Lesson
24
Expansion Devices

The specific objectives of this lecture are to:

1. Discuss the basic functions of expansion devices used in refrigeration systems and their classification (*Section 24.1*)
2. Discuss the operating principle, concept of balance point, the effect of load variation, selection of capillary tubes using analytical and graphical methods and the advantages and disadvantages of capillary tubes (*Section 24.2*)
3. Explain the working principle of an automatic expansion valve, its performance under varying loads and its applications (*Section 24.3*)
4. Present a simple analysis for fluid through orifices (*Section 24.4*)
5. Explain the working principle of a thermostatic expansion valve, its performance under varying loads, variations available such as cross-charging, external equalizer and limit charging, advantages and disadvantages of TEVs (*Section 24.5*)
6. Explain the working principle of low-side and high-side float valves (*Section 24.6*)
7. Explain the working principle of an electronic expansion valve (*Section 24.7*)
8. Discuss briefly some of the practical problems with expansion devices (*Section 24.8*)

At the end of the lecture, the student should be able to:

1. Explain the basic functions of expansion devices in refrigeration systems
2. Explain the working principle and salient features of capillary tube, automatic expansion valve, thermostatic expansion valve, float type expansion valve and electronic expansion valve
3. Estimate the required length of capillary tubes using analytical and graphical methods
4. Describe advantages, disadvantages and applications of different types of expansion valves, and
5. Discuss some of the practical problems encountered in the operation of various types of expansion devices in refrigeration systems

24.1. Introduction

An expansion device is another basic component of a refrigeration system. The basic functions of an expansion device used in refrigeration systems are to:

1. Reduce pressure from condenser pressure to evaporator pressure, and
2. Regulate the refrigerant flow from the high-pressure liquid line into the evaporator at a rate equal to the evaporation rate in the evaporator

Under ideal conditions, the mass flow rate of refrigerant in the system should be proportional to the cooling load. Sometimes, the product to be cooled is such that a constant evaporator temperature has to be maintained. In other cases, it is desirable that liquid refrigerant should not enter the compressor. In such a case, the mass flow rate has to be controlled in such a manner that only superheated vapour leaves the evaporator. Again, an ideal refrigeration system should have the facility to control it in such a way that the energy requirement is minimum and the required criterion of temperature and cooling load are satisfied. Some additional controls to control the capacity of compressor and the space temperature may be required in addition, so as to minimize the energy consumption.

The expansion devices used in refrigeration systems can be divided into *fixed opening type* or *variable opening type*. As the name implies, in fixed opening type the flow area remains fixed, while in variable opening type the flow area changes with changing mass flow rates. There are basically seven types of refrigerant expansion devices. These are:

1. Hand (manual) expansion valves
2. Capillary Tubes
3. Orifice
4. Constant pressure or Automatic Expansion Valve (AEV)
5. Thermostatic Expansion Valve (TEV)
6. Float type Expansion Valve
    a) High Side Float Valve
    b) Low Side Float Valve
7. Electronic Expansion Valve

Of the above seven types, Capillary tube and orifice belong to the fixed opening type, while the rest belong to the variable opening type. Of the above seven types, the hand operated expansion valve is not used when an automatic control is required. The orifice type expansion is used only in some special applications. Hence these two are not discussed here.

24.2 Capillary Tube

A capillary tube is a long, narrow tube of constant diameter. The word “capillary” is a misnomer since surface tension is not important in refrigeration application of capillary tubes. Typical tube diameters of refrigerant capillary tubes range from 0.5 mm to 3 mm and the length ranges from 1.0 m to 6 m.

The pressure reduction in a capillary tube occurs due to the following two factors:

1. The refrigerant has to overcome the frictional resistance offered by tube walls. This leads to some pressure drop, and

2. The liquid refrigerant flashes (evaporates) into mixture of liquid and vapour as its pressure reduces. The density of vapour is less than that of the liquid. Hence, the average density of refrigerant decreases as it flows in the tube. The mass flow rate and tube diameter (hence area) being constant, the velocity of refrigerant increases since $\dot{m} = \rho VA$. The increase in velocity or acceleration of the refrigerant also requires pressure drop.

Several combinations of length and bore are available for the same mass flow rate and pressure drop. However, once a capillary tube of some diameter and length has been installed in a refrigeration system, the mass flow rate through it will vary in such a manner that the total pressure drop through it matches with the pressure difference between condenser and the evaporator. Its mass flow rate is totally dependent upon the pressure difference across it; it cannot adjust itself to variation of load effectively.

24.2.1. Balance Point of Compressor and Capillary Tube

The compressor and the capillary tube, under steady state must arrive at some suction and discharge pressures, which allows the same mass flow rate through the compressor and the capillary tube. This state is called the balance point. Condenser and evaporator pressures are saturation pressures at corresponding condenser and evaporator temperatures. Figure 24.1 shows the variation of mass flow rate with evaporator pressure through the compressor and the capillary tube for three values of condenser temperatures namely, 30, 40 and 50^{o}C.

The mass flow rate through the compressor decreases if the pressure ratio increases since the volumetric efficiency of the compressor decreases with the increase of pressure ratio. The pressure ratio increases when either the evaporator pressure decreases or the condenser pressure increases. Hence, the mass flow rate through the compressor decreases with increase in condenser pressure and/or with decrease in evaporator pressure.

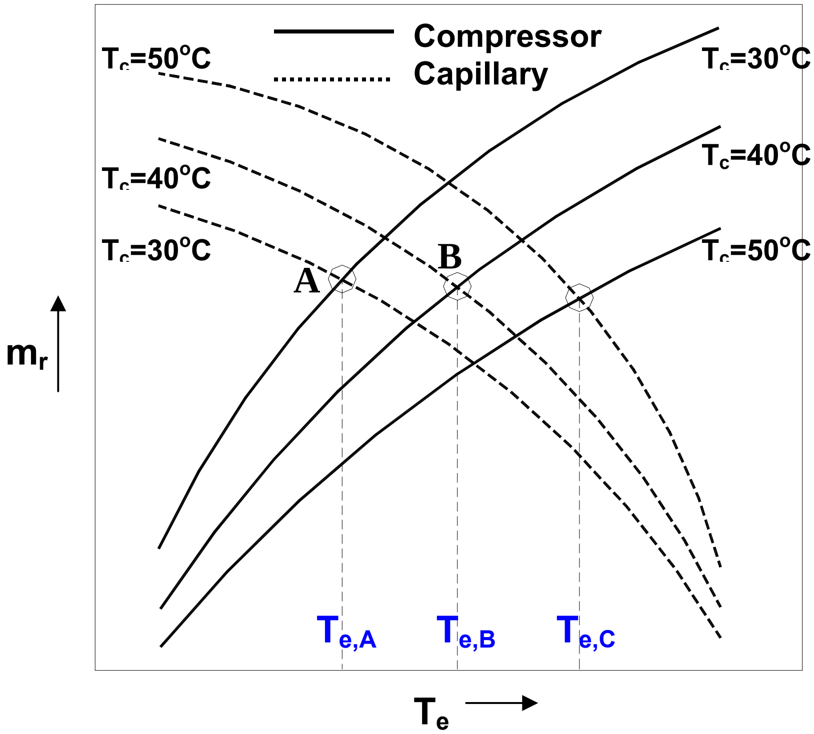

***Fig.24.1**: Variation of refrigerant mass flow rate through compressor and capillary tube with evaporator and condenser temperatures (A,B & C are the balance points)*

The pressure difference across the capillary tube is the driving force for the refrigerant to flow through it, hence mass flow rate through the capillary tube increases with increase in pressure difference across it. Thus the mass flow rate through the capillary tube increases as the condenser pressure increases and/or the evaporator pressure decreases. The variation of mass flow rate through capillary tube is shown for three condenser temperatures, namely, 30, 40 and 50^{o}C in Figure 24.1. This is the opposite of the effect of pressures on the compressor mass flow rate. Hence, for a given value of condenser pressure, there is a definite value of evaporator pressure at which the mass flow rates through the compressor and the evaporator are the same. This pressure is the balance point that the system will acquire in steady state. Hence, for a given condenser temperature, there is a definite value of evaporator temperature at which the balance point will occur. Figure 28.1 shows a set of three balance points *A*, *B* and *C* for the three condenser temperatures. These balance points occur at evaporator temperatures of $T_{e,A}$, $T_{e,B}$ and $T_{e,C}$. It is observed that the evaporator temperature at balance point increases with increase of condenser temperature.

24.2.2. Effect Of load variation

The situation described above is in steady state. However, in practice the refrigeration load may vary due to several reasons, such as the variation of ambient temperatures etc. It is possible for the load to increase or decrease. This variation of load affects the operation of compressor and capillary tube and affects the balance point between them.

Increase in refrigeration Load:

If the refrigeration load increases, there is a tendency for the evaporator temperature to increase due to higher rate of evaporation. This situation is shown in Figure 24.2 for a condenser temperature of 40^{o}C. The balance point for design load is shown by point B. As the load increases, the evaporator temperature rises to C. At point C the mass flow rate through compressor is more than the mass flow rate through the capillary tube. In such a situation, the compressor will draw more refrigerant through the evaporator than the capillary tube can supply to it. This will lead to *starving* of the evaporator. However, emptying of evaporator cannot continue indefinitely. The system will take some corrective action since changes are occurring in the condenser also. Since the capillary tube feeds less refrigerant to the evaporator, the refrigerant accumulates in the condenser. The accumulation of refrigerant in the condenser reduces the effective area of the condenser that is available for heat transfer. The condenser heat transfer rate is given by, $Q_c = U_c A_c (T_c\text{-}T_\infty)$. If heat transfer coefficient U_c and T_∞ are constant, then for same heat transfer rate a decrease in area A_c will lead to a higher condenser temperature $T_{c.}$ It is observed from Figure 24.1 that an increase in condenser temperature leads to a decrease in compressor mass flow rate and an increase in capillary mass flow rate. Hence, the system will find a new balance point at higher condenser temperature.

The second possibility is that at lower evaporator mass flow rate, the Reynolds number decreases and as a result, the heat transfer coefficient of evaporator decreases. Or in a flooded evaporator, the reduction in mass flow rate reduces the wetted surface area and the heat transfer coefficient. Therefore, larger temperature difference is required in the evaporator for the same amount of heat transfer. This decreases the evaporator temperature and corresponding pressure to the previous values.

Decrease In refrigeration Load

If the refrigeration load decreases, there is a tendency for the evaporator temperature to decrease, say to state *A as* shown in Figure 28.2. In this condition the capillary tube feeds more refrigerant to the evaporator than the compressor can remove. This leads to accumulation of liquid refrigerant in the evaporator causing *flooding* of the evaporator. This may lead to dangerous consequences if the liquid refrigerant overflows to the compressor causing *slugging* of the compressor. This has to be avoided at all costs; hence the capillary tube based refrigeration systems use *critical charge* as a safety measure. Critical charge is a definite amount of refrigerant that is put into the refrigeration system so that in the eventuality of all of it accumulating in the evaporator, it will just fill the evaporator up to its brim and never overflow from the evaporator to compressor. The flooding of the evaporator is also a transient phenomenon, it cannot continue indefinitely. The system has to take some corrective action. Since the capillary tube feeds more refrigerant from the condenser, the liquid seal at the condenser-exit breaks and some vapour enters the capillary tube. The vapour has a very small density compared to the liquid; as a result the mass flow rate through the capillary tube decreases drastically. This situation is shown in Figure 28.2. This is not desirable since the refrigeration effect decreases and the COP also decreases. Hence, attempts are made in all the refrigeration plants to subcool the refrigerant before entry to the expansion device. A vapour to liquid subcooling heat exchanger is usually employed, wherein the low temperature refrigerant vapour leaving the evaporator subcools the liquid leaving the condenser.

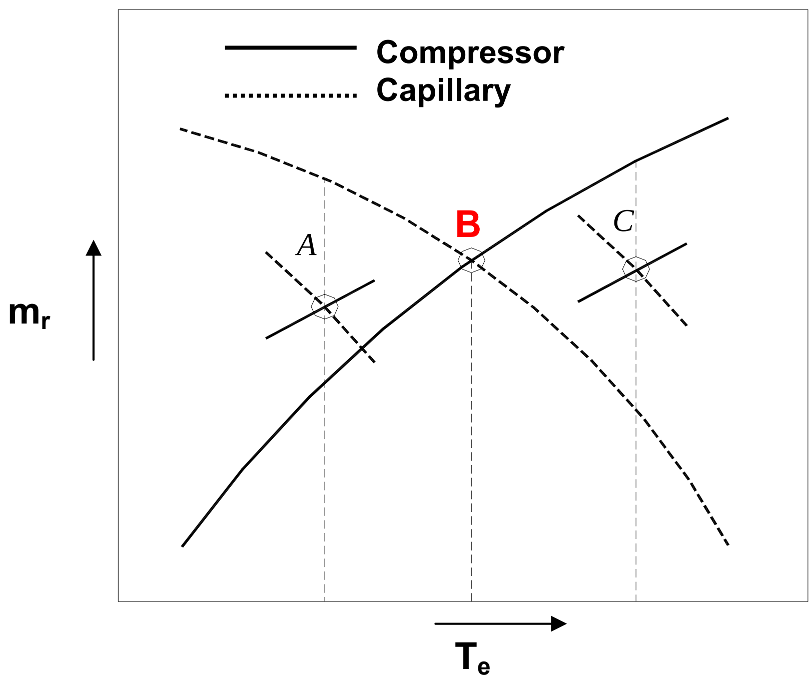

Fig.24.2: *Effect of load variation on capillary tube based refrigeration systems. B: Design point; A: At low load; C: At high load*

24.2.3. Selection of Capillary Tube

For any new system, the diameter and the length of capillary tube have to be selected by the designer such that the compressor and the capillary tube achieve the balanced point at the desired evaporator temperature. There are analytical and graphical methods to select the capillary tube. The fine-tuning of the length is finally done by *cut-and-try* method. A tube longer than the design (calculated) value is installed with the expected result that evaporating temperature will be lower than expected. The tube is shortened until the desired balance point is achieved. This is done for mass production. If a single system is to be designed then tube of slightly shorter length than the design length is chosen. The tube will usually result in higher temperature than the design value. The tube is pinched at a few spots to obtain the required pressure and temperature.

