an electronic sightglass, Figure 5-3. An electronic sightglass determines whether there is vapor entrained in the liquid. This assists the technician in charging the system and ensures 100% liquid at the metering device. Electronic sightglasses are especially helpful when the refrigeration system has no sightglass. The electronic sightglass works when its sensors are clamped about 1-1/2" apart on the liquid line. The technician then waits for beeping sounds or flashing lights to indicate the presence of vapor and liquid in the liquid line. Electronic sightglasses can also be used on the evaporator outlet or start of the suction line to let the technician know if there is liquid entrained with the vapors. Electronic sightglasses are also used extensively in automotive and commercial/residential air conditioning system charging.

Charging Rules

The following rules should be used when charging TXV/receiver/sightglass systems:

1. Always charge a TXV/receiver/sightglass system under a high load. Remember, systems with 50 lb or more of refrigerant that leak significantly must be leak checked and repaired (see Chapter Ten). Once the leak is found, the system refrigerant recovered, and the leak repaired, the system is ready to be evacuated. If the vacuum is pulled on a refrigeration system with a deep vacuum pump, once the desired vacuum is reached and the pump is isolated from the system, charge liquid refrigerant into the high side of the system until high and low side pressures equalize and refrigerant stops flowing. It is preferable to charge refrigerant into the receiver if valving allows. Putting liquid refrigerant into the low side of the system can damage the compressor at start-up and



Figure 5-3. Electronic sightglass (Courtesy, TIF Instruments, Inc.)

dilute the oil in the crankcase with refrigerant, causing oil flash and scored bearings.

2. Once liquid has been charged into the high side of the system and system pressures have equalized, let the system set idle for about 10 minutes. This helps vaporize any liquid the TXV has released into the evaporator while charging. After a 10 minute wait, start the system and monitor the amp draw. Keep the doors open and the system under a high load to allow the TXV to throttle open fully. This will draw a maximum amount of liquid from the receiver. Let the system run for a while at this high heat load to reach an equilibrium. If the sightglass bubbles, charge refrigerant vapor into the low side of the system until the sightglass stops bubbling. Refrigerant vapor can be charged into the low side of any running system as long as the technician knows that the undercharge is not caused by a leak. Remember, all leaks should be fixed, and systems with 50 lb and more of refrigerant cannot be left to leak or severe fines will be paid (see Chapter Ten). *Note: If the refrigerant is a newer refrigerant that has a temperature glide associated with it, liquid refrigerant must be throttled through the low side to avoid fractionation. The proper technique is to employ a sightglass and valve, which allow liquid to be removed from the cylinder while manually adjusting the valve, causing enough restriction to "flash" the liquid before it enters the suction valve. This flashing will be visible through the sightglass, Figure 5-4. When properly charged, the sightglass should not bubble under high or low loads.*

3. Take evaporator superheat, compressor superheat, and condenser subcooling readings. Evaporator superheat reading guidelines are listed in Table 5-1. If evaporator superheat settings are not within the guidelines, adjust the TXV superheat spring. Always give the TXV enough time to react to the change before taking the next reading.

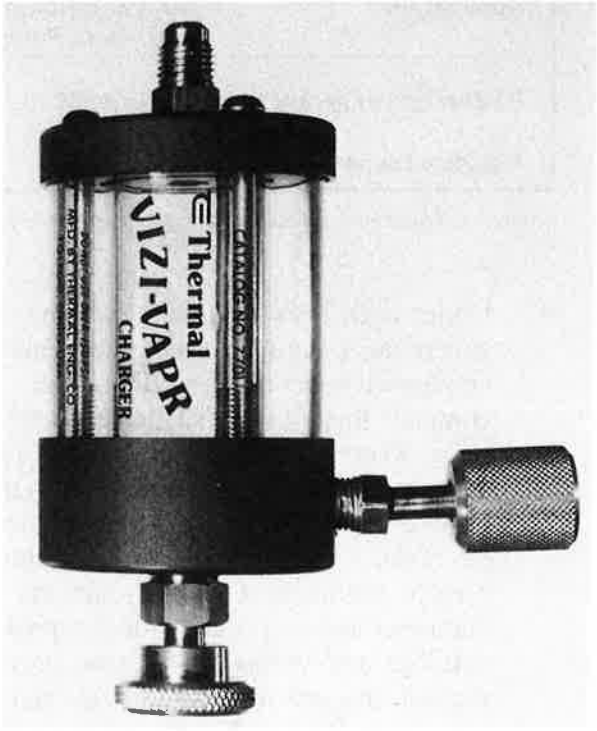


Figure 5-4. Charging device that allows visibility of both liquid and vapor while charging (Courtesy, Thermal Engineering Company)

Make sure condenser subcooling is at least 6°F. This is just a starting figure, because actual subcooling amounts will vary depending on the type of refrigerant used and how much pressure drop the subcooled liquid will see from the condenser outlet to the inlet of the TXV. This ensures that the condenser is producing enough liquid to deliver a solid column of 100% liquid to the TXV.

Compressor superheats should always be between 20° and 30°F. This ensures that the compressor will not see any liquid refrigerant at lower loads. This superheat range will also prevent compressor overheating and high amp draws.

Application	Air Conditioning and Heat Pump	Commercial Refrigeration	Low Temperature Refrigeration
Evaporator temperature	40° to 50°F	0° to 40°F	-40° to 0°F
Suggested superheat setting	8° to 12°F	6° to 8°F	4° to 6°F

Table 5-1. Guidelines for evaporator superheat settings

4. Under high heat load, take evaporator superheat, compressor superheat, and condenser subcooling readings and compare them to the suggested guidelines. Record both the condensing and evaporating pressure. Monitor the amp draw of the compressor with an ammeter. Take a voltage reading at the compressor terminals. Check it with the manufacturer amp curve for the pressure readings and voltage taken (see previous section on performance curves), and make sure it is within the range (±5%) for the pressures and voltage recorded.

5. Record the condensing and evaporating pressure of the system. Compare the amp reading of the compressor to the manufacturer amp curves for the pressures recorded. Make sure the amp draw is within range (±5%) on the curves for the two pressures and voltage recorded.

After applying these basic rules, a TXV/receiver/sightglass refrigeration system can be charged with the proper amount of refrigerant.

Charging Capillary Tube or Fixed Orifice Refrigeration Systems

Capillary tube systems vary somewhat from TXV systems in their charging procedures. Most capillary tube systems do not have sightglasses in their liquid lines. Capillary tube systems have fixed orifices and do not throttle open and closed like TXV or AXV metering devices, so a receiver is usually not needed to hold any liquid coming from the condenser. This is one reason why a capillary tube system is a critically charged system.

Critically charged systems are systems that specify an exact amount of refrigerant charge. These charges are usually specified to the quarter ounce. The manufacturer nameplate, usually located on the condensing unit, evaporator faceplate, or unit frame, will specify the critical charge. Manufacturers critically charge capillary tube systems in order to achieve peak system efficiency and performance. Capillary tube systems are also critically charged to prevent floodback of liquid refrigerant under low heat loads or high ambients exposed to the condenser.

If the system has been running for some time and an undercharge is suddenly suspected, a leak probably exists. It is recommended procedure to leak check the system. For systems that have over 50 lb of refrigerant for commercial and industrial refrigeration, leakage rates of more than 35% annually are considered substantial leaks and must be repaired. If the equipment is a chiller or comfort cooling air conditioner containing over 50 lb of refrigerant, a 15% annual leakage rate is considered substantial and must be repaired. Once the refrigerant is recovered and the leak is repaired, a deep vacuum must be pulled on the system. After successful evacuation, the nameplate charge must be measured and charged into the system. Most capillary tube systems are small systems, and it is good practice to recover, leak check, evacuate, and then critically charge the system. These procedures take very little time if systems are small. This eliminates guesswork as to whether the system has the right amount of charge.

