An Investigation of Compressor Slugging Problems

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Abstract

Mathematical models have been popularly applied to study the compression processes of superheated refrigerant vapor in compressors. As to the case when refrigerant enters a compressor cylinder in the state of liquid and vapor mixture, however, the availability of fundamental studies is very limited. A mathematical model is presented to simulate the compression processes of two-phase saturated refrigerant mixture. Factors leading to slugging are analyzed. It is explained why certain types of compressors are less prone to slug.

Introduction

Under normal operation of compressors, refrigerant is drawn into compressor cylinders in the state of super heated vapor. However, liquid refrigerant, or a portion of liquid refrigerant, can enter compressor cylinders under certain circumstances, such as during the cold start of a compressor. In some of the scroll compressor applications [1-3], refrigerant liquid is even deliberately injected into the gas pockets to decrease the discharge gas temperature under high compression ratios and thereby to extend the operating range of such compressors.

Liquid refrigerant entering compressor cylinders has been a concern for a long time. However, not enough has been achieved from the view point of mathematical modeling. An extensive literature review was done in this area, but no satisfactory thermodynamic model, which could be used to simulate the compressor cylinder processes, was found. Simpson and Lis [4] measured cylinder pressures during compressor liquid ingestion in reciprocating compressors and presented an empirical relation for predicting the maximum cylinder pressure in terms of compressor power, bore size, and stroke length. Singh, Nieter, and Prater [5-6] applied the polytropic equation to simulate compressor cylinder pressure, but the polytropic index in their model is considered as a variable and can be chosen arbitrarily. It is necessary to develop a mathematical model to predict the cylinder refrigerant thermodynamic properties instantaneously and to study the effects of liquid entering the cylinder on the compressor performance by simulating the thermodynamic process of compressing the liquid and vapor mixture.

Slugging problems are detrimental to compressors. Under extreme conditions, the compressor cylinder pressures could be as high as ten times the peak pressures under normal operations [5], due to the incompressibility of liquid, consequently damaging the compressor valves and reducing the reliability and lifetime of compressors. These effects should always be avoided, if possible. Does the liquid and vapor mixture entering compressor cylinders necessarily lead to a slugging problem? It depends on the quality (the ratio of vapor mass to the total mass of the liquid and vapor mixture) of the refrigerant entering the compressor cylinders, the kinematics of the compressor, and whether the compressor is a high side design or a low side design. For compressors with low side design, since the compressor shell is filled with cool suction gas and the cylinder temperature is not very hot, a slugging problem is likely to happen because of the lack of heat transfer from the cylinder to the liquid-vapor mixture, especially during the cold start. On the other hand, the cylinder of the compressor in a high side design is heated by the relatively high temperature discharge gas, therefore the possibility of liquid slugging is reduced. Common sense tells us that slugging is more likely to occur in compressors when refrigerant of low quality flows into the cylinders. However, the influence of the kinematics of a compressor on the possibility of slugging is not so obvious, and has to be studied through mathematical modeling.

Thermodynamic Model

In order to derive the thermodynamic equations, the following assumptions are made [7]: (1) the properties of the refrigerant are uniform throughout the cylinder control volume at any moment, (2) gravitational and kinetic energies of the refrigerant are neglected, and (3) the energy of the refrigerant in the control volume can be approximated by the internal energy since the average velocities are small. The first assumption implies that the liquid and vapor refrigerant is a homogeneous mixture.

Applying the principles of conservation of energy and mass to the control volume of the saturated two-phase mixture inside a compressor cylinder yields

$$\frac{dU}{d\alpha} = \frac{dQ}{d\alpha} + \frac{dW_{net}}{d\alpha} + h_i \frac{dm_i}{d\alpha} - h_o \frac{dm_o}{d\alpha} \tag{1}$$

$$\frac{dm_c}{d\alpha} = \frac{dm_i}{d\alpha} - \frac{dm_o}{d\alpha} \tag{2}$$

where U is the total internal energy of the refrigerant, Q is the heat transfer from the cylinder wall to the refrigerant in the control volume, W_{net} is the work done to the refrigerant in the control volume, h_i is the specific enthalpy of the refrigerant entering the control volume, m_i is the mass entering the control volume, h_o is the specific enthalpy of the refrigerant exiting the control volume, m_o is the mass exiting the control volume, m_c is the refrigerant mass inside the control volume, and α is the crank angle.