Analytical Method

The analysis of flow through a capillary tube is one of the interesting problems that illustrate how a simple one-dimensional analysis yields good results. In a capillary tube the flow is actually compressible, three-dimensional and two-phase flow with heat transfer and thermodynamic meta-stable state at the inlet of the tube. However, in the simplified analysis, the flow is assumed to be steady, one-dimensional and in single phase or a homogenous mixture. One-dimensional flow means that the velocity does not change in the radial direction of the tube. Homogeneous means annular flow or plug flow model etc. or not considered for the two-phase flow. Figure 28.3 shows a small section of a vertical capillary tube with momentum and pressure at two ends of an elemental control volume.

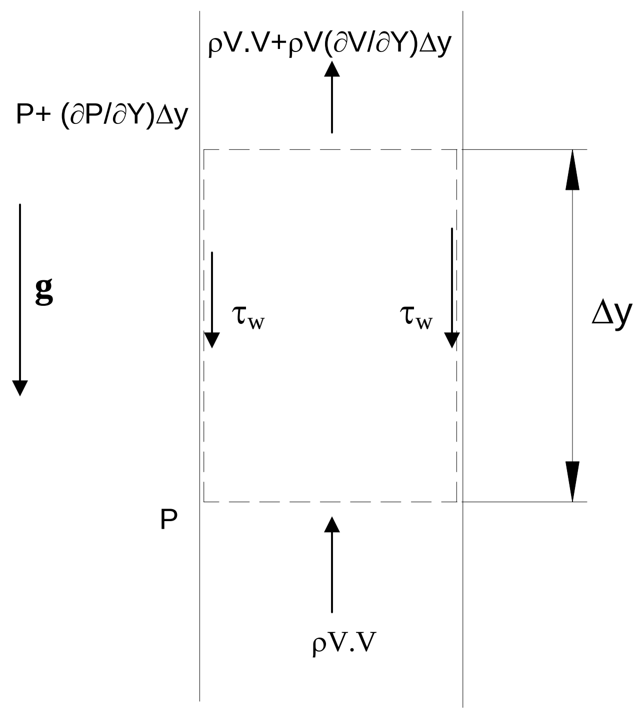

***Fig.24.3**: A small section of a capillary tube considered for analysis*

Applying mass and momentum conservation for a control volume shown in Fig. 24.3, we get:

<u>Mass Conservation</u>:

$$\rho \mathbf{VA} + \frac{\partial(\rho \mathbf{V})}{\partial \mathbf{y}} \Delta \mathbf{yA} - \rho \mathbf{VA} = \mathbf{0} \qquad (24.1)$$

$$\frac{\partial(\rho \mathbf{V})}{\partial \mathbf{y}} = \mathbf{0.0} \quad \therefore \quad \rho V = \text{constant}$$

<u>Momentum Conservation:</u>

The momentum theorem is applied to the control volume. According to this, $[\text{Momentum}]_{out}$ – $[\text{Momentum}]_{in}$ = Total forces on control volume

$$\pi R^2[\rho VV + \rho V\frac{\partial V}{\partial y}\Delta y] - \pi R^2 [\rho VV] = - \pi R^2\frac{\partial p}{\partial y}\Delta y - \rho_{avg} g \pi R^2 \Delta y - 2\pi R \Delta y \tau_w \quad (24.2)$$

At the face $y + \Delta y$, Taylor series expansion has been used for pressure and momentum and only the first order terms have been retained. The second order terms with second derivatives and higher order terms have been neglected. If the above equation is divided by $\pi R^2 \Delta y$ and limit $\Delta y \rightarrow 0$ is taken; then all the higher order terms will tend to zero if these were included since these will have Δy or its higher power of Δy multiplying them. Also, ρ_{avg} will tend to ρ since the control volume will shrink to the bottom face of the control volume where ρ is defined. Further, neglecting the effect of gravity, which is very small, we obtain:

$$\rho V\frac{\partial \mathbf{V}}{\partial \mathbf{y}} = - \frac{\partial \mathbf{p}}{\partial \mathbf{y}} - 2\ \frac{\tau_w}{\mathbf{R}} \quad (24.3)$$

The wall shear stress may be written in terms of friction factor. In fluid flow through pipes the pressure decreases due to shear stress. This will be referred to as frictional pressure drop and a subscript '*f*' will be used with it and it will be written in terms of friction factor. The Darcy's friction factor is for fully developed flow in a pipe. In fully developed flow the velocity does not change in the flow direction. In case of a capillary tube it increases along the length. Still it is good approximation to approximate the shear stress term by friction factor. For fully developed flow the left hand side of Equation (28.3) is zero, hence the frictional pressure drop Δp_f may be obtained from the following equation:

$$\tau_w = R\ \Delta p_f /(\ 2\Delta y\ ) \quad (24.4)$$

The friction factor is defined as

$$\Delta p_f = \rho\, f\ \frac{\Delta y}{D}\frac{V^2}{2} \quad (24.5)$$

Substituting Eqn.(28.5) in Eqn.(28.4) we get

$$\tau_w = \rho\, f\, V^2 / 8 \quad (24.6)$$

Substituting for τ_w in Eqn.(28.3) we have:

$$\rho \mathbf{V} \frac{\partial \mathbf{V}}{\partial \mathbf{y}} = - \frac{\partial \mathbf{p}}{\partial \mathbf{y}} - \frac{\rho \mathbf{f} \mathbf{V}^2}{\mathbf{2D}} \qquad (24.7)$$

Mass conservation Eqn.(28.1) indicates that the product ρV is constant in the tube. In fact it is called mass velocity and is denoted by G,

$$\boldsymbol{G = \rho V}$$

We have mass flow rate $\dot{m} = [\pi D^2/4]\, \rho V$

$$\therefore \rho \mathbf{V} = \dot{m}/\boldsymbol{A} = \boldsymbol{G} = \mathbf{constant} \qquad (24.8)$$

Hence Eqn.(28.7) is rewritten as follows

$$\boldsymbol{G} \frac{\partial \mathbf{V}}{\partial \mathbf{y}} = - \frac{\partial \mathbf{p}}{\partial \mathbf{y}} - \frac{\mathbf{f\,V\,G}}{\mathbf{2D}} \qquad (24.9)$$

In this equation the term on the left hand side is the acceleration of fluid. The first term on the right hand side is the pressure drop required to accelerate the fluid and to overcome the frictional resistance. The second term on the right hand side is the frictional force acting on the tube wall. The friction factor depends upon the flow Reynolds number and the wall roughness for the fully developed flow. For the developing flow it is function of distance along the tube also in addition to Reynolds number. The flow accelerates along the tube due to vapour formation, as a result, the Reynolds number increases along the tube. The velocity and Reynolds number vary in a complex manner along the tube and these are coupled together. Hence, an exact solution of Eqn.(24.9) is not possible. To a good approximation the integral of product $f\,V$, that is, $\int \boldsymbol{f\,V}\,\mathbf{d}\boldsymbol{y}$ can be calculated by assuming average value of the product $f\,V$ over a small length ΔL of the capillary tube.

Accordingly, integrating Equation (24.9) over a small length ΔL of the capillary tube we obtain

$$\boldsymbol{G\,\Delta V = -\Delta p - [fV]_{\mathbf{mean}}\,\mathbf{G}\Delta L/2D} \qquad (24.10)$$

$$\boldsymbol{\Delta p = G\,\Delta V + [G\,/\,2D]\,[f\,V]_{\mathbf{mean}}\,\Delta L} \qquad (24.11)$$

Where, $\Delta V = V_{i+1} - V_i$ and $\Delta p = p_{i+1} - p_i$

Δp is negative since $p_i > p_{i+1}$.

Equation (24.11) may be expressed as follows

$$\Delta p = \Delta p_{accln} + \Delta p_f$$

This means that total pressure drop over a length ΔL is the sum of that required for acceleration and that required to overcome frictional resistance.

For laminar flow the effect of wall roughness in negligible and friction factor is given by

$$f = 64/Re \qquad (24.12)$$

For turbulent flow the friction factor increases with increase in roughness ratio. Moody's chart gives the variation of friction factor with Reynolds numbers for various roughness ratios. A number of empirical expressions are also available for friction factor in standard books on Fluid Mechanics. One such expression for the smooth pipe, known as Blasius Correlation is as follows:

$$f = 0.3164\, Re^{-0.25} \approx 0.32\, Re^{-0.25} \quad : \text{for } Re < 10^5 \qquad (24.13)$$

The solution procedure for Eqn.(24.11) as suggested by Hopkins and Copper and Brisken is as follows:

The condenser and evaporator temperatures T_c and T_e, the refrigerant and its mass flow rate are usually specified and the length and bore of capillary tube are required. Eqn.(24.11) is valid for a small length of the tube. Hence, the tube is divided into small lengths ΔL_i such that across each incremental length a temperature drop Δt_i of say 1 or 2 degrees takes place depending upon the accuracy of calculation required. The length of the tube ΔL_i for temperature to drop by say, 1^oC is found from Eqn.(24.11). The temperature base is taken for calculations instead of pressure base since the refrigerant properties are available on basis of temperature.

1. Assume an appropriate diameter D for the tube. At condenser exit and inlet to capillary tube point "0" shown in Figure 24.4, say the state is saturated liquid state hence,

$v_0 = v_f$, $h_0 = h_f$, $\mu_0 = \mu_f$ and

$\dot{m}$ is known from thermodynamic cycle calculation for the given cooling capacity.

$$\therefore \text{Re} = 4\dot{m}/(\pi D\mu),$$

$$G = \dot{m}/A = \rho V = V/v$$

The constants in Eqn.(24.11) G, $G/(2D)$ and $4\dot{m}/\pi D$ required for solution are then calculated.

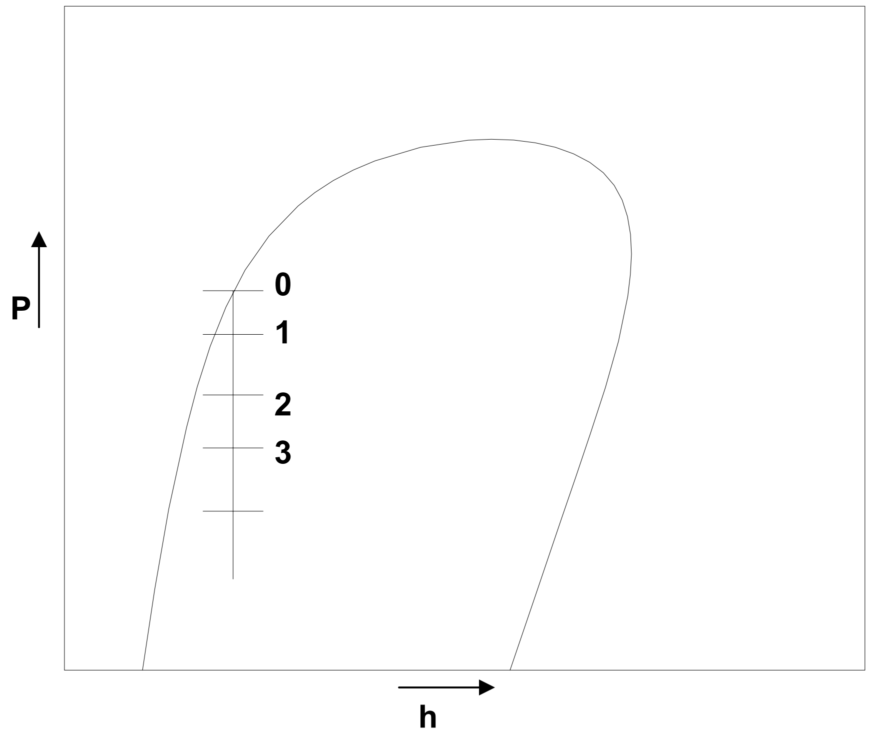

Fig.24.4: *Step-wise calculation procedure for capillary tube length on p-h diagram*

2. At inlet $i = 0$: $Re_0 = 4\dot{m}/(\pi D\mu_0)$, $f_0 = 0.32\ Re^{-0.25}$ and $V_0 = v_0 G$

3. At $i = 1$ in Figure 10.6: $t_1 = t_c - \Delta t_1$, find the saturation pressure p_1 at t_1. The saturation properties v_{1f}, v_{1g}, h_{1f}, h_{1g} and μ_{1f} and μ_{1g} are obtained at t_1. It is assumed that the enthalpy remains constant during expansion as shown in Figure 28.5.