Charging from a Vacuum

Once the desired vacuum is reached and the vacuum pump is isolated from the system, add

the specified nameplate amount of liquid refrigerant to the high side of the refrigeration system through a gauge manifold. Adding this refrigerant can be done with a programmable electronic charging device, Figure 5-5, or a critical charging cylinder, Figure 5-6, designed for capillary tube systems. Often, a combination critical charging cylinder, vacuum pump, and manifold set can be purchased, Figure 5-7. Once the critical amount of liquid refrigerant is charged into the system, let the liquid charge bleed from the high side of the system through the capillary tube. This will cause a good portion of the liquid to vaporize and end up in the evaporator as a vapor. Evidence of this can be seen by letting the system idle for about 10 minutes after liquid charging to the high side. Within a minute, the low side pressure will gradually rise, demonstrating that the capillary tube is not plugged and that refrigerant is traveling through it. Figure 5-8 shows how to charge a system with an electronic, programmable charging device.



Figure 5-5. Programmable electronic charging device (Courtesy, TIF Instruments, Inc.)

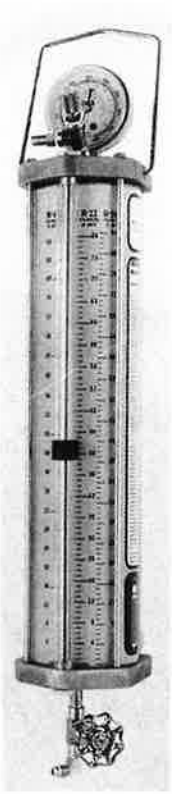


Figure 5-6. Critical charging cylinder (Courtesy, Thermal Engineering Company)

Once the low side pressure rises to equal the high side pressure, the system can be started. It is important to allow the high and low side pressures to equalize, because this allows the compressor motor to start more easily, Figure 5-9. Capillary tube systems are often not equipped with start capacitors (hard start kits), and system pressures must equalize before attempting to start the compressor. This will prevent locked rotor situations and often tripped compressor overloads.

To illustrate the need for a critically charged system, consider a 3-ton system that calls for 5 lb of refrigerant to be critically charged. A 10% undercharge is only 8 oz of refrigerant, but this undercharge will reduce system efficiency by 20%. A 23% undercharge will reduce system efficiency by 50%. The correct charge is critical, so do not to use cheap scales, long charging hoses, or equipment that has not been kept up. *Note: It is of utmost importance to weigh in the exact charge into a capillary tube or fixed orifice system.*

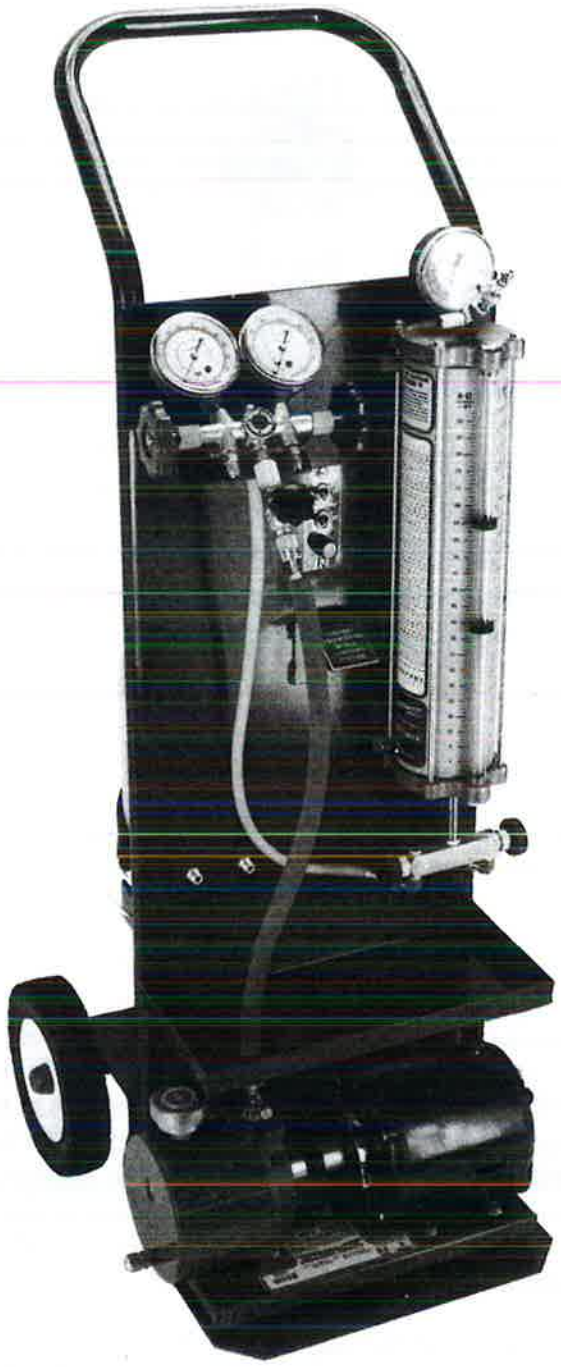


Figure 5-7. Portable combination charging cylinder, vacuum pump, and manifold gauge set (Courtesy, Thermal Engineering Company)

Compressor motors employing capillary tube systems are usually low starting torque motors that cannot start with large pressure differences between high and low side pressures. Not waiting for the system to equalize will cause the compressor to lock rotor and open its overload protection device. The overload protector will continue to open and reset until the pressures are just about equal, then the low starting torque motor will start.

Once the system starts, take a compressor amp reading and measure voltage at the compressor terminals. Make sure the amp reading is the compressor only and not the total amps of the system. Compare the amp reading to the manufacturer compressor curve or system curves to make sure the system is charged and running as efficiently as possible (see Chapter Three). System performance curves are preferred in this case, because compressor curves reveal only compressor performance, not system performance. If the system is operating correctly or potentially correctly, then compressor curves can be used.

Always remember to correct for voltage when using compressor curves. If undercharged, add refrigerant vapor to the low side of the refrigeration system while it is running. *Note: If the refrigerant is a newer refrigerant with a temperature glide, liquid must be throttled in the low side of the system with the compressor running to avoid fractionation. The proper technique is to employ a sightglass or a charging device, which allows liquid to be removed from the system while manually adjusting a valve. This causes enough restriction to "flash" the liquid before it enters the suction valve.*

An overcharge of refrigerant will cause elevated head and suction pressures. The amp draw will also be much higher than the curves indicate it should be. An undercharge of refrigerant will cause the head pressure and suction pressure to go down. The amp draw will also be much lower than the curves indicate if undercharged. A filter drier change is always recommended before recharging a system. Often, the replacement filter drier will be larger than the original drier. The larger drier will hold more liquid refrigerant, and the system may be a bit undercharged even if the nameplate charge is used. This is why it is important to use performance curves after charging.