The refrigerant in the control volume has several properties such as pressure, temperature, density, enthalpy, internal energy and so on, but only two of those properties are independent. For instance, the state of the refrigerant is defined whenever the temperature and the quality of the refrigerant mixture are known. Therefore, the above two differential equations can be used to solve for the two independent variables. The next question is how to solve the two independent variables from the above two equations.

For saturated refrigerant of quality x and temperature T, some of the other properties can be expressed as

$$p = f(T) \tag{3}$$

$$v = x v_g + (1-x) v_l \tag{4}$$

$$h = x h_g + (1-x) h_l \tag{5}$$

where p is pressure, v is specific volume, h is specific enthalpy, the variables with subscript g represent the properties of the saturated vapor, and those with subscript l are the properties of saturated liquid. All those subscripted variables are only functions of temperature.

The derivative of the specific internal energy with respect to rotation angle can be expressed as

$$\frac{du}{d\alpha} = \frac{d}{d\alpha}(h - pv) = \frac{dh}{d\alpha} - p\frac{dv}{d\alpha} - v\frac{dp}{d\alpha} \tag{6}$$

where

$$\frac{dh}{d\alpha} = \left(\frac{\partial h}{\partial T}\right)_x \frac{dT}{d\alpha} + \left(\frac{\partial h}{\partial x}\right)_T \frac{dx}{d\alpha} = \left[x\frac{dh_g}{dT} + (1-x)\frac{dh_l}{dT}\right]\frac{dT}{d\alpha} + \left(h_g - h_l\right)\frac{dx}{d\alpha} \tag{7}$$

$$\frac{dv}{d\alpha} = \frac{d}{d\alpha}\left(\frac{V}{m_c}\right) = \frac{1}{m_c}\frac{dV}{d\alpha} - \frac{V}{m_c^2}\frac{dm_c}{d\alpha} \tag{8}$$

$$\frac{dp}{d\alpha}=\frac{dp}{dT}\frac{dT}{d\alpha} \tag{9}$$

therefore, substituting equations (7-9) into equation (6) yields

$$\frac{du}{d\alpha}=\left[x\frac{dh_g}{dT}+(1-x)\frac{dh_l}{dT}-v\frac{dp}{dT}\right]\frac{dT}{d\alpha}+\left(h_g-h_l\right)\frac{dx}{d\alpha}-\frac{p}{m_c}\frac{dV}{d\alpha}+\frac{pV}{m_c^2}\frac{dm_c}{d\alpha} \tag{10}$$

where V is the instantaneous cylinder volume, a function of crank angle α.

The specific volume v is defined as the ratio of total volume to mass, and can also be expressed as in equation (4). Thus

$$xv_g+(1-x)v_l=\frac{V}{m_c} \tag{11}$$

Differentiating both sides of the above equation, we obtain

$$v_g\,dx+x\left(\frac{dv_g}{dT}\right)dT-v_l\,dx+(1-x)\left(\frac{dv_l}{dT}\right)dT=\frac{m_c\,dV-V\,dm_c}{m_c^2} \tag{12}$$

dx can be solved from the above equation as

$$dx=\frac{1}{v_g-v_l}\left[\frac{m_c\,dV-V\,dm_c}{m_c^2}-x\left(\frac{dv_g}{dT}\right)dT-(1-x)\left(\frac{dv_l}{dT}\right)dT\right] \tag{13}$$

or

$$\frac{dx}{d\alpha}=\frac{1}{v_g-v_l}\left\{\frac{1}{m_c}\frac{dV}{d\alpha}-\frac{V}{m_c^2}\frac{dm_c}{d\alpha}-\left[x\left(\frac{dv_g}{dT}\right)+(1-x)\left(\frac{dv_l}{dT}\right)\right]\frac{dT}{d\alpha}\right\} \tag{14}$$