4. If x_1 is the dryness fraction at $i = 1$, then

$$h_0 = h_1 = x_1 h_{1g} + (1 - x_1)\, h_{1f} \tag{24.14}$$

$$\therefore x_1 = [h_0 - h_{1f}] / [h_{1g} - h_{1f}]$$

5. Find $v_1 = x_1 v_{1g} + (1 - x_1)\, v_{1f}$

Assuming that viscosity of mixture can be taken as weighted sum of viscosity of saturated liquid and vapour we get,

$$\mu_1 = x_1 \mu_{1g} + (1 - x_1)\, \mu_{1f}$$

$$Re_1 = 4\dot{m}/(\pi D\mu_1), \quad f_1 = 0.32\ Re^{-0.25} \text{ and } V_1 = v_1 G$$

$$\Delta V = V_1 - V_0$$

$$\Delta p = p_0 - p_1$$

$[fV]_{mean} = [\, f_0\, V_0 + f_1\, V_1\,]/2$

Hence, from Eqn.(24.11) the incremental length of capillary tube for the first step, ΔL_1 is,

$$\Delta L_1 = \frac{-\Delta p - G\,\Delta V}{(G/2D)\ (fV)_{mean}}$$

6. For the next section i = 2 : $t_2 = t_1 - \Delta t_2$, find the saturation pressure p_2 at t_2. The saturation properties v_{2f}, v_{2g}, h_{2f}, h_{2g} and μ_{2f} and μ_{2g} are obtained at temperature t_2.

7. Assuming the enthalpy to remain constant, that is $h_2 = h_1 = h_0$, the quality x_2 is found and steps 4 and 5 are repeated to find the incremental length ΔL_2.

Steps 4 and 5 are repeated for all the intervals up to evaporator temperature and all the incremental lengths are summed up to find the total length of the capillary tube.

It is observed from Eqn.(24.11) that the total pressure drop is the sum of pressure drops due to acceleration that is, $\Delta p_{accln} = G\,\Delta V$ and the pressure drop due to friction, that is, $\Delta p_f = [G/2D]\,[fV]_{mean}\,\Delta L$. It may so happen under some conditions that after a few steps of calculation, the total pressure drop required for a segment may become less than the pressure drop required for acceleration alone, $\Delta p < \Delta p_{accln}$. The increment length ΔL for this segment will turn out to be negative which has no meaning. This condition occurs when the velocity of refrigerant has reached the velocity of sound (sonic velocity). This condition is called *choked flow condition.* The velocity of fluid cannot exceed the velocity of sound in a tube of constant diameter, hence the calculation cannot proceed any further. The flow is said to be choked-flow and the mass flow rate through the tube has reached its maximum value for the selected tube diameter. For a capillary tube of constant diameter, choked flow condition represents the minimum suction pressure that can be achieved. If further pressure drop is required a tube of larger diameter should be chosen in which the velocity of sound occurs at larger length.

Figure 24.5 shows the variation mass flow rate with suction pressure for fixed condenser pressure. The mass flow rate through the capillary tube increases as the evaporator pressure decreases. However at a pressure of p^* the flow is choked. If the choking occurs at some interior point of the tube, the length of the tube from this point to the exit will offer frictional resistance to the flow and the pressure must decrease to overcome this. The pressure however cannot decrease since the flow is choked. Hence, adjustment in the inlet conditions occurs and the mass flow rate is reduced so that the flow will (always) be choked at the exit of the tube with reduced mass flow rate. This is typical of compressible sonic flow where upstream influence occurs; otherwise the downstream pressure decides the mass flow rate.

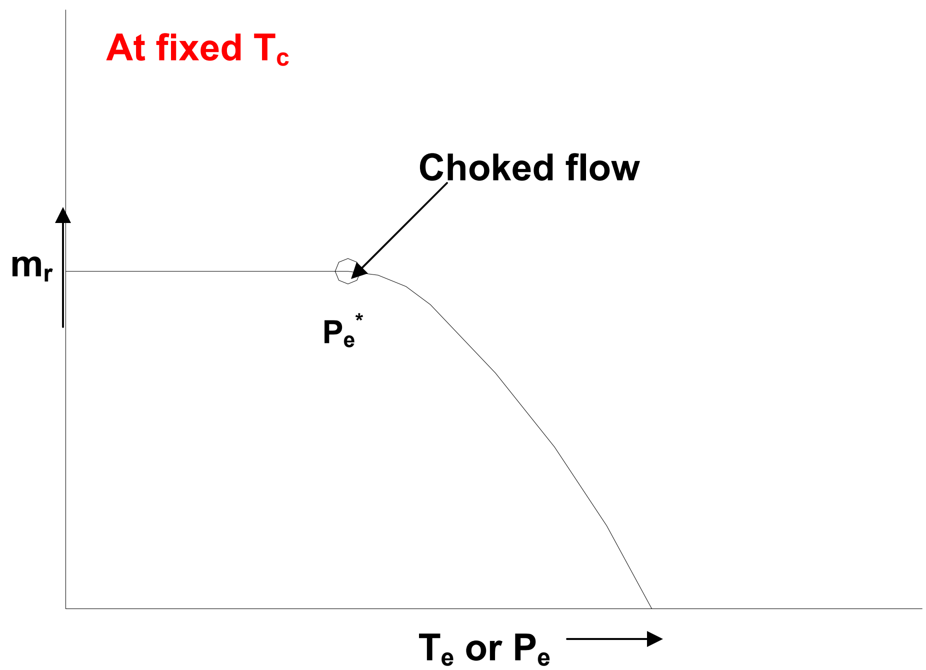

***Fig.24.5**: Variation mass flow rate with suction pressure for fixed condenser pressure*

Shortcomings of the above analysis

It is assumed in the above analysis that the expansion is a constant enthalpy process. This is strictly not true inside a capillary tube since there is a large change in kinetic energy due to change in velocity along the length due to flashing of refrigerant liquid. In fact kinetic energy increases at a very fast rate as the velocity becomes sonic and the flow becomes choked. First law of thermodynamics indicates that in absence of heat transfer, work done and change in potential energy for a system in steady state, the sum of enthalpy and the kinetic energy must remain constant. Hence, if the kinetic energy increases the enthalpy must decrease, as a result the quality of the refrigerant will be lower than calculated by assuming constant enthalpy. The actual state of refrigerant in a constant diameter adiabatic tube is represented by *Fanno line*, which is shown in Fig.24.6 on *h–s* diagram along with the saturation curve. Fanno line is the solution of steady, compressible adiabatic flow with friction through a tube of constant diameter.

It is observed that in the early part of the capillary tube, the constant enthalpy line does not deviate very much from the Fanno line. In the latter part, the deviation from the Fanno line increases. Most of the length of the capillary tube happens to be in the latter portion where quality and velocity changes are very significant; hence constant enthalpy approximation may introduce significant error.

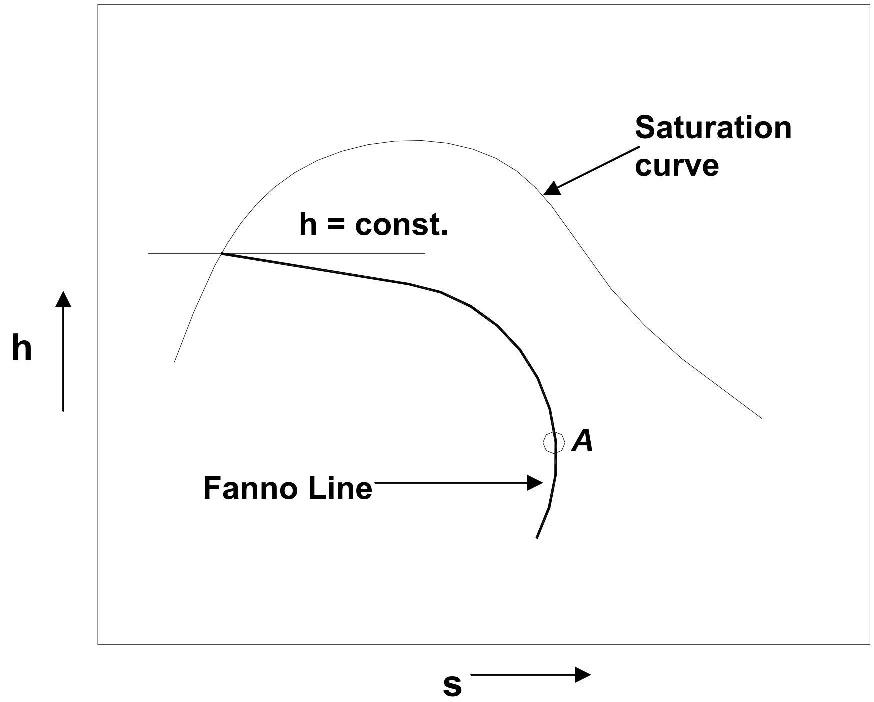

Fig.24.6: *Fanno line for capillary tube on h-s diagram*

Point *A* on the Fanno line is the point where the entropy is maximum. This point corresponds to choked flow condition. Pressure cannot drop below this value since it will require a decrease in entropy under adiabatic condition, which is not possible in a real system. This would mean violation of second law of thermodynamics.

Modified Procedure

It is observed that the Kinetic energy changes significantly in the latter part of the capillary tube. In step 4 of the calculation procedure enthalpy was assumed to be constant. To improve upon it, the quality is calculated by considering energy balance, that is, the sum of enthalpy and kinetic energy is assumed to remain constant. The quality of the mixture is not found from Eqn.(24.14). Instead, sum of enthalpy and kinetic energy is taken as constant. For the first segment we get

$$h_0 + V_o^2/2 = h_1 + V_1^2/2 = h_1 + G^2 v_1^2/2 \qquad (24.15)$$

Substituting for h_1 and v_1 in terms of quality x_1 and properties at saturation, we get

$$x_1 h_{1g} + (1 - x_1)\, h_{1f} + G^2\, [x_1 v_{1g} + (1 - x_1)\, v_{1f}\,]^2\, /2 = h_0 + V_o^2/2 \text{ , or}$$

$$h_{1f} + x_1 h_{1fg} + G^2\, [v_{1f} + x_1 v_{1fg}\,]^2\, /2 = h_0 + V_o^2/2, \text{ or}$$

$$x_1^2\, [v_{1fg}^2\, G^2/2] + x_1[G^2\, v_{1f}\, v_{1fg} + h_{1fg}\,] + (h_{1f} - h_0) + (G^2/2)\, v_{1f}^2 - V_o^2/2 = 0$$

This is a quadratic equation for x_1 that can be solved to find x_1. The positive root of this equation is taken as the value of x_1. The enthalpy is usually given in kJ/kg and velocity in m/s, hence to make the equation dimensionally consistent, the enthalpy is multiplied by 1000, that is,

$$\boldsymbol{x_1^2[v_{1fg}^2\, G^2/2]+x_1[G^2 v_{1f}\, v_{1fg}+1000h_{1fg}]+1000(h_{1f}-h_0)+(G^2/2)v_{1f}^2-V_o^2/2=0} \quad (24.16)$$

The remaining part of the procedure from step 5 to 6 remains the same. For all subsequent steps, the quality is calculated from Eqn.(24.1).

If the entry state of refrigerant to the capillary tube is subcooled, then length required for the pressure to drop from the condenser pressure to the saturated state (which occurs at an intermediate pressure) is calculated and is added to the length required to reduce the pressure from the intermediate saturated pressure to the final evaporator pressure. Calculation of the length for the first part (i.e., in the subcooled liquid region) can be done in a single step as there is no change of phase. For this single phase region, the enthalpy can be assumed to be constant as the change in kinetic energy is negligible. Thus from the known inlet enthalpy corresponding to the subcooled state at condenser pressure, drawing an isenthalpic line, gives the intermediate saturation pressure. For the two-phase region, the above procedure has to be used with the inlet conditions corresponding to the saturated intermediate pressure.

Graphical Procedure

A graphical procedure for capillary tube selection has been presented in ASHRAE Handbook. A representative Figure 24.7 gives the mass flow rate of refrigerant through capillary tube at various inlet pressures, sub-cooling and dryness fraction through a capillary tube of 1.63 mm diameter and 2.03 m length. The companion Figure 24.8 gives the flow correction factor ϕ for diameters and lengths different from that used in Fig.24.8. The mass flow rate for any diameter d_i and length L_c is given by:

$$\mathbf{m_{di,Lc} = m_{1.63\,mm,\,2.03\,m}\cdot\phi} \qquad (24.17)$$

These plots are for choked flow conditions. Corrections for non-choked flow conditions are given in ASHRAE Handbook.