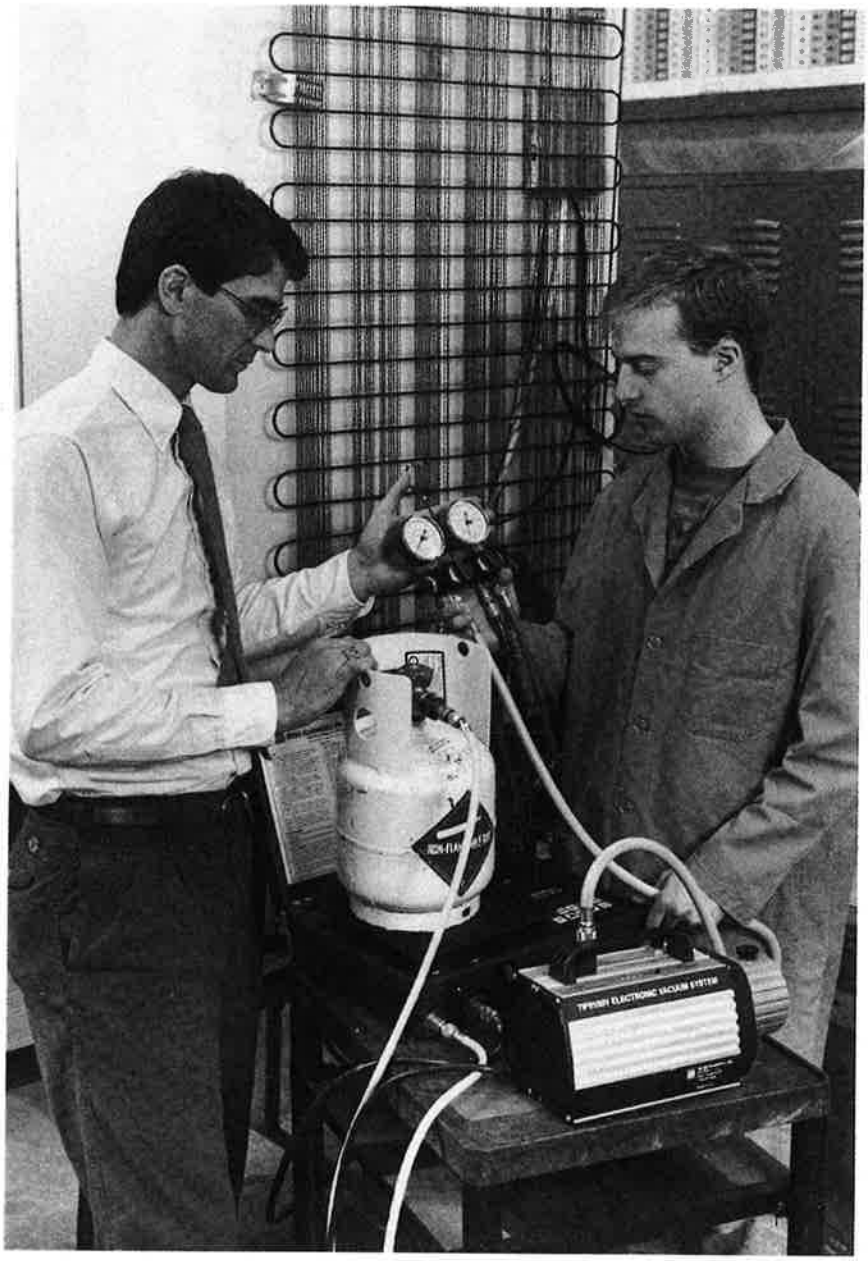


Figure 5-8. Charging a system using a programmable charging scale manufactured by TIF Instruments, Inc. (Photo by Bill Bitzinger, Office of University Communication Services, Ferris State University)

Charging Capillary Tube or Fixed Orifice Air Conditioning Systems

Charging an air conditioning comfort system is much different than charging a refrigeration system. Air conditioning systems rely on certain volumetric amounts of air flow referred to as cubic feet per minute (cfm) across their evaporator coils. Air conditioning processes are both sensible and latent processes, which take both heat and moisture from the air. Because of this, both wet bulb (wb) and dry bulb (db) temperatures, along with air flows (cfm) and compressor superheats are needed to charge these systems.

The most accurate method of charging a capillary or fixed orifice system is to weigh the charge into an empty system; however, time does not always allow recovering and evacuat-

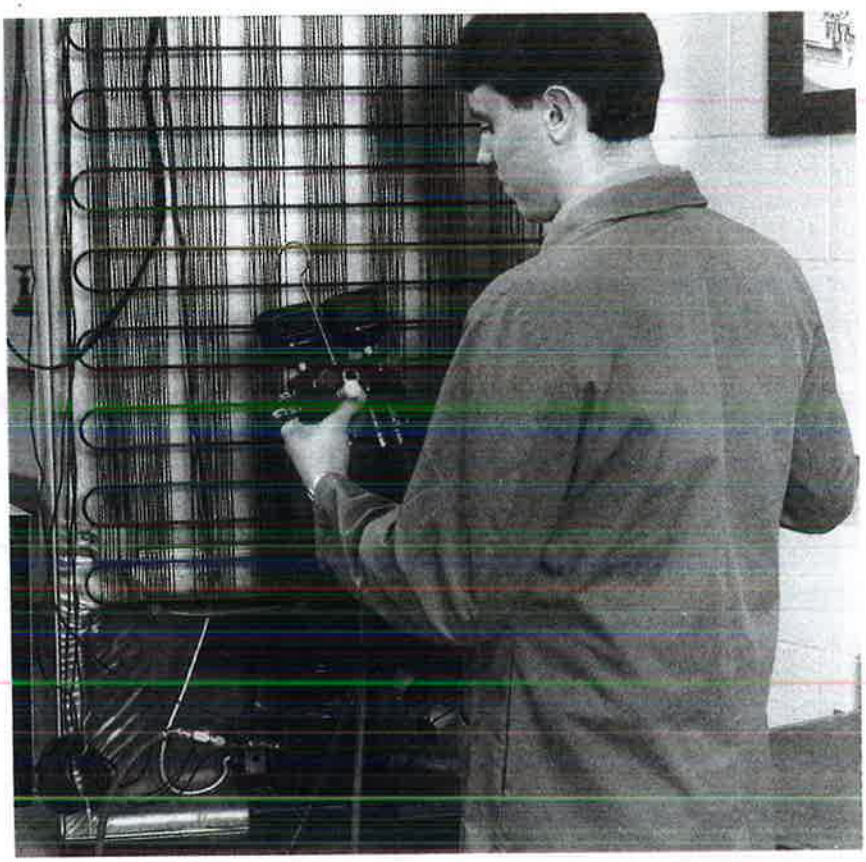


Figure 5-9. Waiting for system pressures to equalize (Photo by Bill Bitzinger, Office of University Communication Services, Ferris State University)

ing before charging. The next most efficient and accurate way of charging a capillary tube system is to use a superheat charging table or curve, Figure 5-10. Charging by the compressor superheat method best balances the system's ability to absorb heat with the available heat load on the evaporator. Superheat charging can be used on capillary or fixed orifice metering devices when charging in ambients above 60°F. Compressor superheat in these types of systems will vary depending on the air conditions flowing over the condenser and evaporator coils. The amount of superheat at the compressor must be adjusted to meet these varying conditions. The superheat charging method shows compressor superheat for various entering air conditions at the evaporator and condenser. Remember that compressor superheat will vary as conditions vary, and no one superheat value can be used for all conditions. If this charging method is performed correctly, the system will work safely and efficiently at all conditions as long as the ambient is over 60°F at the condenser.

To use the superheat charging method perform the following steps:

1. Make sure the evaporator has the rated cfm of air flow across its coil (400 cfm/ton ±10% is a good rule). Velometers and/or manometers with proper equations will assist in calculating system cfm. Check to see if blower assemblies and coils are free of dirt and that the duct system is sized properly. Remember, the unit manufacturer controls condenser air volume by design, and the installer and designer control evaporator air volume. Fan speed must be adjusted to overcome design and installation variables to obtain 400 cfm/ton ±10%.

CONDENSER AIR INLET TEMP.-°F-D.B.	EVAPORATOR AIR INLET TEMP.-°F-W.B.											
	54	56	58	60	62	64	66	68	70	72	74	76
60	13	17	18	20	24	26	28	30				
65	11	13	17	17	18	22	25	28	30			
70	8	11	12	14	16	18	22	25	28	30		
75		7	10	12	14	16	18	23	26	28	30	
80			6	8	12	14	16	18	23	27	28	30
85				6	8	12	14	17	20	25	27	28
90					6	9	12	15	18	22	25	28
95						7	11	13	16	20	23	27
100							8	11	14	18	20	25
105							6	8	12	15	19	24
110								7	11	14	18	23
115									8	13	16	21

Figure 5-10. Superheat charging table (Courtesy, Addison Products Company)

To measure cfm across the evaporator coil, use the following equation (TD is the temperature difference across the unit coil or heat exchanger):

$$cfm = \frac{Btu}{(1.08)(TD)}$$

For fossil fuel furnaces in the heating mode, the formula becomes:

$$cfm = \frac{(Btuh\ input)(Combustion\ efficiency)}{(1.08)(TD)}$$

For single-phase electric heating elements, the formula becomes:

$$cfm = \frac{((Average\ volts)(Average\ amps))(3.414\ Btu/watt)}{(1.08)(TD)}$$

If the heating element is three phase, the formula becomes:

$$cfm = \frac{((Average\ volts)(Average\ amps))(3.414\ Btu/watt)(1.73)}{(1.08)(TD)}$$

Note: When measuring the average current, it must include the current of the heating element and fan motor. This is because the fan motor's heat also enters the air stream. In addition, temperature sensors must be placed out of the direct sight line of the heating elements so as not to include any radiant heating effects. This is usually done after the first elbow in the ductwork.