Substituting the above equation into equation (10) yields

$$\frac{du}{d\alpha}=\left[x\left(\frac{dh_g}{dT}-\frac{h_g-h_l}{v_g-v_l}\frac{dv_g}{dT}\right)+(1-x)\left(\frac{dh_l}{dT}-\frac{h_g-h_l}{v_g-v_l}\frac{dv_l}{dT}\right)-v\frac{dp}{dT}\right]\frac{dT}{d\alpha}+\left(\frac{h_g-h_l}{v_g-v_l}-p\right)\frac{1}{m_c}\frac{dV}{d\alpha}+\frac{V}{m_c^2}\left(p-\frac{h_g-h_l}{v_g-v_l}\right)\frac{dm_c}{d\alpha} \tag{15}$$

The derivative of the total internal energy with respect to rotation angle is

$$
\begin{aligned}
\frac{dU}{d\alpha} &= m_c \frac{du}{d\alpha} + u\frac{dm_c}{d\alpha} \\
&= m_c\left[x\left(\frac{dh_g}{dT} - \frac{h_g - h_l}{v_g - v_l}\frac{dv_g}{dT}\right) + (1-x)\left(\frac{dh_l}{dT} - \frac{h_g - h_l}{v_g - v_l}\frac{dv_l}{dT}\right) - v\frac{dp}{dT}\right]\frac{dT}{d\alpha} + \\
&\quad \left(\frac{h_g - h_l}{v_g - v_l} - p\right)\frac{dV}{d\alpha} + \frac{V}{m_c}\left(\frac{h}{v} - \frac{h_g - h_l}{v_g - v_l}\right)\frac{dm_c}{d\alpha}
\end{aligned}
\tag{16}
$$

Substituting this into the energy balance equation (1) and solving for $\frac{dT}{d\alpha}$ give

$$
\frac{dT}{d\alpha} = \frac{1}{a}\left\{\frac{dQ}{d\alpha} - \frac{h_g - h_l}{v_g - v_l}\frac{dV}{d\alpha} + \left(v\frac{h_g - h_l}{v_g - v_l} - h\right)\frac{dm_c}{d\alpha} + h_i\frac{dm_i}{d\alpha} - h_o\frac{dm_o}{d\alpha}\right\}
\tag{17}
$$

where

$$
a = m_c\left[x\left(\frac{dh_g}{dT} - \frac{h_g - h_l}{v_g - v_l}\frac{dv_g}{dT}\right) + (1-x)\left(\frac{dh_l}{dT} - \frac{h_g - h_l}{v_g - v_l}\frac{dv_l}{dT}\right) - v\frac{dp}{dT}\right]
\tag{18}
$$

and

$$
x = \frac{v - v_l}{v_g - v_l} = \frac{\frac{V}{m_c} - v_l}{v_g - v_l}
\tag{19}
$$

Equations (2) and (17) can be integrated to determine the temperature and the mass of the refrigerant inside the cylinder control volume. All the other terms in those two equations either are known or can be determined whenever T and m_c are obtained.

Computer Simulation

As mentioned earlier, all the variables with subscripts g or l are only functions of temperature. In order to numerically solve the highly nonlinear differential equations (2) and (17), we need to know those functions. For different kinds of refrigerants, they are different functions of temperature. The following relations are obtained through numerical interpolation of the published data from ASHRAE [8] for saturated liquid and saturated vapor of R22, which are valid for refrigerant R22 in the state of saturation from T=-45 F to T=195 F:

$$
z = c_0 + c_1 T + c_2 T^2 + c_3 T^3 + c_4 T^4 + c_5 T^5
\tag{20}
$$

where z stands for each of those mentioned properties. The interpolations corresponding to each property are listed in the Appendix. Similarly, the derivatives of those properties with respect to temperature, such as $\frac{dp}{dT}$, $\frac{dv_g}{dT}$, $\frac{dv_l}{dT}$, $\frac{dh_g}{dT}$, and $\frac{dh_l}{dT}$, can also be obtained in the form of equation (20).