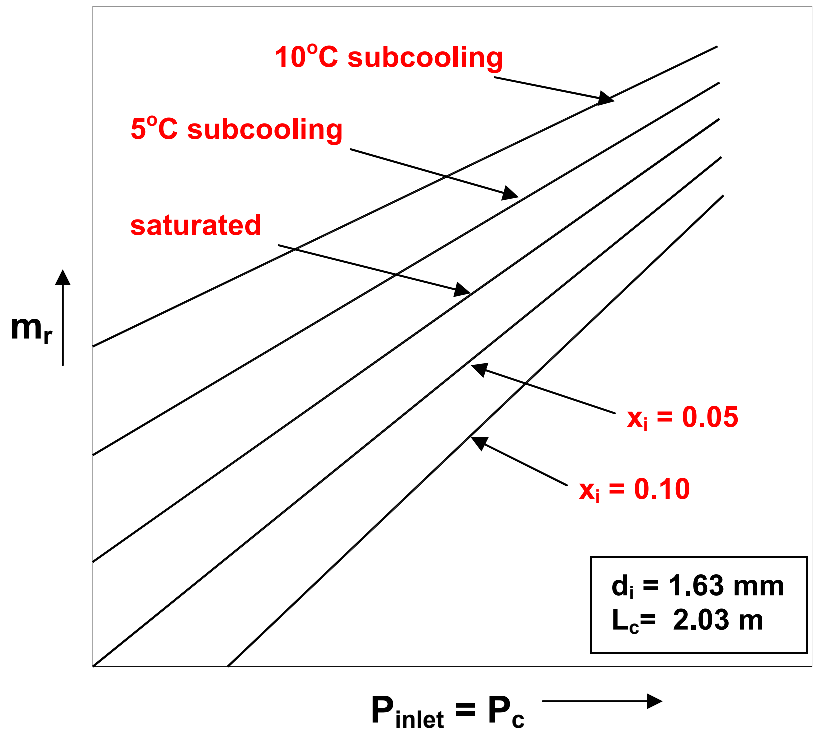

Fig.24.7: *Variation of refrigerant mass flow rate with inlet state for the standard capillary tube (Choked flow condition)*

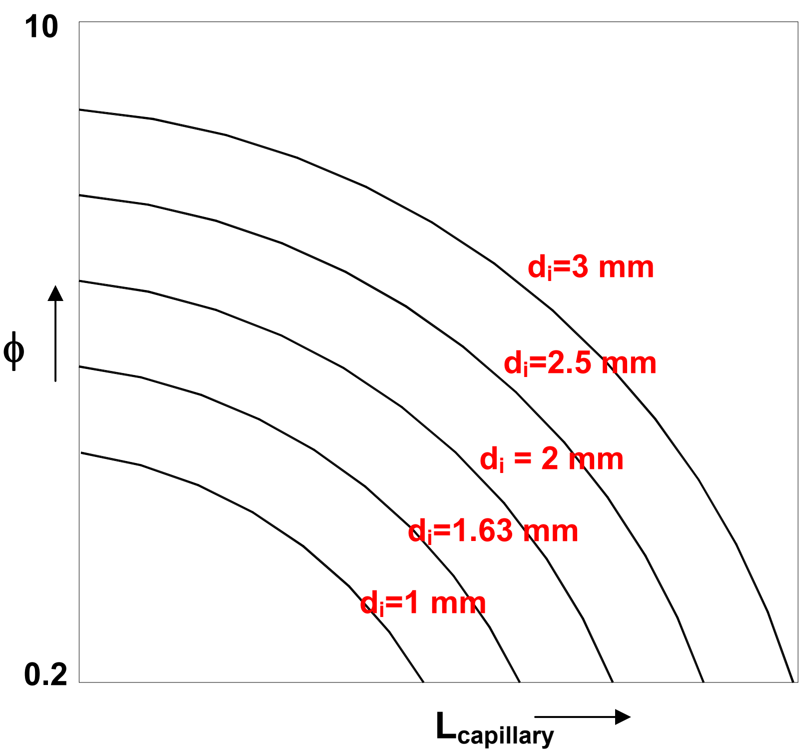

Fig.24.8: *Variation of flow correction factor ϕ with capillary tube length and diameter (Choked flow condition)*

24.2.4. Advantages and disadvantages of capillary tubes

Some of the advantages of a capillary tube are:

1. It is inexpensive.
2. It does not have any moving parts hence it does not require maintenance
3. Capillary tube provides an open connection between condenser and the evaporator hence during off-cycle, pressure equalization occurs between condenser and evaporator. This reduces the starting torque requirement of the motor since the motor starts with same pressure on the two sides of the compressor. Hence, a motor with low starting torque (squirrel cage Induction motor) can be used.
4. Ideal for hermetic compressor based systems, which are critically charged and factory assembled.

Some of the disadvantages of the capillary tube are:

1. It cannot adjust itself to changing flow conditions in response to daily and seasonal variation in ambient temperature and load. Hence, COP is usually low under off design conditions.
2. It is susceptible to clogging because of narrow bore of the tube, hence, utmost care is required at the time of assembly. A filter-drier should be used ahead of the capillary to prevent entry of moisture or any solid particles
3. During off-cycle liquid refrigerant flows to evaporator because of pressure difference between condenser and evaporator. The evaporator may get flooded and the liquid refrigerant may flow to compressor and damage it when it starts. Therefore critical charge is used in capillary tube based systems. Further, it is used only with hermetically sealed compressors where refrigerant does not leak so that critical charge can be used. Normally an accumulator is provided after the evaporator to prevent slugging of compressor

24.3. Automatic Expansion Valve (AEV)

An Automatic Expansion Valve (AEV) also known as a constant pressure expansion valve acts in such a manner so as to maintain a constant pressure and thereby a constant temperature in the evaporator. The schematic diagram of the valve is shown in Fig. 24.9. As shown in the figure, the valve consists of an adjustment spring that can be adjusted to maintain the required temperature in the evaporator. This exerts force F_s on the top of the diaphragm. The atmospheric pressure, P_o also acts on top of the diaphragm and exerts a force of $F_o = P_o A_d$, A_d being the area of the diaphragm. The evaporator pressure P_e acts below the diaphragm. The force due to evaporator pressure is $F_e = P_e A_d$. The net downward force $F_s + F_o - F_e$ is fed to the needle by the diaphragm. This net force along with the force due to follow-up spring F_{fs} controls the location of the needle with respect to the orifice and thereby controls the orifice opening.

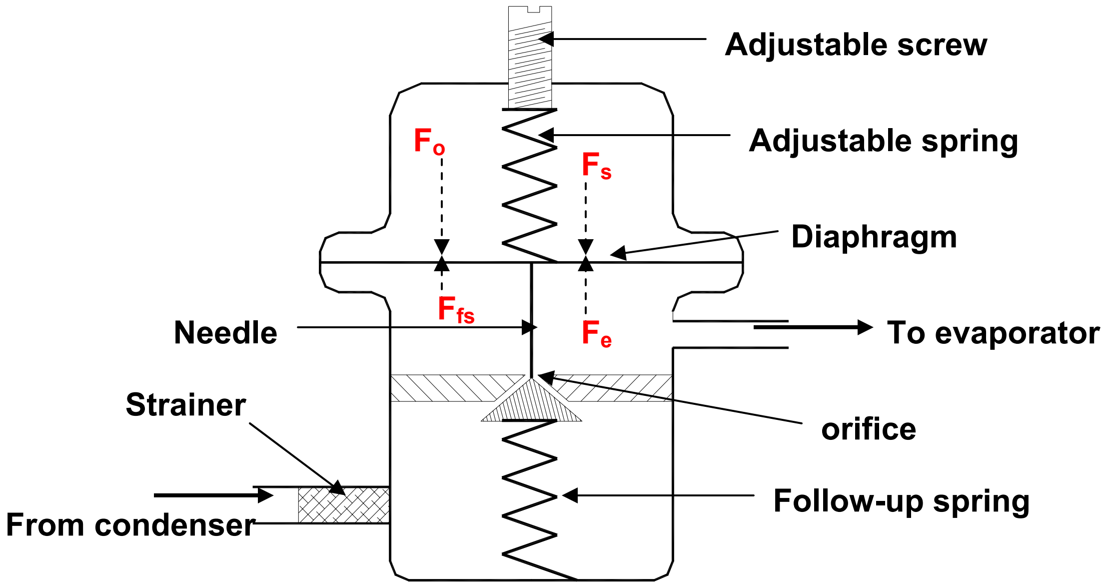

Fig.24.9*: Schematic of an Automatic Expansion Valve*

If $F_e + F_{fs} > F_s + F_o$ the needle will be pushed against the orifice and the valve will be fully closed.

On the other hand if $F_e + F_{fs} < F_s + F_o$, the needle will be away from the orifice and the valve will be open. Hence the relative magnitude of these forces controls the mass flow rate through the expansion valve.

The adjustment spring is usually set such that during off-cycle the valve is closed, that is, the needle is pushed against the orifice. Hence,

$$\boldsymbol{F_{eo} + F_{fso} > F_{so} + F_o}$$

Where, subscript $_o$ refers to forces during off cycle. During the off-cycle, the refrigerant remaining in the evaporator will vaporize but will not be taken out by the compressor, as a result the evaporator pressure rises during the off-cycle as shown in Fig.24.10.

When the compressor is started after the off-cycle period, the evaporator pressure P_e starts decreasing at a very fast rate since valve is closed; refrigerant is not fed to evaporator while the compressor removes the refrigerant from the evaporator. This is shown in Fig.24.10. As P_e decreases the force F_e decreases from F_{eo} to $(F_{eo} - \Delta F_e)$. At one stage, the sum $F_e + F_{fs}$ becomes less than $F_s + F_o$, as a result the needle stand moves downwards (away from the needle stand) and the valve opens. Under this condition,

$$(F_{eo} - \Delta F_e) + F_{fso} < F_{so} + F_o$$

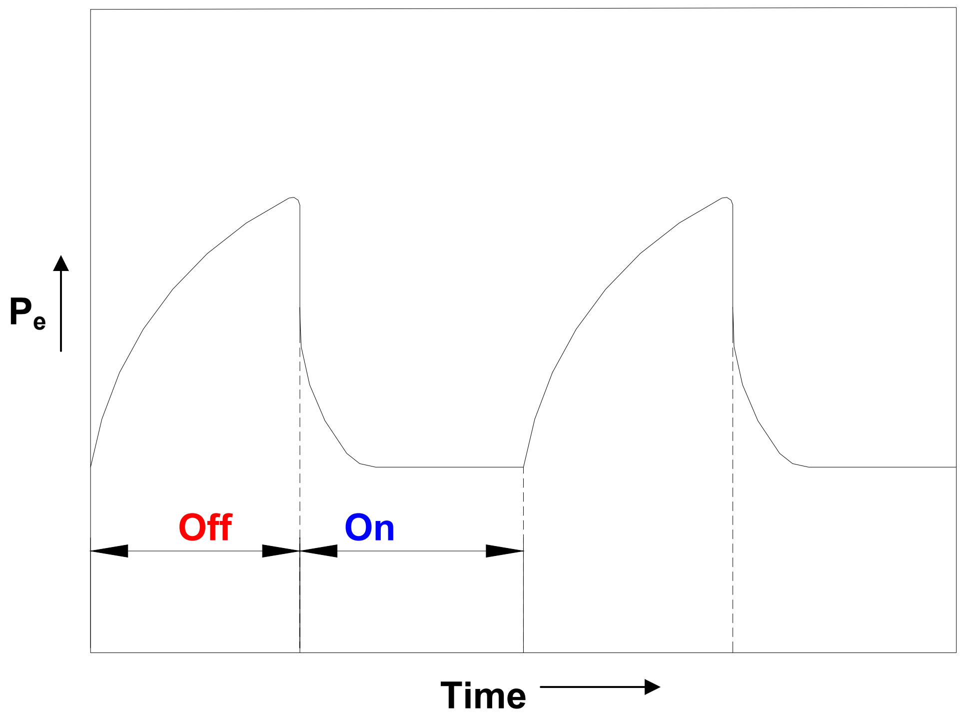

***Fig.24.10**: Variation of evaporator pressure during on- and off-cycles of an AEV based refrigeration system*

When the refrigerant starts to enter the evaporator, the evaporator pressure does not decrease at the same fast rate as at starting time. Thus, the movement of the needle stand will slow down as the refrigerant starts entering. As the needle moves downwards, the adjustment spring elongates, therefore the force F_s decreases from its off-cycle value of F_{s0}, the decrease being proportional to the movement of the needle.

As the needle moves downwards, the follow-up spring is compressed; as a result, F_{fs} increases from its off-cycle value. Hence, the final equation may be written as,

$$(F_{eo} - \Delta F_e) + (F_{fso} + \Delta F_{fs}) = (F_{so} - \Delta F_s) + F_o \quad \textbf{or}$$

$$F_e + F_{fs} = F_s + F_o \quad = \text{constant} \qquad (24.18)$$

The constant is sum of force due to spring force and the atmospheric pressure, hence it depends upon position of adjustment spring. This will be the equilibrium position. Then onwards, the valve acts in such a manner that the evaporator pressure remains constant as long as the refrigeration load is constant. At this point, the mass flow rate through the valve is the same as that through the compressor.

24.3.1. Effect of Load Variation

The mass flow rate through the valve is directly proportional to the pressure drop through the orifice (P_c–P_e) and the area of the orifice opening (needle position). At constant condenser pressure the mass flow rate will decrease if the evaporator pressure p_e increases or as the orifice opening becomes narrower.