If a technician measures the wattage input to an electric heater coil or gas flow to a gas-fired furnace in the ductwork, the cfm can be calculated using the above formulas.

Subtract the static pressure (Sp) from the total pressure (Tp) in the ductwork, Figure 5-11, with an inexpensive U-tube water manometer to obtain air velocity pressure (Vp). The actual air velocity in feet per minute (fpm) can then be calculated using the following equation:

$$Velocity\ (fpm) = (\square\overline{Vp})(4005)$$

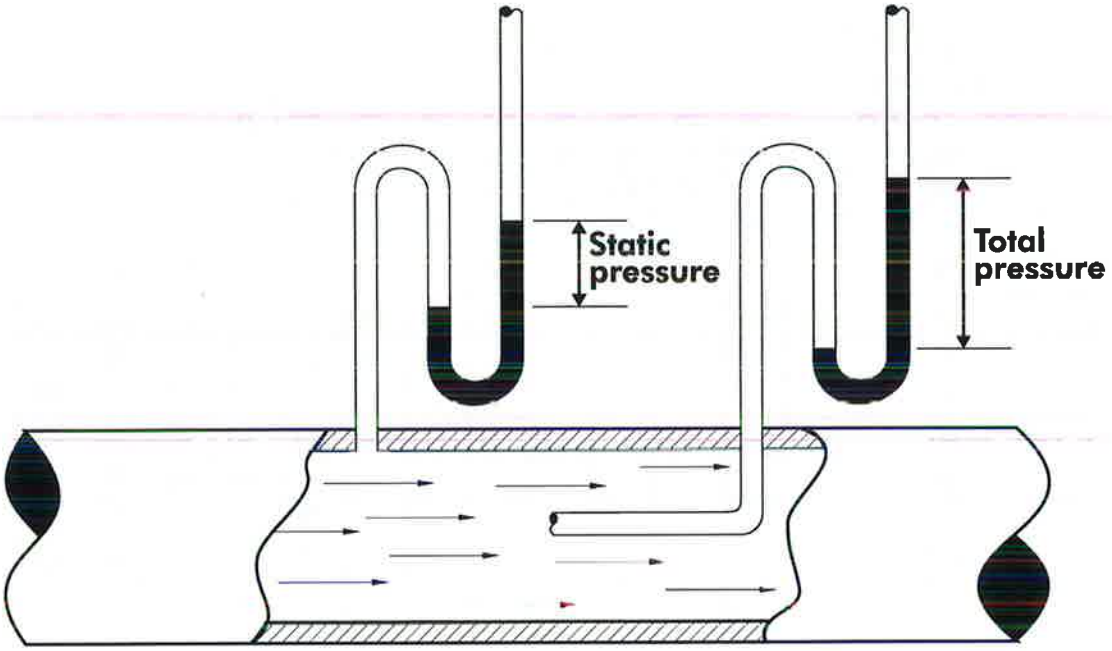


Figure 5-11. Measuring static and total pressure

Some instruments measure both total and static pressures at the same time and are connected to a delicate meter in such a way that the subtraction is done automatically, and the actual velocity of the air is read out in feet per minute (fpm), Figures 5-12 and 5-13. These instruments usually require the operator to traverse the duct with a long pitot tube at certain traverse points, Figure 5-14. An average velocity is then calculated.

Because the velocity pressure is a square root function, they cannot be first averaged together and then converted to velocity. This is because it is not mathematically possible to average a square root; a reading must be converted to velocity in feet per minute (fpm), then averaged together. The following table shows how to determine an average (Courtesy, Refrigeration Service Engineers Society):

Pitot tube reading	Calculation		Ft/min
.04	4005$\sqrt{.04}$	=	800
.05	4005$\sqrt{.05}$	=	895
.06	4005$\sqrt{.06}$	=	980
.07	4005$\sqrt{.07}$	=	1060
.05	4005$\sqrt{.05}$	=	895
.06	4005$\sqrt{.06}$	=	980
			5610

The average is as follows:

$$\frac{5610 \text{ fpm}}{6} = 935 \text{ fpm}$$

Once the air velocity in fpm is known, use the following equation to obtain unit cfm (area is the cross-sectional area of the air duct in square feet):

cfm = (Velocity in fpm) (Area in square feet)

Area for rectangular or square ducts = (length) (width)

Area for round or circular ducts = $(3.14)r^2$, where r = the radius of the round or circular duct

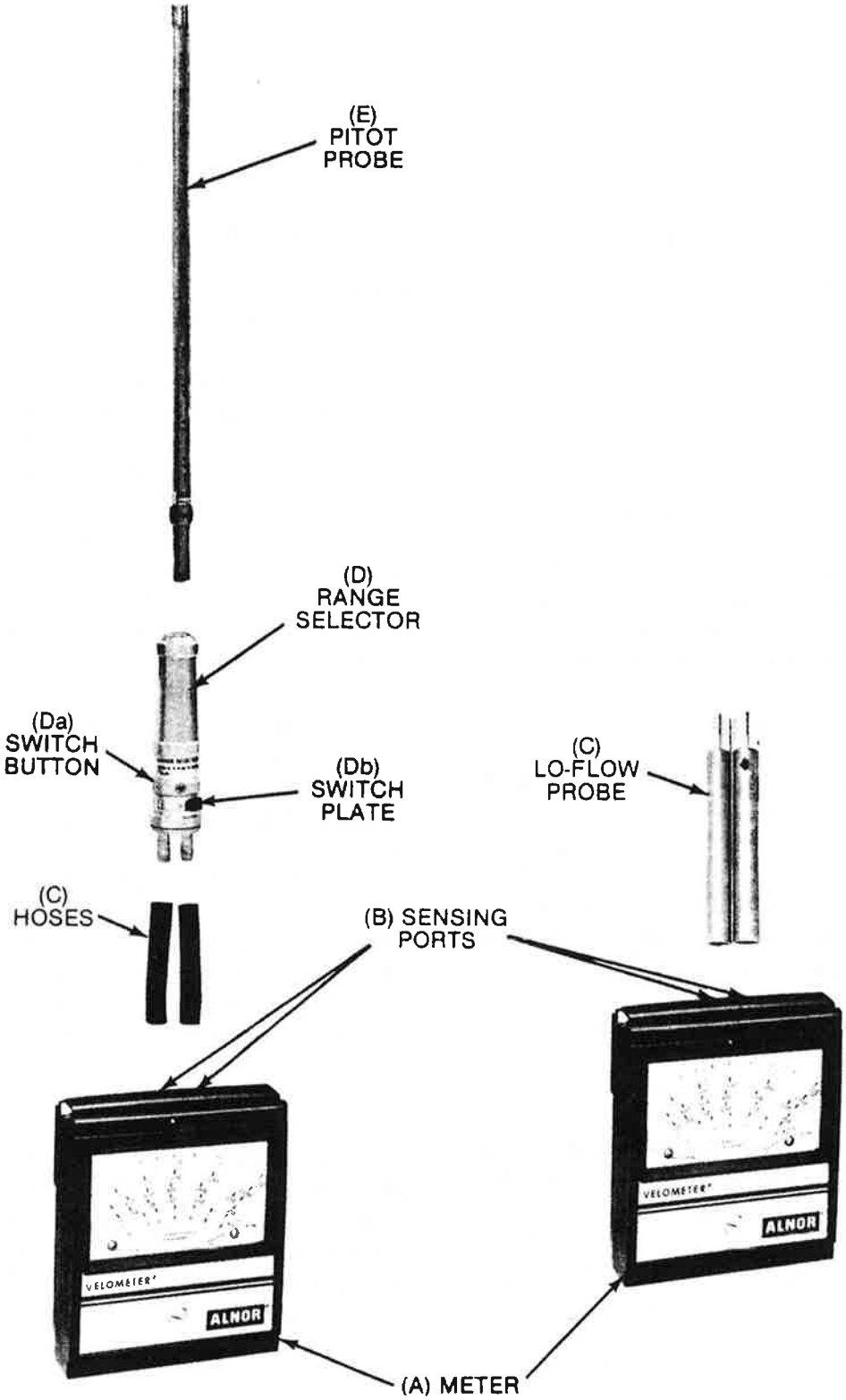


Figure 5-12. Air velocity reading instruments (Courtesy, Alnor Instrument Company)

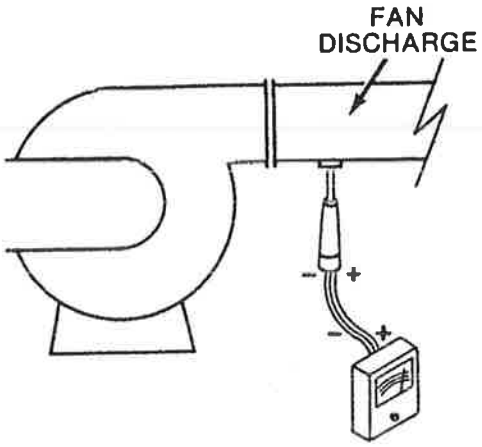


Figure 5-13. Air velocity reading instrument in discharge duct (Courtesy, Alnor Instrument Company)

Static pressures (Sp) may also be measured across a working air conditioning coil (wet coil), and cfm may be obtained directly from manufacturer tables based on cfm for coil air-side pressure drops, Figure 5-15. Always refer to manufacturer literature for specific cfm based on static pressure drops across coils.