After the two-phase refrigerant mixture enters the compressor cylinder, it may or may not still exist in the same state as liquid and vapor mixture. This depends on the cylinder processes and the suction condition. The other possibilities are: the refrigerant changes from the two-phase state into superheated gas during the compression process, or it is compressed into subcooled liquid. When the first case occurs, the thermodynamic model derived in this paper should be implemented together with the thermodynamic model for superheated vapor, as described in Reference [9], to simulate the properties of the refrigerant. However, when the refrigerant becomes pure liquid, there is no thermodynamic model to predict the refrigerant properties. Under liquid compression, the cylinder pressure can reach a very high value. This condition is called slugging.

Numerical Results and Discussions

The thermodynamic model is applied to simulate a variable speed rolling piston rotary compressor, with a speed range from 1100 rpm to 9000 rpm, which is depicted in Figure 1. The effect of increasing the liquid portion in the refrigerant mixture can be seen from the plots of Figure 2 - 5, which are computed under the same conditions p_s = 64.5 psia, p_d = 242.3 psia, and Ω = 7200 rpm. Those plots show the instantaneous values of the cylinder pressure, temperature, refrigerant mass, and the quality, during suction, compression, and discharge processes. As the quality of the refrigerant entering the cylinder decreases but does not cause the refrigerant to change to pure liquid during the cycle, the cylinder pressure does not change dramatically except during the discharge process due to the fact that the denser refrigerant needs more pressure to be pushed out of the cylinder. Reducing the quality in the suction side increases the capacity of the compressor since more refrigerant is drawn into the cylinder, as shown in Figure 4, and decreases the cylinder gas temperature, as shown in Figure 3.

The quality of the refrigerant entering the cylinder also affects the state of the refrigerant during the compression process. When the suction refrigerant quality is large, the cylinder refrigerant mixture becomes superheated vapor sometime during the process, as represented by the dash line in Figure 5. However, the opposite can also happen when a larger portion of saturated liquid flows into the compressor, as shown in Figure 6. The latter case is what should be prevented, since it is destructive to the compressor due to the incurred catastrophic cylinder pressure build up.

The fact that the saturated liquid and vapor two phase mixture can be compressed to pure liquid seems to be in contradiction with our intuition. But this can be proven from the pressure - enthalpy diagram, as shown in Figure 7. As the two phase liquid and vapor mixture is compressed along the isentropic line AB of Figure 7, its value of quality becomes smaller and smaller, and finally the two phase mixture changes to pure liquid. From the pressure - enthalpy diagram, whether the quality of the saturated refrigerant mixture increases or decreases depends on the slopes of both the constant quality lines and the constant entropy lines. If the slopes of the constant entropy lines are steeper than the slopes of the constant quality lines, the two phase mixture can be compressed to pure liquid, which occurs to the left side of the pressure - enthalpy diagram. However, when the slopes of the constant quality lines are steeper than those of the constant entropy lines, which is typical to the right side of the pressure - enthalpy diagram, the vapor and liquid saturated mixture can be compressed to pure vapor. From this point of view, the starting conditions of the compression process determines whether the quality is to increase or to decrease. If the compression is operated in the region that the slopes of the constant entropy lines are greater that those of the constant quality, slugging is very likely to happen to the compressor.

When liquid enters the compressor cylinder, usually the big concern is whether it leads to a slugging problem. After simulating the cylinder pressure under many different conditions, it is found that the cylinder pressure will most likely not reach a destructive level if the quality of the refrigerant is greater than zero at any moment of the cycle. In other words, the refrigerant is compressible as long as it is not pure liquid.

Factors Related to Slugging

The value of the quality is a very important factor since it indicates whether slugging happens or not during the cylinder compression process. A natural question arises: which factors cause the quality change? This question can be answered by studying equation (14). For convenience, to investigate the quality variation during the compression and discharge processes, it is re-listed:

$$\frac{dx}{d\alpha}=\frac{1}{v_g-v_l}\left\{\frac{1}{m_c}\frac{dV}{d\alpha}-\frac{V}{m_c^2}\frac{dm_c}{d\alpha}-\left[x\left(\frac{dv_g}{dT}\right)+(1-x)\left(\frac{dv_l}{dT}\right)\right]\frac{dT}{d\alpha}\right\}$$

In the above equation, $\frac{1}{v_g - v_l}$ is greater than zero, $\frac{dV}{d\alpha}$ is negative during both the compression and the discharge process, $\frac{dm_c}{d\alpha}$ is zero during the compression process and negative during the discharge process, $\frac{dT}{d\alpha}$ is always positive during the compression process since the pressure is increased, $\frac{dv_g}{dT}$ is negative, and $\frac{dv_l}{dT}$ is negligible comparing with $\frac{dv_g}{dT}$ since saturated liquid does not change its specific volume with respect to temperature very much. Therefore, $\frac{dV}{d\alpha}$ is the only term in the above equation that causes the quality to decrease.