Decrease In Load

If the refrigeration load decreases, there is a tendency in the evaporator for the evaporator temperature to decrease and thereby the evaporator pressure (saturation pressure) also decreases. This decreases the force F_e. The sum F_e+F_{fs} will become less than the sum on right hand side of Equation (28.18) and the needle stand will be pushed downwards opening the orifice wider. This will increase the mass flow rate through the valve. This is opposite of the requirement since at lower load, a lower mass flow rate of the refrigerant is required. This is the drawback of this valve that it counteracts in an opposite manner since it tries to keep the evaporator pressure at a constant value. In Figure 24.11, point *A* is the normal position of the value and *B* is the position at reduced load and wider opening. It is observed that both these are at same evaporator pressure. The compressor capacity remains the same as at *A*. The valve feeds more refrigerant to the evaporator than the compressor can remove from the evaporator. This causes accumulation of liquid refrigerant in the evaporator. This is called “flooding” of the evaporator. The liquid refrigerant may fill the evaporator and it may overflow to the compressor causing damage to it.

Increase In Load

On the other hand if the refrigeration load increases or the evaporator heat transfer rate increases, the evaporator temperature and pressure will increase for a flooded evaporator. This will increase F_e. A look at the schematic diagram reveals that this will tend to move the needle stand upwards, consequently making the orifice opening narrower and decreasing the mass flow rate. Again the valve counteracts in a manner opposite to what is required. This shifts the operating point from *A* to point *C* where the compressor draws out more refrigerant than that fed by the expansion valve leading to *starving* of the evaporator.

The adjustment of evaporator pressure and temperature is carried out by adjustment spring. An increase in the tension of adjustment spring increases F_s

so that the evaporator pressure at which balance occurs, increases. That is, the regulated temperature increases.

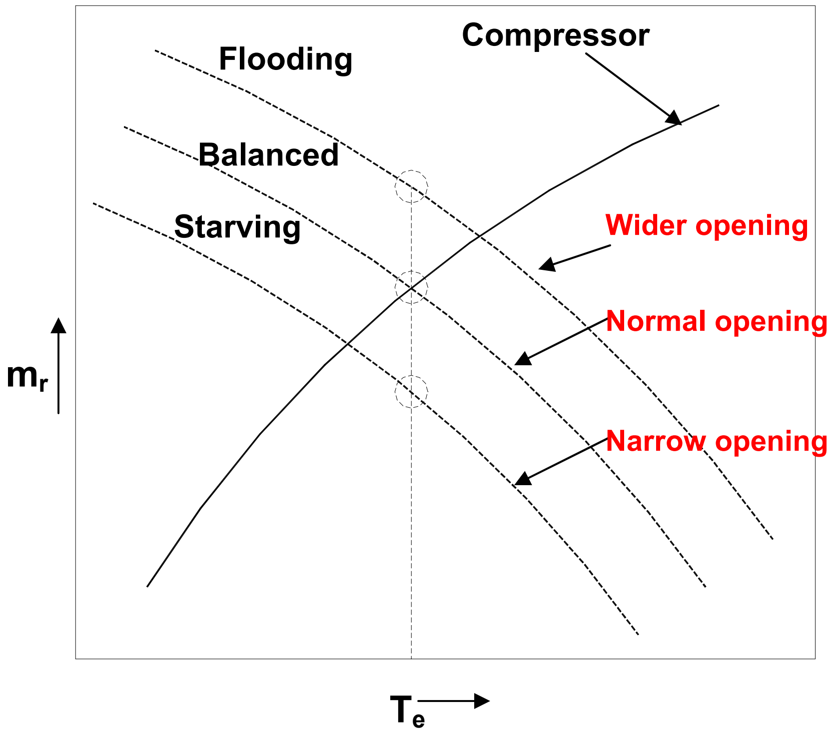

Fig.24.11*: Effect of load variation on balance point of the system using AEV*

24.3.2. Applications of automatic expansion valve

The automatic expansion valves are used wherever constant temperature is required, for example, milk chilling units and water coolers where freezing is disastrous. In air-conditioning systems it is used when humidity control is by DX coil temperature. Automatic expansion valves are simple in design and are economical. These are also used in home freezers and small commercial refrigeration systems where hermetic compressors are used. Normally the usage is limited to systems of less than 10 TR capacities with critical charge. Critical charge has to be used since the system using AEV is prone to flooding. Hence, no receivers are used in these systems. In some valves a diaphragm is used in place of bellows.

24.4. Flow Rate through orifice

In variable area type expansion devices, such as automatic and thermostatic expansion valves, the pressure reduction takes place as the fluid flows through an orifice of varying area. Let A_1 and A_2 be the areas at the inlet and the outlet of the orifice where, $A_1 > A_2$. Let V_1 and V_2 be the velocities, P_1 and

P_2 are the pressures and ρ_1 and ρ_2 be the densities at the inlet and outlet respectively of the orifice as shown in Figure 24.12.

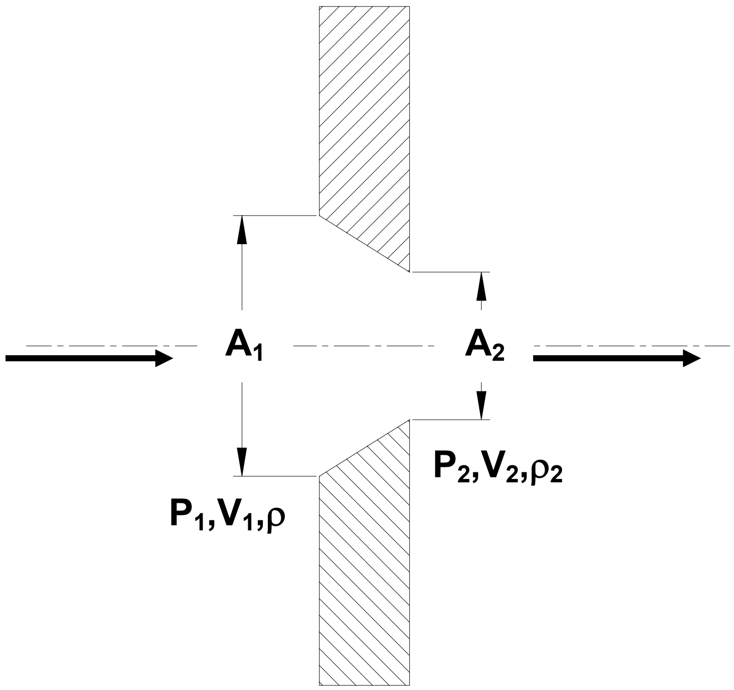

Fig.24.12: *Fluid flow through an orifice*

Then assuming steady, incompressible, inviscid flow and neglecting gravity, Bernoulli's equation may be used to write the flow rate through the orifice as follows.

<u>Mass Conservation</u>:

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 \qquad (24.19)$$

Assuming $\rho_1 = \rho_2$ we get

$$V_1 / V_2 = A_2 / A_1$$

<u>Bernoulli's Equation</u>:

$$\frac{P_1}{\rho_1} + \frac{V_1^2}{2} = \frac{P_2}{\rho_2} + \frac{V_2^2}{2} \qquad (24.20)$$

Therefore,

$$\frac{P_1 - P_2}{\rho_1} = \frac{V_2^2}{2}\left(1.0 - \frac{V_1^2}{V_2^2}\right) = \frac{V_2^2}{2}\left(1.0 - \frac{A_2^2}{A_1^2}\right) \qquad (24.21)$$

Ideal Flow Rate :

$$Q_{ideal} = A_2 V_2 = A_2 \sqrt{\frac{2(P_1 - P_2)}{\rho_1}} \frac{1.0}{\sqrt{1.0 - (A_2 / A_1)^2}} \qquad (24.22)$$

Defining

$$M = \frac{1.0}{\sqrt{1.0 - (A_2 / A_1)^2}}, \text{ we get}$$

$$Q_{ideal} = M A_2 \sqrt{\frac{2(P_1 - P_2)}{\rho_1}} \qquad (24.23)$$

The actual flow through the orifice is less than ideal flow because viscous effects are not included in the above treatment. An empirical coefficient C_D, called discharge coefficient is introduced to account for the viscous effects.

$$Q_{actual} = C_D\, Q_{ideal} = C_D M A_2 \sqrt{\frac{2(P_1 - P_2)}{\rho_1}} \qquad (24.24)$$

Introducing flow coefficient $K = C_D M$

$$Q_{actual} = K A_2 \sqrt{\frac{2(P_1 - P_2)}{\rho_1}}$$

To account for compressibility another empirical constant Y is introduced for actual mass flow rate. Hence, the mass flow rate is expressed as,

$$\dot{m} = K \rho_1 Y A_2 \sqrt{\frac{2(P_1 - P_2)}{\rho_1}} \qquad (24.25)$$

The area of the orifice opening is usually controlled to control the mass flow rate through the expansion valve. It is observed that the mass flow rate depends upon the difference between the condenser and evaporator pressures also. It is curious that single phase relations have been given above while it was shown that during expansion of high pressure liquid, the refrigerant flashes into a low pressure mixture of liquid and vapour as it flows through the expansion valve. Actually, studies show that the refrigerant remains in a thermodynamic metastable liquid state as it flows through the orifice of the expansion valve. That is, it remains a liquid at a lower pressure and temperature during its passage through the orifice. It flashes into a mixture of liquid and vapour as soon as it emerges out of the orifice of the valve. This kind of phenomenon has been observed in the initial sections of transparent capillary tubes also.

24.5. Thermostatic Expansion Valve (TEV)

Thermostatic expansion valve is the most versatile expansion valve and is most commonly used in refrigeration systems. A thermostatic expansion valve maintains a constant degree of superheat at the exit of evaporator; hence it is most effective for dry evaporators in preventing the slugging of the compressors since it does not allow the liquid refrigerant to enter the compressor. The schematic diagram of the valve is given in Figure 24.13. This consists of a feeler bulb that is attached to the evaporator exit tube so that it senses the temperature at the exit of evaporator. The feeler bulb is connected to the top of the bellows by a capillary tube. The feeler bulb and the narrow tube contain some fluid that is called *power fluid*. The power fluid may be the same as the refrigerant in the refrigeration system, or it may be different. In case it is different from the refrigerant, then the TEV is called *TEV with cross charge*. The pressure of the power fluid P_p is the saturation pressure corresponding to the temperature at the evaporator exit. If the evaporator temperature is T_e and the corresponding saturation evaporator pressure is P_e, then the purpose of TEV is to maintain a temperature $T_e+\Delta T_s$ at the evaporator exit, where ΔT_s is the degree of superheat required from the TEV. The power fluid senses this temperature $T_e+\Delta T_s$ by the feeler bulb and its pressure P_p is the saturation pressure at this temperature. The force F_p exerted on top of bellows of area A_b due to this pressure is given by:

$$\mathbf{F_p = A_b\, P_p} \qquad (24.26)$$

The evaporator pressure is exerted below the bellows. In case the evaporator is large and has a significant pressure drop, the pressure from evaporator exit is fed directly to the bottom of the bellows by a narrow tube. This is called pressure-equalizing connection. Such a TEV is called *TEV with external equalizer,* otherwise it is known as *TEV with internal equalizer*. The force F_e exerted due to this pressure P_e on the bottom of the bellows is given by

$$\mathbf{F_e = A_b\, P_e} \qquad (24.27)$$

The difference of the two forces F_p and F_e is exerted on top of the needle stand. There is an adjustment spring below the needle stand that exerts an upward spring force F_s on the needle stand. In steady state there will be a force balance on the needle stand, that is,

$$\mathbf{F_s = F_p - F_e} \qquad (24.28)$$

During off-cycle, the evaporator temperature is same as room temperature throughout, that is, degree of superheat ΔT_s is zero. If the power fluid is the same as the refrigerant, then $P_p = P_e$ and $F_p = F_e$. Therefore any arbitrarily small spring force F_s acting upwards will push the needle stand against the orifice and keep the TEV closed. If it is *TEV with cross charge* or if there is a little degree of superheat during off-cycle then for TEV to remain closed during off-cycle, F_s should be slightly greater than $(F_p - F_e)$.