2. Measure the compressor superheat at the compressor, Figure 5-16.

3. With the unit running and stabilized, measure condenser inlet air dry bulb (db) temperature and evaporator inlet air wet bulb (wb) temperature.

4. Using Figure 5-10, find the proper compressor superheat at the intersection of db and wb temperatures.

5. If the compressor superheat measured in Step #2 is too high, refrigerant must be added. If the compressor superheat is too low, refrigerant must be recovered from the system. *Caution: Wait at least 15 minutes after adding or removing refrigerant from the system before attempting to re-measure compressor superheat.*

Example 5-1
Determine if the charge is correct for an R-22 capillary tube or fixed orifice system with the following information:

- Condenser air inlet = 85°F
- Evaporator air inlet (wb temperature) = 66°F and 400 cfm/ton (±10%) flow rate
- Suction pressure measured at compressor = 60 psig (34°F), Figure 5-17
- Compressor inlet temperature = 56°F

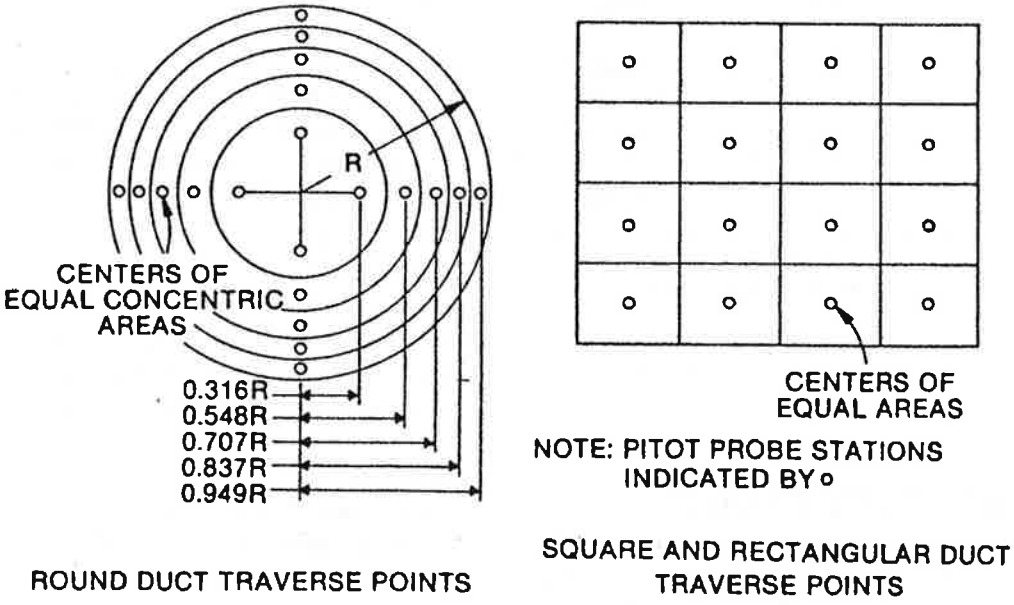
Solution 5-1
From the information given, it is possible to determine that the compressor superheat is 22°F (56° - 34°F). Looking at Figure 5-10, the correct superheat at the intersection of the condenser air inlet (85°F) and the evaporator air inlet (66°F) is 14°F. This system is running a compressor superheat of 22°F. The superheat is too high, so the system is undercharged. Refrigerant vapor must be added to the low side of the system while it is running. After charging is finished, let the system run for at least 15 minutes and check the compressor superheat again. *Note: If the refrigerant is a newer refrigerant that has a temperature glide associated with it, liquid refrigerant must be throttled through the low side to avoid fractionation. The proper technique is to employ a sightglass and valve, which allow liquid to be removed from the cylinder while manually adjusting the valve, causing enough restriction to "flash" the liquid before it enters the suction valve. This flashing will be visible through the sightglass, Figure 5-4. When properly charged, the sightglass should not bubble under high or low loads.*

As the technician adds refrigerant, the suction pressure should increase and the compressor inlet temperature should decrease or become colder because of less superheat. These two temperatures will come closer together as refrigerant is added. *Caution: If, during charging, the indoor wet bulb (evaporator air inlet) temperature changes more than 2°F or the outdoor ambient temperature changes more than 3°F, check the superheat table for a new superheat and charge to this new superheat.*

Measuring Duct Velocity

The velocity of an air stream in a duct is not unifom throughout the cross-section; air near the walls moves more slowly due to friction. Elbows, transitions and obstructions also cause variations in the velocity at any one cross-section.

To obtain the average velocity in ducts of 4" diameter or larger, drill a ½" diameter or larger hole in the duct and take a series of duct velocity readings (commonly referred to as a traverse), with the Pitot Probe, at points of equal area across the duct. A formal pattern of sensing points is recommended and these points are referred to as traverse point readings. Shown are recommended velocity reading point locations for traversing round and square (or rectangular) ducts.



ROUND DUCT TRAVERSE POINTS

SQUARE AND RECTANGULAR DUCT
TRAVERSE POINTS

In round ducts, take velocity readings at the center of equal concentric areas; take at least 20 readings along two diameters. In square or rectangular ducts, take a minimum of 16 and a maximum of 64 readings at centers of equal areas. Calculate the average of all readings.

For maximum accuracy, observe the following precautions:

1. Perform the traverse in a section of the duct where the air stream is as uniform as practical. This is generally a location of eight or more duct diameters of straight duct upstream from the traverse location.
2. Do not take the traverse near a duct elbow, transition or obstruction.
3. Make a complete, careful and accurate traverse and record the results on a worksheet.

Figure 5-14. Method for measuring air velocity in an air duct using a pitot tube traverse (Courtesy, Alnor Instrument Company)

Rated air flow		Cooling capacity Btu/hr
CFM	H_2O*	
820	.15	24,400
800	.15	24,600
1050	.20	30,000
800	.10	28,200
1050	.21	30,800
1200	.30	36,000
1200	.25	38,000
1200	.20	35,800
1325	.20	42,000
1230	.15	40,000

* Static pressure loss for add-on coils (wet coil) and available static pressure for duct system on blower coils.

Figure 5-15. Static pressure drop vs cfm for evaporator coil

Superheat Charging Curves

Manufacturers may vary the style of superheat charging curves they offer; however, the same underlying principle holds for all tables and curves. Figure 5-18 is an example of a superheat charging curve instead of a table. The curve is based on 400 cfm/ton air flow at 50% relative humidity across the evaporator coil. The steps to charge a system according to this curve are as follows:

1. Measure indoor db temperature. This is the return air at the air handler. *Note: Use wb temperature if the relative humidity is above 70% or below 20%.*
2. Measure outdoor db temperature at outdoor unit. This is air temperature at the condenser.
3. Measure suction pressure at the compressor, and convert to a temperature using a pressure/temperature chart.
4. Measure compressor temperature on suction line near compressor.
5. Calculate the amount of compressor superheat.
6. Find the intersection where the outdoor temperature and indoor temperature meet and read compressor superheat.