To avoid an increase of the liquid portion in the mixture, it is helpful either to increase $\frac{dT}{d\alpha}$ through transferring heat into the cylinder refrigerant or reducing the gradient of the cylinder volume $\frac{dV}{d\alpha}$ by utilizing a gradual compression mechanism. This means that, everything else being equal, reciprocating compressors have theoretically more of a tendency to slug than, say, scroll compressors because of the difference in $\frac{dV}{d\alpha}$.

Conclusions

Destructive cylinder pressure associated with liquid slugging can only occur when the refrigerant in the cylinder is purely liquid. This could happen under the following cases: (1) refrigerant pumped into compressor cylinders is subcooled liquid, and (2) refrigerant pumped into compressor cylinders is two-phase mixture, but it transforms into pure liquid during the compression process. The first case normally happens during the cold start of a compressor. The second case is influenced by several factors such as the initial quality of the refrigerant entering the compressor cylinder, the heat transfer between refrigerant and cylinders, and the kinematics of the compressor. In this simulation, liquid slugging is detected by checking whether the quality is greater than zero or not.

Among reciprocating, rolling piston rotary, and scroll compressors, reciprocating compressors have the largest volume compression gradient, and scroll ones have the smallest, since the compression and discharge processes of those three kinds of compressors are respectively 180 degrees, 360 degrees, and 540 degrees, of rotation. Therefore, everything else assumed being equal, slugging problems will more likely happen in reciprocating compressors. This also explains why the liquid injection techniques are mostly applied to scroll compressors, since this device is less sensitive.

Acknowledgment

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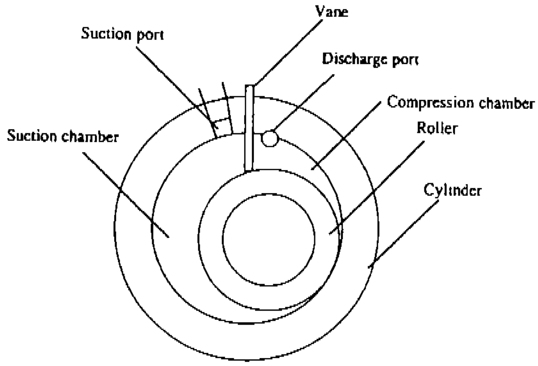

Figure 1 A rolling piston rotary compressor mechanism

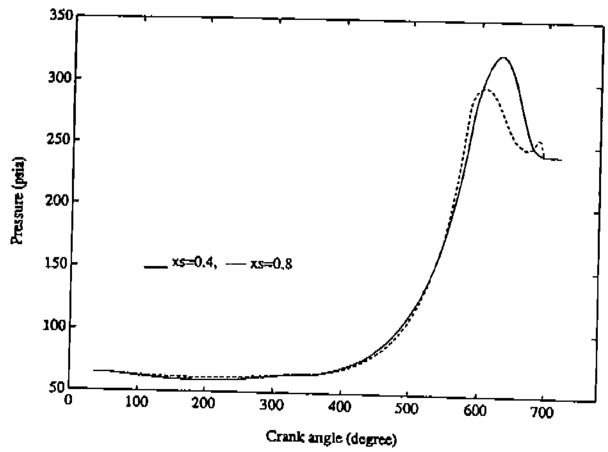

Figure 2 Refrigerant pressure during cylinder processes (p_s = 64.5 psia, p_d = 242.3 psia, and Ω = 7200 rpm.)