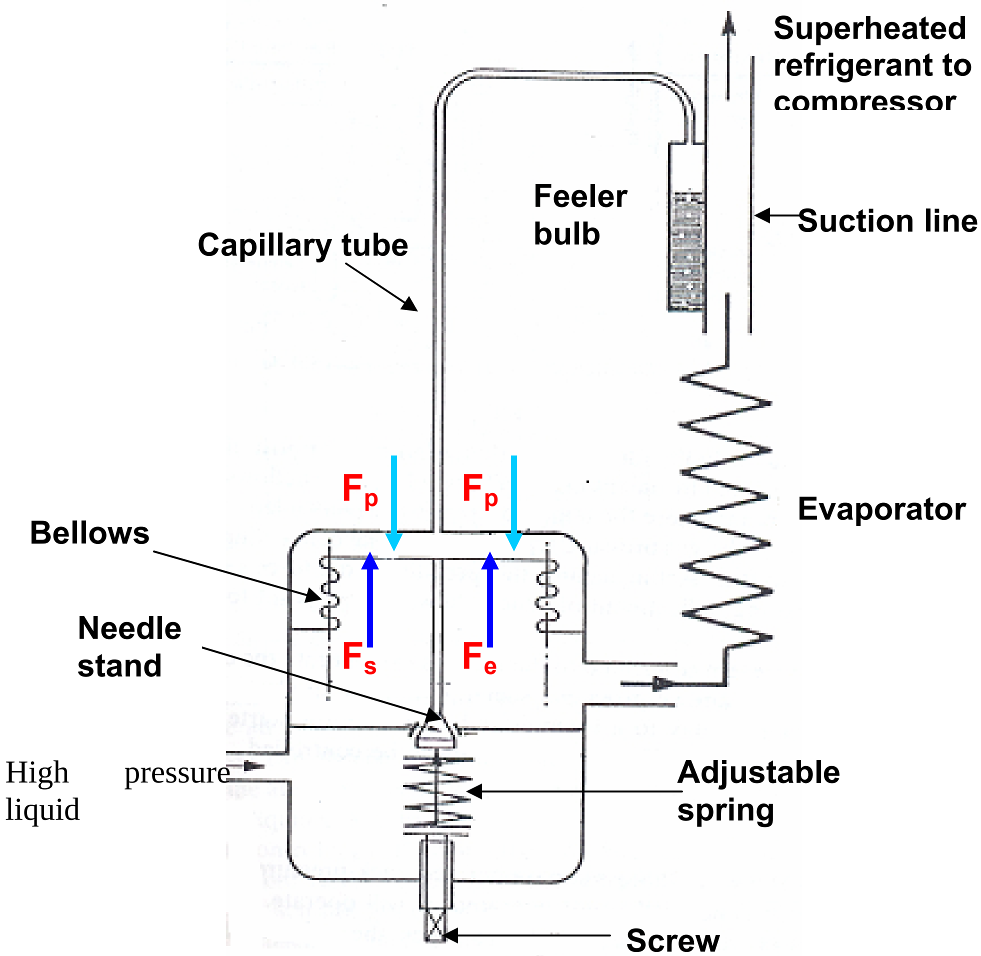

***Fig.24.13**: Schematic of a Thermostatic Expansion Valve (TEV)*

As the compressor is started, the evaporator pressure decreases at a very fast rate hence the force F_e decreases at a very fast rate. This happens since TEV is closed and no refrigerant is fed to evaporator while compressors draws out refrigerant at a very fast rate and tries to evacuate the evaporator. The force F_p does not change during this period since the evaporator temperature does not change. Hence, the difference F_p-F_e, increases as the compressor runs for some time after starting. At one point this difference becomes greater than the spring force F_s and pushes the needle stand downwards opening the orifice. The valve is said to open up. Since a finite downward force is required to open the valve, a minimum degree of superheat is required for a finite mass flow rate.

As the refrigerant enters the evaporator it arrests the fast rate of decrease of evaporator pressure. The movement of needle stand also slows down. The spring, however gets compressed as the needle stand moves downward to open the orifice. If F_{s0} is the spring force in the rest position, that is, off-cycle, then during open valve position

$$\mathbf{F_s = F_{s0} + \Delta F_s}$$

Eventually, the needle stand reaches a position such that,

$$F_s = F_p - F_e = A_b\,(P_p - P_e) \qquad (24.29)$$

That is, F_p is greater than F_e or P_p is greater than P_e. The pressure P_p and P_e are saturation pressures at temperature ($T_e + \Delta T_s$) and T_e respectively. Hence, for a given setting force F_s of the spring, TEV maintains the difference between F_p and F_e or the degree of superheat ΔT_s constant.

$$\begin{aligned} \Delta T_s &\propto (F_p - F_e) \qquad (24.30)\\ &\propto F_s \end{aligned}$$

This is irrespective of the level of P_e, that is, evaporator pressure or temperature, although degree of superheat may be slightly different at different evaporator temperatures for same spring force, F_s. It will be an ideal case if the degree of superheat is same at all evaporator temperatures for a given spring force.

24.5.1. Effect of Load Variation

If the load on the plant increases, the evaporation rate of liquid refrigerant increases, the area available for superheating the vapour increases. As the degree of superheat increases, pressure of power fluid P_p increases, the needle stand is pushed down and the mass flow rate of refrigerant increases. This is the ideal case. The evaporation rate of refrigerant is proportional to the load and the mass flow rate supplied through the expansion valve is also proportional to the load.

On the other hand, if the load on the plant decreases, the evaporation rate of refrigerant decreases, as a result the degree of superheat decreases. The thermostatic expansion valve reacts in such a way so as to reduce the mass flow rate through it. The flow rate of refrigerant in this valve is proportional to the evaporation rate of refrigerant in the evaporator. Hence, this valve always establishes balanced flow condition of flow between compressor and itself.

24.5.2. TEV with cross charge

Figure 24.14 shows the saturated vapour line with pressure along the ordinate. The difference between P_p and P_e is proportional to the spring force, F_s and their corresponding projection from the saturated vapour line is the degree of superheat given by a set of P_p and P_e. The figure shows three sets of P_p and P_e

for the same spring force at three evaporator temperatures say –40^{o}C, -20^{o}C and 5^{o}C. It is observed that at location *A*, the degree of superheat is very large whereas at location *C* the degree of superheat is very small for the same spring force setting proportional to (P_p–P_e). This would not have been the case if the saturated vapour line was a straight line. It is observed that if the spring is set for say a superheat of 10^{o}C at –40^{o}C evaporator temperature, the degree of superheat will become almost zero at higher temperature (Fig.24.14). As a result; when the plant is started at warm temperature, there is a possibility of flooding of evaporator. If degree of superheat is set to avoid flooding at say 5^{o}C, then at the design point of say – 40^{o}C, the superheat will be very large and it will starve the evaporator. This can be corrected if a fluid different from refrigerant is used in the feeler bulb as power fluid. Such a TEV is called *TEV with cross charge*. Figure 24.15 shows the saturated vapour line for the power fluid as well as the refrigerant in the system. The projection for P_p is taken from the saturation line for power fluid and it shows the temperature at the exit of the evaporator. The power fluid is such that at any temperature it has lower saturation pressure than that of the refrigerant in the system, so that as the evaporator temperature increases the degree of superheat increases. The projection for P_e is taken from the saturation line of refrigerant and it indicates the evaporator temperature. It is observed that for the two different locations A and B, the degree of superheat is almost same for all evaporator temperatures. Hence cross charge helps in maintaining the same degree of superheat at all evaporator temperatures. Cross-charged valves perform satisfactorily in a narrow range of temperatures that must be specified while ordering a valve.

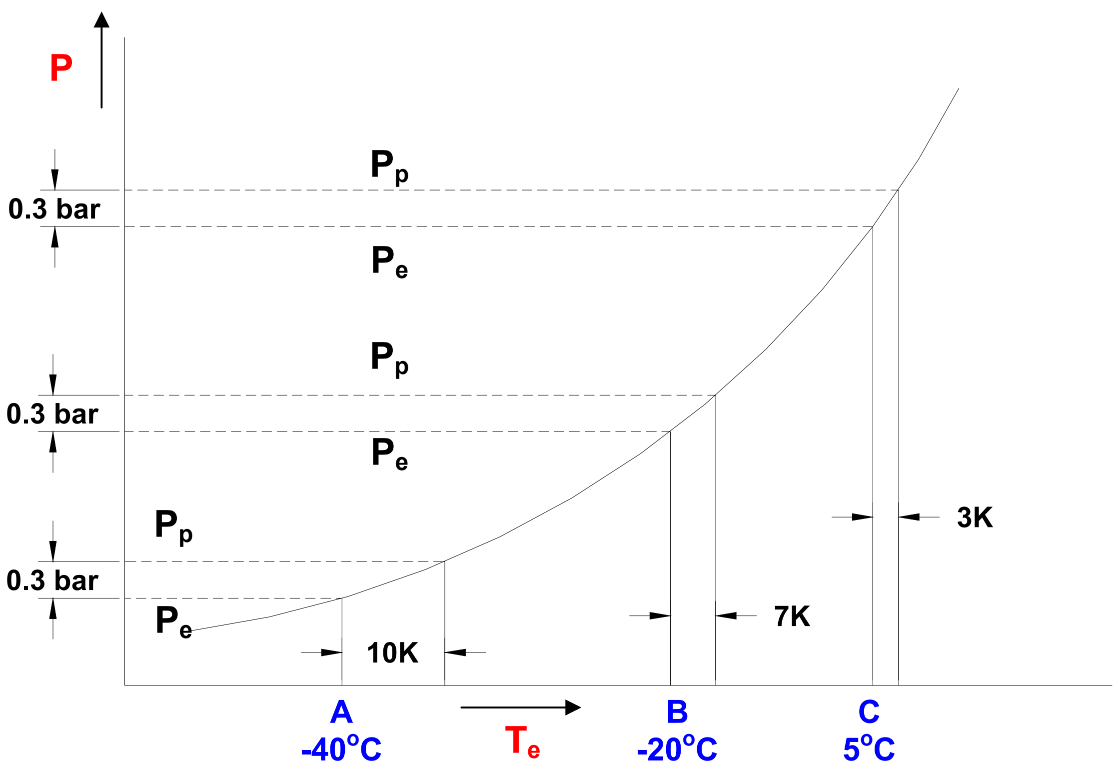

Fig.24.14: Vapour pressure curve of refrigerant and power fluid

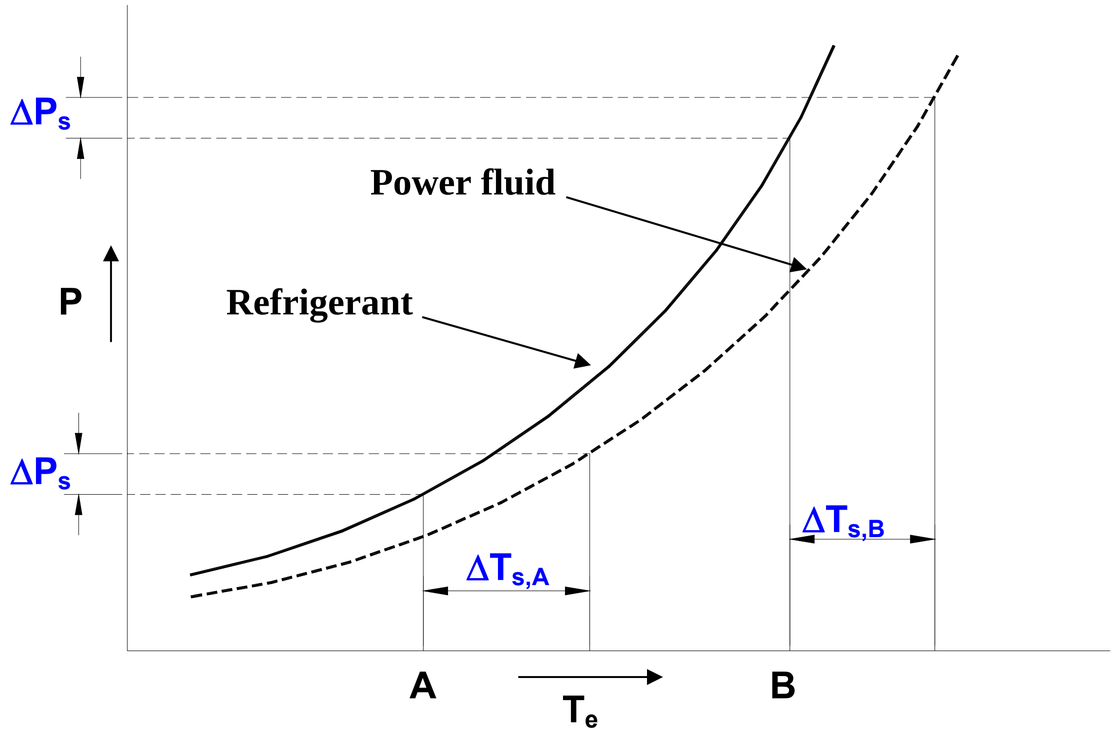

***Fig.24.15**: Vapour pressure curves of refrigerant and power fluid (cross-charged TEV)*

24.5.3. TEV with External Pressure Equalizer

The pressure drop of the refrigerant is quite significant in large evaporators, for example in direct expansion coils with a single long tube. Thermostatic expansion valve maintains $F_p - F_e = A_b(P_p - P_e)$ at a constant value equal to spring force. The pressure P_p is the saturation pressure at ($T_e + \Delta T_s$) while P_e is saturation pressure at T_e. In a large evaporator, due to pressure drop ΔP_e, the pressure at exit is say, $P_e - \Delta P_e$ and corresponding saturation temperature at exit of evaporator is $T_e - \Delta T_e$. The superheat ΔT_s corresponds to evaporator pressure P_e and temperature T_E. Therefore, effective superheat at evaporator exit is $\Delta T_s + \Delta T_e$. This may become very large and may result in low COP and lower volumetric efficiency of compressor. To correct for this, TEV is provided with a tapping, which feeds the pressure $P_e - \Delta P_e$ from evaporator exit to the bottom of bellows. This will result in a degree of superheat equal to the set value ΔT_s. A TEV with this provision is called *TEV with External Pressure Equalizer.* In this TEV a stuffing box is provided between pushpins and the valve body so that evaporator inlet pressure is not communicated to the bottom of bellows. Figure 24.16 shows a TEV with an external equalizer arrangement with pressure tapping.