If the compressor superheat of the system is more than 5°F above what the chart reads, add refrigerant vapor into the low side of the operating system until the superheat is within 5°F of the chart. If the compressor superheat of the system is more than 5°F below what the chart reads, recover refrigerant until the superheat is within 5°F of the chart. *Note: Always let the system run at least 15 minutes after adding or recovering refrigerant from the system before recalculating compressor superheat. If using a newer alternate refrigerant that has a temperature glide, liquid refrigerant must be throttled into the low side of the system while the system is running to avoid fractionation.*

Example 5-2

Determine if the charge is correct for an R-22 capillary tube or fixed orifice system with the following information:

- Indoor dry bulb temperature = 80°F
- Outdoor dry bulb temperature = 90°F
- Suction pressure at compressor = 60 psig (34°F)
- Compressor inlet temperature on suction line = 54°F

Solution 5-2

From the information given, it is possible to determine that the compressor superheat is 20°F (54° - 34°F). Looking at Figure 5-18, the correct superheat at the intersection of 90°F outdoor temperature with 80°F indoor temperature is about 17°F. This system is running a

The diagram below illustrates the proper Manifold Gauge and Temperature Analyzer connections for charging a split cooling system.

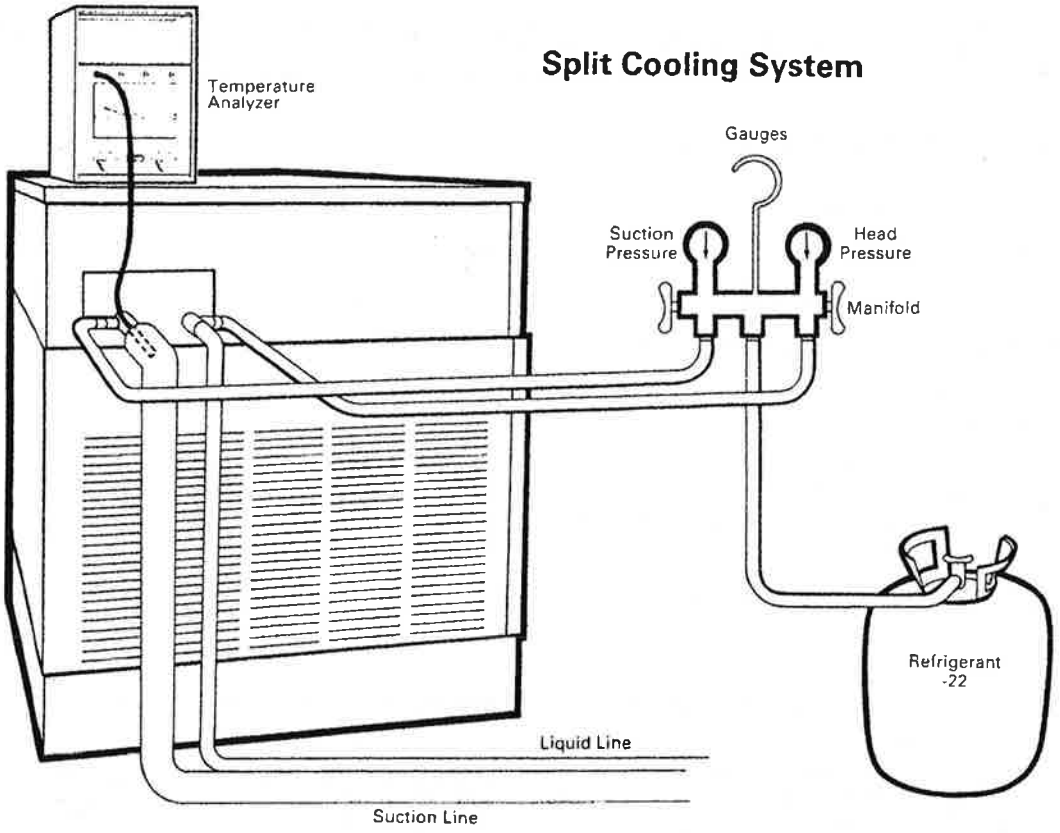


Figure 5-16. Measuring superheat at the compressor (Courtesy, Trane Company)

compressor superheat of 20°F. This is within 5°F of the superheat chart, meaning the system is fully charged and no refrigerant needs to be added to the system.

Superheat Curve Theory

The theory behind superheat tables and curves is simple. Notice in Figure 5-18 that when moving to the right on the bottom axis, the outdoor temperature rises. For a constant indoor db or wb temperature (lines that slant downward from left to right), as outdoor ambient increases the operating compressor, superheat decreases. The reason for this is that there is more head pressure pushing the subcooled liquid out of the condenser bottom through the liquid line and the capillary tube. This forces more refrigerant into the evaporator and gives less superheat, which is why some systems flood and slug liquid at hot outdoor ambients when

SUCTION PRESSURE PSIG	SATURATED SUCTION TEMP. °F.
54.9	30
57.5	32
60.1	34
62.8	36
65.6	38
68.5	40
71.5	42
74.5	44
77.6	46
80.8	48
84.0	50
87.4	52
90.8	54
94.3	56
97.9	58
101.6	60

Figure 5-17. Pressure/temperature chart for R-22

Charging Capillary Tube – Cooling Mode Only

Current Superheat Method

1. Measure indoor dry bulb temperature. (Return air at air handler).
2. Measure outdoor dry bulb temperature. (Measure at outdoor unit).
3. Measure suction pressure at pressure tap on outdoor unit.
4. Measure suction temperature near pressure tap on outdoor unit.
5. Determine degrees superheat from temperature pressure chart or low side manifold gage.
6. *Find the intersection where the outdoor temperature and indoor temperature meet and read degrees superheat. If unit superheat is more than 5° above chart value, add R-22 until within 5°. If unit superheat is more than 5° below chart value, remove R-22 until within 5°.
7. If superheat is below the 5° limit line, DO NOT ADD R-22.

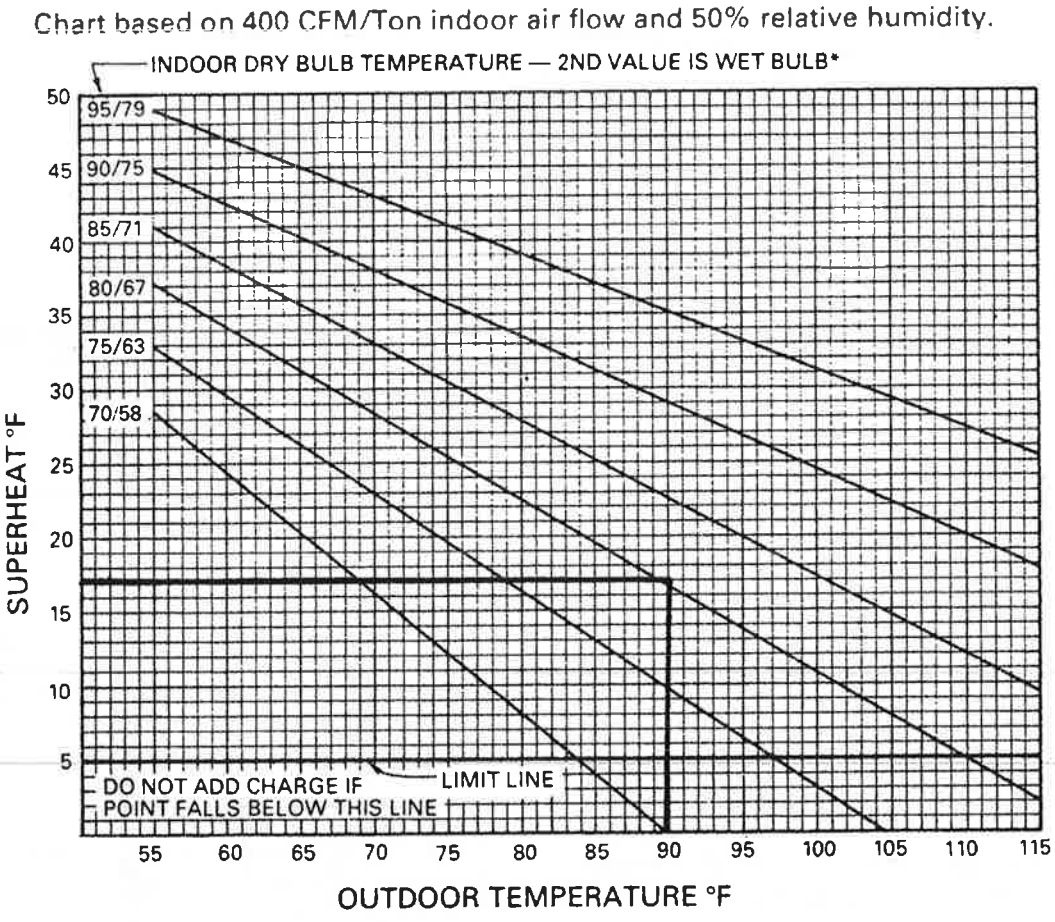


Figure 5-18. Superheat charging curve (Courtesy, Trane Company)

they are overcharged. The superheat curve prevents this from occurring if followed properly.

Referring to Figure 5-18, consider a constant indoor db temperature across the evaporator coil of 75°F. If outdoor db temperature is increased from 70° to 105°F, the operating compressor superheat will fall from 23° to 0°F. This is caused by hotter outdoor ambients, resulting in higher head pressures pushing more liquid through the capillary tube and into the evaporator. It is normal for the system to run 23°F of compressor superheat when the outdoor ambient is 70°F. Adding additional refrigerant to this system would not be wise, because if the outdoor ambient climbs to 95°F later in the day, the system compressor will slug or flood. *Note: If the relative humidity is above 70% or below 20%, use wb temperatures instead of db temperatures across the evaporator coil for Figure 5-18 to compensate for the varying latent (moisture) loads.*

If the outdoor temperature stays constant and the indoor db or wb temperature increases, the operating superheat will increase. This loading of the indoor coil with sensible heat, latent heat, or both, will cause a more rapid vaporization of refrigerant in the evaporator, causing high compressor superheats. This is a normal occurrence. Many technicians will add refrigerant in this case and overcharge the system. It is completely normal for a capillary or fixed orifice metering device system to run high superheat at high evaporator loads.

Again referring to Figure 5-18, consider if the outdoor ambient stays constant at 95°F. In this case, as the indoor db temperature across the evaporator coil rises from 75° to 95°F, the operating compressor superheat will rise from 6° to 33°F. At a 95°F indoor air db temperature and a 95°F outdoor air db temperature, the superheat should normally be 33°F according to the chart. This seems like an inefficient system with an inactive evaporator, which is the greatest disadvantage of a fixed orifice metering device. However, this is the only way a fixed orifice system can prevent slugging and flooding of refrigerant with varying indoor and outdoor loads that air conditioning systems experience. The main advantage of fixed orifice metering devices is their inexpensive cost. If this system had a TXV as a metering device, it would control a constant superheat and keep the evaporator active at high or low heat loads.

Technicians are often hesitant to measure and use wb temperatures when working on air conditioning systems. However, it is very important to get these measurements when in the field. Wet bulb temperature gives an indication of both the latent (moisture) and sensible heat loads on the coil. A simple thermocouple with some moist cotton wrapped around it placed in a hole drilled in the air duct will suffice. There are more sophisticated devices on the market for measuring wb temperatures. A psychrometer is probably one of the most popular devices used. It consists of a sock wrapped around an ordinary db thermometer. Psychrometers that are slung around a swivel connection are referred to as sling psychrometers, Figure 5-19. The technician simply wets the sock with distilled water and places it in the air stream until a temperature stabilizes. With both a db and wb temperature, the technician may obtain the relative humidity (rh) of the air by referring to charts or graphs, such as that found in Table 5-2.

When charging capillary tube and fixed orifice air conditioning systems, consult with the manufacturer of the air conditioning system and use their exact method of charging. Some manufacturers use different curves and tables for different models of their equipment, Figure 5-20. Other manufacturers have eliminated the need for a wb temperature because of custom-made charging curves that represent their laboratory tests on the equipment. Some manufacturers use a slide-rule superheating charging calculator, Figure 5-21. In general, the charging table in Figure 5-10 can be used with all capillary tube and fixed orifice air conditioning systems if wb and db temperatures are used, and the 400 cfm/ton (±10%) of evaporator air is established.

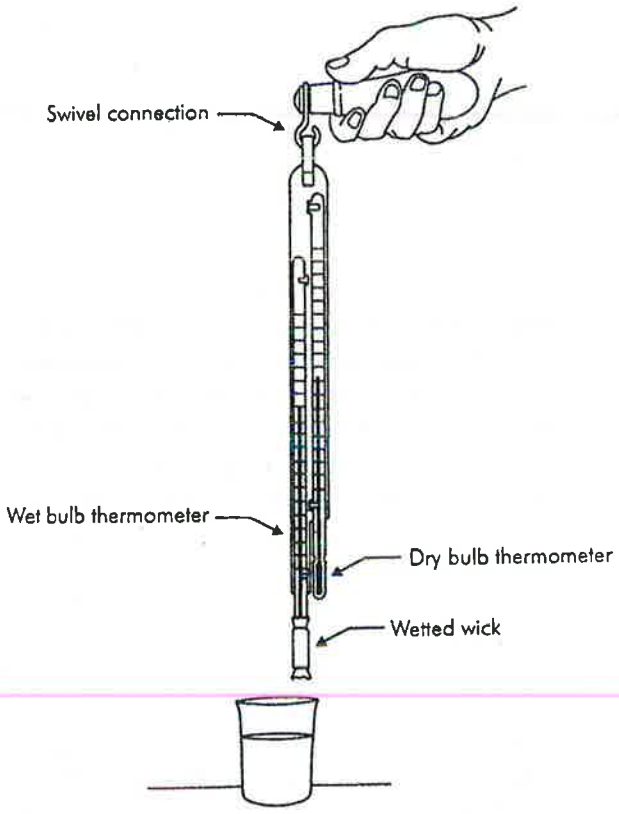


Figure 5-19. Sling psychrometer

Dry Bulb Temp. °F	Wet Bulb Depression 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
32	90	79	69	60	50	41	31	22	13	4																				
36	91	82	73	65	56	48	39	31	23	14	6																			
40	92	84	76	68	61	53	46	38	31	23	16	9	2																	
44	93	85	78	71	64	57	51	44	37	31	24	18	12	5																
48	93	87	80	73	67	60	54	48	42	36	31	25	19	14	8	3														
52	94	88	81	75	69	63	58	52	46	41	36	30	25	20	15	10	6	0												
56	94	88	82	77	71	66	61	55	50	45	40	35	31	26	21	17	12	8	4											
60	94	89	84	78	73	68	63	58	53	49	44	40	35	31	27	22	18	14	6	2										
64	95	90	85	79	75	70	66	61	56	52	48	43	39	35	31	27	23	20	16	12	9									
68	95	90	85	81	76	72	67	63	59	55	51	47	43	39	35	31	28	24	21	17	14									
72	95	91	86	82	78	73	69	65	61	57	53	49	46	42	39	35	32	28	25	22	19									
76	96	91	87	83	78	74	70	67	63	59	55	52	48	45	42	38	35	32	29	26	23									
80	96	91	87	83	79	76	72	68	64	61	57	54	51	47	44	41	38	35	32	29	27	24	21	18	16	13	11	8	6	4
84	96	92	88	84	80	77	73	70	66	63	59	56	53	50	47	44	41	38	35	32	30	27	25	22	20	17	15	12	10	8
88	96	92	88	85	81	78	74	71	67	64	61	58	55	52	49	46	43	41	38	35	33	30	28	25	23	21	18	16	14	12
92	96	92	89	85	82	78	75	72	69	65	62	59	57	54	51	48	45	43	40	38	35	33	30	28	26	24	22	19	17	15
96	96	93	89	86	82	79	76	73	70	67	64	61	58	55	53	50	47	45	42	40	37	35	33	31	29	26	24	22	20	18
100	96	93	90	86	83	80	77	74	71	68	65	62	59	57	54	52	49	47	44	42	40	37	35	33	31	29	27	25	23	21
104	97	96	90	87	84	80	77	74	72	69	66	63	61	58	56	53	51	48	46	44	41	39	37	35	33	31	29	27	25	24
108	97	96	90	87	84	81	78	75	72	70	67	64	62	59	57	54	52	50	47	45	43	41	39	37	35	33	31	29	28	26

Note: Wet bulb depression is the difference between the dry bulb and wet bulb temperatures.