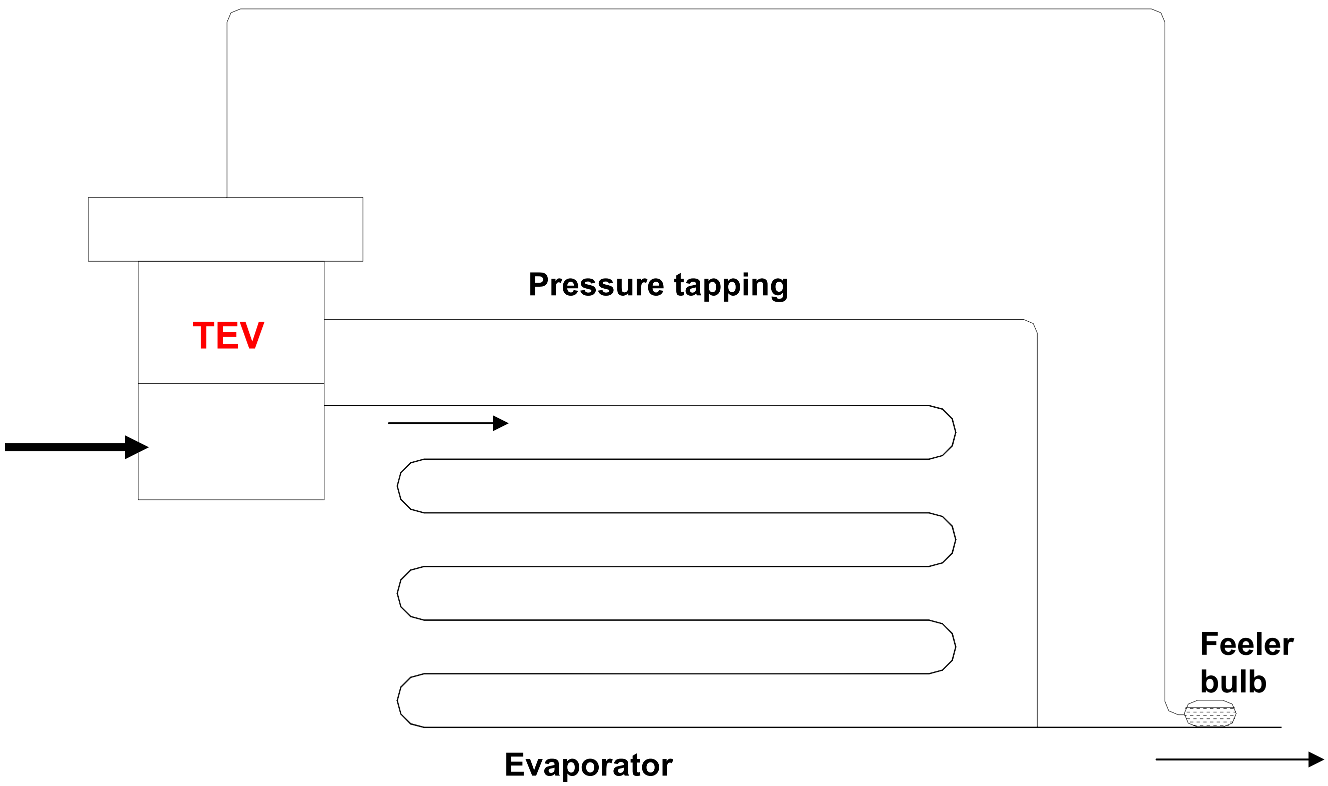

***Fig.24.16**: A Thermostatic Expansion Valve with an external equalizer*

In any case a large evaporator pressure drop leads to a lower COP; hence a number of parallel paths or circuits are provided in the evaporator. The refrigerant is fed to these paths by a single TEV fitted with a distributor. In such a case, it is recommended that external pressure equalizer be used and care taken to ensure that all the paths are symmetric and have the same length.

24.5.4. Fade-out point and pressure limiting characteristics of TEV:

The volume of power fluid in the feeler bulb and the connecting tube is constant, therefore heating and cooling of power fluid is a constant specific volume process. Figure 24.17 shows the pressure-temperature variation of the power fluid. The bulb usually has a mixture of liquid and vapour and the pressure exerted by power fluid corresponds to its saturation pressure. The pressure of the power fluid increases rather rapidly as its temperature increases since the liquid evaporates and it has to be accommodated in fixed volume. This sharp rise in pressure with temperature continues until point *B* on the saturation curve, where no liquid is left. Since the pressure of the power fluid does not increase significantly beyond *B*, the valve does not open any wider, $p_p \approx$ constant, hence for a fixed spring setting p_e remains almost constant and thereby limits the pressure in the evaporator to *Maximum Operating pressure.* It was observed in an earlier lecture on reciprocating compressors that the power requirement of a reciprocating compressor is maximum at a certain evaporator pressure. The air-conditioning systems usually operate near the peak while the refrigeration systems such as those for ice cream or frozen food operate on the left side of the peak power. It was shown that during pull-down, the power requirement would pass through the power peak if the evaporator were kept fully supplied with liquid. It is however uneconomical to provide a large electric motor to meet the power requirement of the peak for small times during pull-down. The power requirement at the design point on the left leg is small. A motor capable of providing normal power can be used if the TEV makes the evaporator starve (reduces mass flow rate to it) and limits the pressure during pull-down when the load is high. Charging the bulb with limited mass of power fluid so that it is entirely vapour above a maximum evaporating pressure and temperature achieves this purpose. If rapid cooling is required from the refrigeration system then this cannot be used.

The limit charged valve is prone to failure known as *reversal*. The feeler bulb has vapour only. The head of the feeler bulb is usually colder than the rest of it, as a result a small amount of vapor can condense in this region. This colder region will have lower saturation pressure that will decide the pressure of the feeler bulb and this low pressure may be insufficient to open the valve. This is avoided by keeping the head of the valve warm by internal circulation.

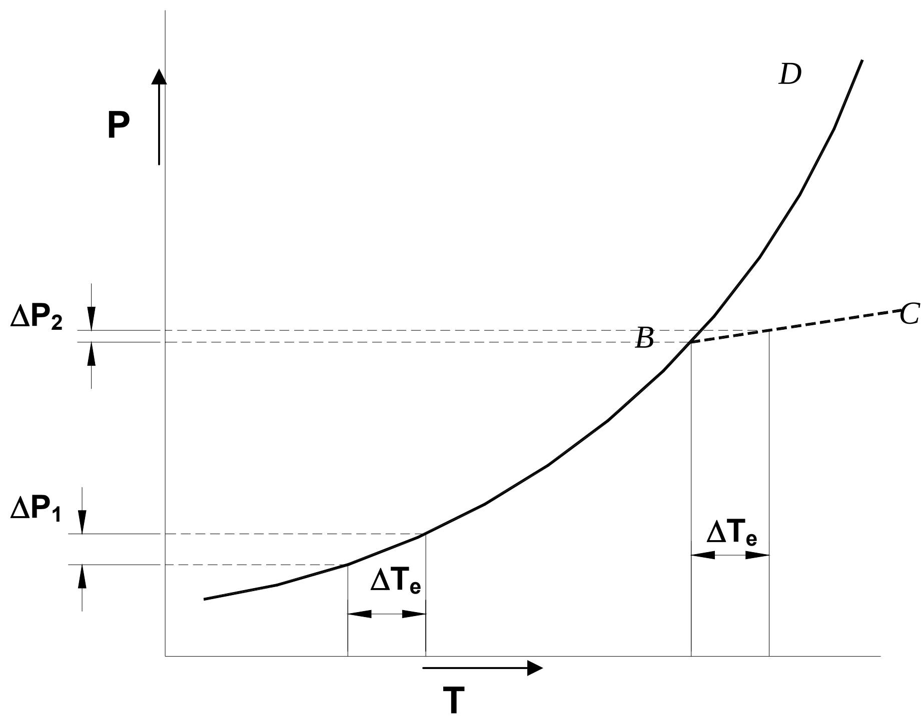

***Fig.24.17**: Variation of power fluid pressure with temperature in a limit charged TEV*

24.4.5. Advantages, disadvantages and applications of TEV

The advantages of TEV compared to other types of expansion devices are:

1. It provides excellent control of refrigeration capacity as the supply of refrigerant to the evaporator matches the demand
2. It ensures that the evaporator operates efficiently by preventing starving under high load conditions
3. It protects the compressor from slugging by ensuring a minimum degree of superheat under all conditions of load, if properly selected.

However, compared to capillary tubes and AEVs, a TEV is more expensive and proper precautions should be taken at the installation. For example, the feeler bulb must always be in good thermal contact with the refrigerant tube. The feeler bulb should preferably be insulated to reduce the influence of the ambient air. The bulb should be mounted such that the liquid is always in contact with the refrigerant tubing for proper control.

The use of TEV depends upon degree of superheat. Hence, in applications where a close approach between the fluid to be cooled and evaporator temperature is desired, TEV cannot be used since very small extent of superheating is available for operation. A counter flow arrangement can be used to achieve the desired superheat in such a case. Alternately, a subcooling HEX may be used and the feeler bulb mounted on the vapour exit line of the HEX. The valves with bellows have longer stroke of the needle, which gives extra sensitivity compared to diaphragm type of valve. But valves with bellows are more expensive.

Thermostatic Expansion Valves are normally selected from manufacturers' catalogs. The selection is based on the refrigeration capacity, type of the working fluid, operating temperature range etc. In practice, the design is different to suit different requirements such as single evaporators, multi-evaporators etc.

24.6. Float type expansion valves:

Float type expansion valves are normally used with flooded evaporators in large capacity refrigeration systems. A float type valve opens or closes depending upon the liquid level as sensed by a buoyant member, called as float. The float could take the form of a hollow metal or plastic ball, a hollow cylinder or a pan. Thus the float valve always maintains a constant liquid level in a chamber called as float chamber. Depending upon the location of the float chamber, a float type expansion valve can be either a low-side float valve or a high-side float valve.

24.6.1. Low-side float valves:

A low-side float valve maintains a constant liquid level in a flooded evaporator or a float chamber attached to the evaporator. When the load on the system increases, more amount of refrigerant evaporates from the evaporator. As a result, the refrigerant liquid level in the evaporator or the low-side float chamber drops momentarily. The float then moves in such a way that the valve opening is increased and more amount of refrigerant flows into the evaporator to take care of the increased load and the liquid level is restored. The reverse process occurs when the load falls, i.e., the float reduces the opening of the valve and less amount of refrigerant flows into the evaporator to match the reduced load. As mentioned, these valves are normally used in large capacity systems and normally a by-pass line with a hand-operated expansion is installed to ensure system operation in the event of float failure.

24.6.2. High-side float valves:

Figure 24.18 shows the schematic of a high-side float valve. As shown in the figure, a high-side float valve maintains the liquid level constant in a float chamber that is connected to the condenser on the high pressure side. When the load increases, more amount of refrigerant evaporates and condenses. As a result, the liquid level in the float chamber rises momentarily. The float then opens the valve more to allow a higher amount of refrigerant flow to cater to the increased load, as a result the liquid level drops back to the original level. The reverse happens when the load drops. Since a high-side float valve allows only a fixed amount of refrigerant on the high pressure side, the bulk of the refrigerant is stored in the low-pressure side (evaporator). Hence there is a possibility of flooding of evaporator followed by compressor slugging. However, unlike low-side float valves, a high-side float valve can be used with both flooded as well as direct expansion type evaporators.

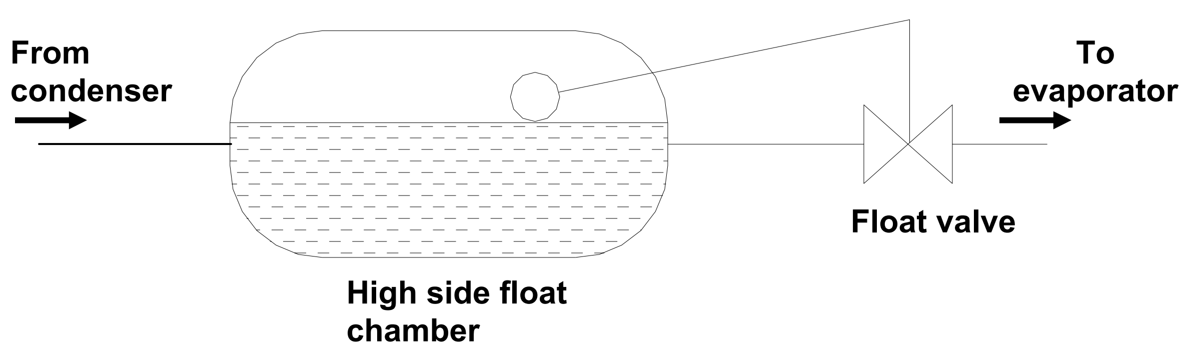

Fig.24.18*: Schematic of a high-side float valve*

24.7. Electronic Type Expansion Valve

The schematic diagram of an electric expansion valve is shown in Fig.24.19. As shown in the figure, an electronic expansion valve consists of an orifice and a needle in front it. The needle moves up and down in response to magnitude of current in the heating element. A small resistance allows more current to flow through the heater of the expansion valve, as a result the valve opens wider. A small negative coefficient thermistor is used if superheat control is desired. The thermistor is placed in series with the heater of the expansion valve. The heater current depends upon the thermistor resistance that depends upon the refrigerant condition. Exposure of thermistor to superheated vapour permits thermistor to selfheat thereby lowering its resistance and increasing the heater current. This opens the valve wider and increases the mass flow rate of refrigerant. This process continues until the vapour becomes saturated and some liquid refrigerant droplets appear. The liquid refrigerant will cool the thermistor and increase its resistance. Hence in presence of liquid droplets the thermistor offers a large resistance, which allows a small current to flow through the heater making the valve opening narrower. The control of this valve is independent of refrigerant and refrigerant pressure; hence it works in reverse flow direction also. It is convenient to use it in year-round-air-conditioning systems, which serve as heat pumps in winter with reverse flow. In another version of it the heater is replaced by stepper motor, which opens and closes the valve with a great precision giving a proportional control in response to temperature sensed by an element.

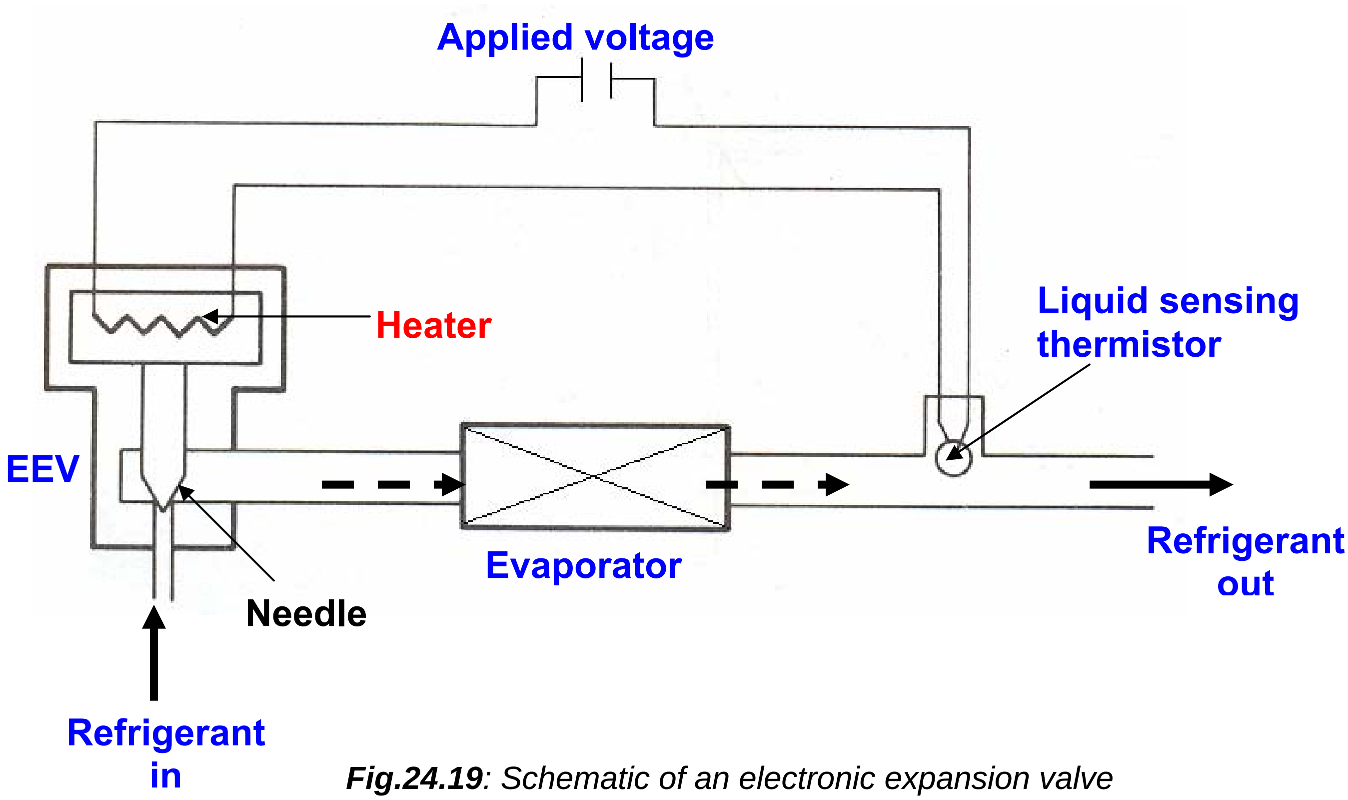

Fig.24.19: *Schematic of an electronic expansion valve*

24.8. Practical problems in operation of Expansion valves

Certain practical problems are encountered with expansion devices if either the selection and/or its operation are not proper. An oversized expansion device will overfeed the refrigerant or hunt (too frequent closing and opening) and not achieve the balance point. It may allow more refrigerant to flow to the evaporator and cause flooding and consequent slugging of the compressor with disastrous results.

A small valve on the other hand passes insufficient quantity of the refrigerant so that balance point may occur at a lower temperature. The mass flow rate through the expansion valve depends upon the pressure difference between condenser and evaporator. The condenser temperature and consequently the pressure decrease during winter for air-cooled as well as water-cooled condensers. As a result, the pressure difference is not sufficient for balance of flow between compressor and the expansion valve. Hence, the evaporator temperature and pressure decrease during winter months. This decreases the volumetric efficiency of the compressor and results in lower mass flow rate and lower cooling capacity. This may lead to disastrous results for hermetic compressors, which rely upon refrigerant flow rate for cooling of motor. At lower mass flow rates hermetic compressor may not be cooled sufficiently and may burn out. Hence, sometimes the condenser pressure must be kept artificially high so that adequate supply of refrigerant is achieved. Thus the natural advantage of lower condenser pressure is lost due to the need for maintaining the condenser pressure artificially high for proper functioning of the expansion device.

During summer months, the mass low rate through expansion valve is large because of large pressure difference. The corrective action taken by the system is to pass vapour through the expansion valve. This problem can occur if there is insufficient charge of refrigerant in the system so that the liquid seal at condenser exit is broken and vapour enters the expansion valve. It can occur because of higher elevation of expansion valve over the condenser so that there is static pressure drop to overcome gravitational force to reach the expansion valve, which causes flashing of refrigerant into a mixture of liquid and vapour. This is however not advisable since it leads to lower COP. Hence, it is advisable to use a liquid to vapour subcooling heat exchanger so that the liquid is subcooled and will not flash before entry into expansion valve.

Since the area available for refrigerant flow in the expansion device is normally very small, there is a danger of valve blockage due to some impurities present in the system. Hence, it is essential to use a filter/strainer before the expansion device, so that only refrigerant flows through the valve and solid particles, if any, are blocked by the filter/strainer. Normally, the automatic expansion valve and thermostatic expansion valves consist of in-built filter/strainers. However, when a capillary tube is used, it is essential to use a filter/dryer ahead of the capillary to prevent entry of any solid impurities and/or unbound water vapour into the capillary tube.

Questions and answers:

1. Which of the following statements are TRUE?

a) A capillary tube is a variable opening area type expansion device
b) In a capillary tube pressure drop takes place due to fluid friction
c) In a capillary tube pressure drop takes place due to fluid acceleration
d) In a capillary tube pressure drop takes place due to fluid friction and acceleration

Ans.: d)

2. Which of the following statements are TRUE?

a) The refrigerant mass flow rate through a capillary tube increases as condenser pressure decreases and evaporator pressure increases
b) The refrigerant mass flow rate through a capillary tube increases as condenser pressure increases and evaporator pressure decreases
c) A capillary tube tends to supply more mass flow rate as refrigeration load increases
c) A capillary tube tends to supply more mass flow rate as refrigeration load decreases

Ans.: b) and d)

3. Which of the following statements are TRUE?

a) A capillary tube based refrigeration system is a critically charged system
b) A capillary tube based refrigeration system does not use a receiver
c) Capillary tube based refrigeration systems employ open type compressors
d) In capillary tube based systems, pressure equalization takes place when compressor is off

Ans.: a), b) and d)

4. Which of the following statements are TRUE?

a) The mass flow rate through a capillary is maximum under choked flow conditions
b) The mass flow rate through a capillary is minimum under choked flow conditions
c) The enthalpy of refrigerant remains constant as it flows through a capillary tube

d) The enthalpy of refrigerant in a capillary tube decreases in the flow direction

Ans.: a) and d)

5. For a given refrigerant mass flow rate, the required length of a capillary tube increases as:

a) The degree of subcooling at the inlet decreases
b) The diameter of the capillary tube increases
c) The diameter of capillary tube decreases
d) Inlet pressure increases

Ans.: b) and d)

6. Which of the following statements are TRUE?

a) An automatic expansion valve maintains a constant pressure in the condenser
b) An automatic expansion valve maintains a constant pressure in the evaporator
c) In an automatic expansion valve, the mass flow rate of refrigerant increases as the refrigeration load increases
d) Automatic expansion valve based systems are critically charged

Ans.: b) and d)

7. A thermostatic expansion valve:

a) Maintains constant evaporator temperature
b) Maintains a constant degree of superheat
c) Increases the mass flow rate of refrigerant as the refrigeration load increases
d) Prevents slugging of compressor

Ans.: b), c) and d)

8. Which of the following statements are TRUE?

a) Cross-charging is used in TEV when the pressure difference across the evaporator is large
b) Cross-charging is used in TEV when the evaporator has to operate over a large temperature range
c) An external equalizer is used when pressure drop in evaporator is large
d) By limiting the amount of power fluid, the power peak during pull-down period can be avoided

Ans.: b), c) and d)

9. Which of the following statements are TRUE?

a) A float valve maintains a constant level of liquid in the float chamber
b) A float valve maintains a constant pressure in the float chamber
c) Low-side float valves are used with direct expansion type evaporators
d) High-side float valves are used in flooded type evaporators

Ans.: a)

10. Which of the following statements are TRUE?

a) An electronic expansion valve is bi-directional
b) In an electronic expansion valve, the refrigerant mass flow rate increases as the amount of liquid at evaporator exit increases
c) In an electronic expansion valve, the refrigerant mass flow rate increases as the temperature of refrigerant at evaporator exit increases
d) Electronic expansion valves are used in all-year air conditioning systems

Ans.: a), c) and d)

11. A thermostatic expansion valve uses R12 as the power fluid, and is used in a R12 based system operating at an evaporator temperature of 4^oC. The adjustable spring is set to offer a resistance equivalent to a pressure of 60 kPa. What is the degree of superheat?

Ans.: From the properties of R12, at 4^oC, the saturation pressure P_e is **350 kPa**.

Hence the pressure acting on the bellows/diaphragm due to the power fluid P_p is:

$$\mathbf{P_p = P_e + P_s = 350 + 60 = 410\ kPa}$$

The saturation temperature corresponding to a pressure of 410 kPa is **9^oC**

Hence the degree of superheat = 9 – 4 = 5^oC (Ans.)

12. For the above thermostat, what is the actual degree of superheat if there is a pressure drop of 22 kPa in the evaporator?

Ans.: The pressure of refrigerant at the exit of evaporator, $P_{e,exit}$ is:

$$\mathbf{P_{e,exit} = P_{e,inlet} - \Delta P_e = 350 - 22 = 328\ kPa}$$

The saturation temperature corresponding to 328 kPa is: **1.9^oC**

Hence the actual degree of superheat = 9 – 1.9 = 7.1^oC (Ans.)

This implies that a TEV with external equalizer is preferable to reduce the superheat

13. A straight-charged Thermostatic Expansion Valve (TEV) is designed to operate at an evaporator temperature of **7^oC** with a degree of superheat of **5 K**. R 134a is the refrigerant used in the refrigeration system as well as the bulb. Find a) The required spring pressure at the design condition; b) Assuming the spring pressure to remain constant, find the degree of superheat, if the same TEV operates at an evaporator temperature of **-23^oC**. The saturation pressure of R134a can be estimated using Antoine's equation given by:

$$p_{sat} = \exp^{\left(14.41 - \frac{2094}{T - 33.06}\right)} \quad \text{where } p_{sat} \text{ is in kPa and } T \text{ is in K}$$

Ans.: At the design conditions the evaporator temperature is 7^oC and degree of superheat is 5 K.

Hence the required adjustable spring pressure, P_s is:

$$P_s = P_{sat}(12^oC) - P_{sat}(7^oC)$$

Using Antoine's equation given above, we find that:

$$P_{sat}(12^oC) = 445.2 \text{ kPa, and}$$

$$P_{sat}(7^oC) = 376.2 \text{ kPa}$$

$$\text{Hence, } Ps = 445.2 - 376.2 = 69 \text{ kPa}$$

If the above TEV is operated at -23^oC evaporator temperature, then the pressure exerted by the power fluid is:

$$Pp = P_{sat}(-23^oC) + P_s = 116.5 + 69 = 185.5 \text{ kPa}$$

The corresponding saturation temperature is T_{sat}(189.5 kPa) = 261 K = **-12^oC**

Hence the degree of superheat at -23^oC = -12 – (-23) = 13 K (Ans.)

This example shows that when the same TEV operates at a lower evaporator temperature, then the required degree of superheat increases implying improper utilization of evaporator area. Hence, it is better to use cross-charging (power fluid is another fluid with a higher boiling point than refrigerant).