Table 5-2. Psychrometric table: Percent relative humidity from dry bulb temperature and wet bulb depression (Fahrenheit temperature scale)

SUPERHEAT METHOD — CAPILLARY TUBE SYSTEM

This method of charging is effective **only** when the indoor conditions are within 2F of desired indoor comfort and the suction line pressure and temperature are stabilized.

CAUTION: USE LIQUID LINE HEAD PRESSURE METHOD IF INDOOR CONDITIONS ARE ABOVE NORMAL TEMPERATURE AND HUMIDITY.

1. Read and record outdoor ambient air dry bulb (DB) temperature entering condenser unit.
2. Read and record suction line pressure and temperature at the service valve or service port at compressor.
3. Enter proper table below as **determined by the model number** at the intersection of suction pressure and outdoor ambient temperature. Suction line temperature should coincide with the table reading.
4. If suction line temperature is not the same, adjust refrigerant charge.
   Adding R-22 will raise suction pressure and lower suction line temperature.
   Removing R-22 will lower suction pressure and raise suction line temperature.
   CAUTION — If adding R-22 raises both suction pressure and temperature, the unit is overcharged.
5. Fifteen to thirty minutes after adjusting the charge, Steps 1 through 3 are to be repeated after suction line pressure and temperature have stabilized.
6. Should the intersection of the suction pressure and outdoor ambient temperature fall in the open areas of the tables, the following are likely causes:

Left of Numbers	Right of Numbers
A. Low indoor airflow	A. Gross overcharge
B. Restricted refrigerant line	B. Defective compressor
C. Low charge	C. Indoor conditions above desired comfort

MODEL SERIES (-)ACC-, (-)ACW-
STANDARD EFFICIENCY

OUTDOOR AMBIENT °F	SUCTION PRESSURE AT COMPRESSOR — PSIG																
	52	54	56	58	60	62	64	66	68	70	72	74	76	78	80	82	84
	SUCTION LINE TEMPERATURE AT COMPRESSOR — °F																
ABOVE 100								43	44	45	46	48	49	50	52	53	
100							43	44	46	47	48	50	51	52	54	55	
95						45	47	48	50	51	52	54	55	56	57	58	
90						50	52	53	55	56	57	59	60	61	63		
85					53	54	56	57	59	60	61	63	64				
80				56	58	59	61	62	64	65	66	68					
75			59	60	62	63	65	66	68	69	70						
70		61	63	64	66	67	69	70	72	73							

MODEL SERIES (-)AFC-, (-)AFW-
HIGH EFFICIENCY

OUTDOOR AMBIENT °F	SUCTION PRESSURE AT COMPRESSOR — PSIG																
	52	54	56	58	60	62	64	66	68	70	72	74	76	78	80	82	84
	SUCTION LINE TEMPERATURE AT COMPRESSOR — °F																
ABOVE 100							42	43	44	45	46	48	49	50	52	53	
100						45	47	48	50	51	52	54	55	56	58	59	
95						50	52	53	55	56	57	59	60	61	63		
90					55	57	58	60	61	62	64	65	66				
85				58	60	61	63	64	66	67	68	70					
80				63	65	66	67	68	70	71	72						
75		66	68	69	71	72	74	75									
70		71	73	74	76												

MODEL SERIES (-)AGD-, (-)AHD-
SUPER HIGH EFFICIENCY

OUTDOOR AMBIENT °F	SUCTION PRESSURE AT COMPRESSOR — PSIG																
	52	54	56	58	60	62	64	66	68	70	72	74	76	78	80	82	84
	SUCTION LINE TEMPERATURE AT COMPRESSOR — °F																
ABOVE 100							42	43	44	45	46	48	49	50	52	53	54
100						43	45	46	48	49	50	52	53	54	56	58	
95						50	52	53	55	56	57	59	60	61	63	64	
90					56	57	59	60	62	63	64	66	67	68			
85					63	64	66	67	69	70	72	74					
80				68	70	71	73	74	76	77	78						
75			70	72	74	75	77	79									
70		72	74	76	77	78											

(REF. 92-20477-84-01)

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Figure 5-20. Superheat method for capillary tube system (Courtesy, Rheem Air Conditioning Division)

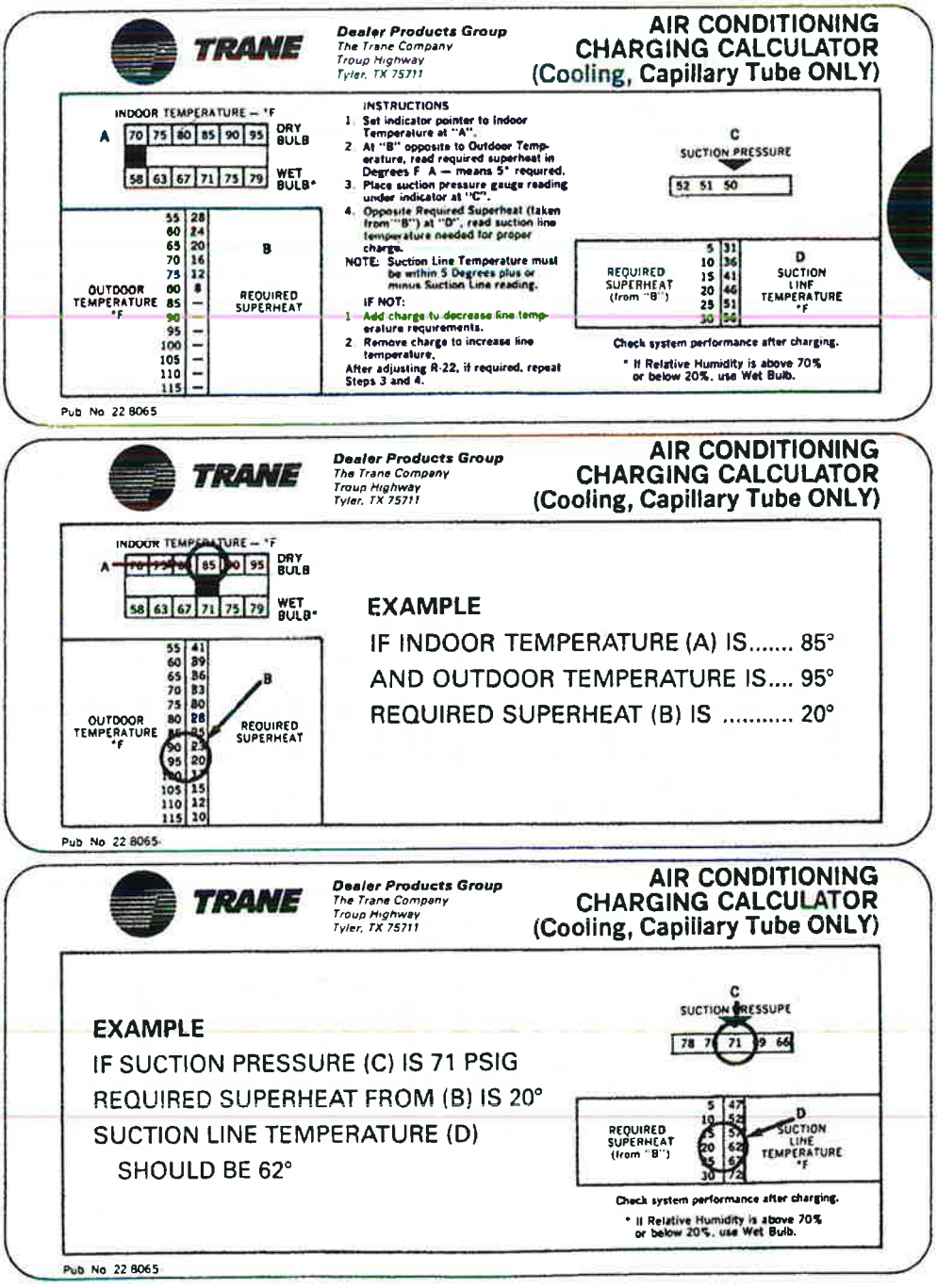


Figure 5-21. Slide-rule calculator (Courtesy, Trane Company)