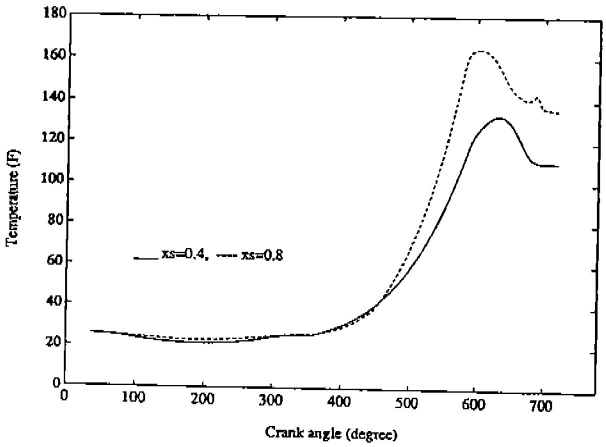

Figure 3 Refrigerant temperature during cylinder processes (p_s = 64.5 psia, p_d = 242.3 psia, and Ω = 7200 rpm.)

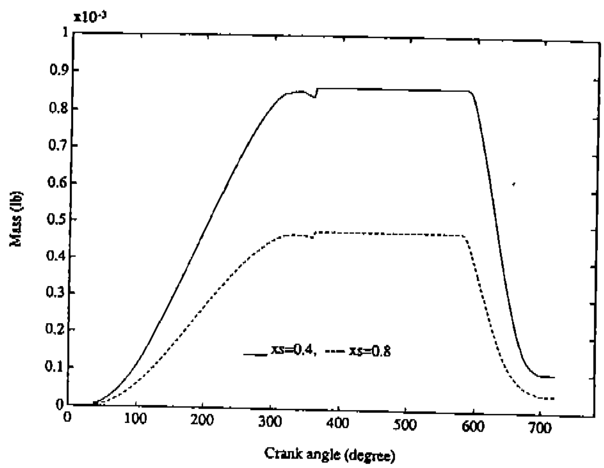

Figure 4 Refrigerant mass during cylinder processes (p_s = 64.5 psia, p_d = 242.3 psia, and W = 7200 rpm.)

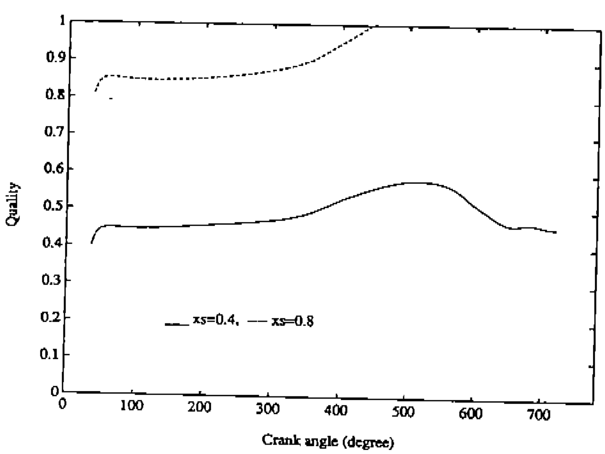

Figure 5 Refrigerant quality during cylinder processes (p_s = 64.5 psia, p_d = 242.3 psia, and Ω = 7200 rpm.)

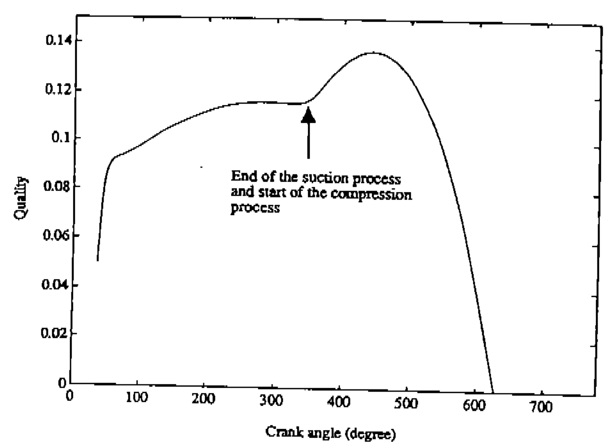

Figure 6 Quality of cylinder refrigerant when slugging occurs (x_s=0.05, p_s = 64.5 psia, p_d = 242.3 psia, and Ω = 7200 rpm.)

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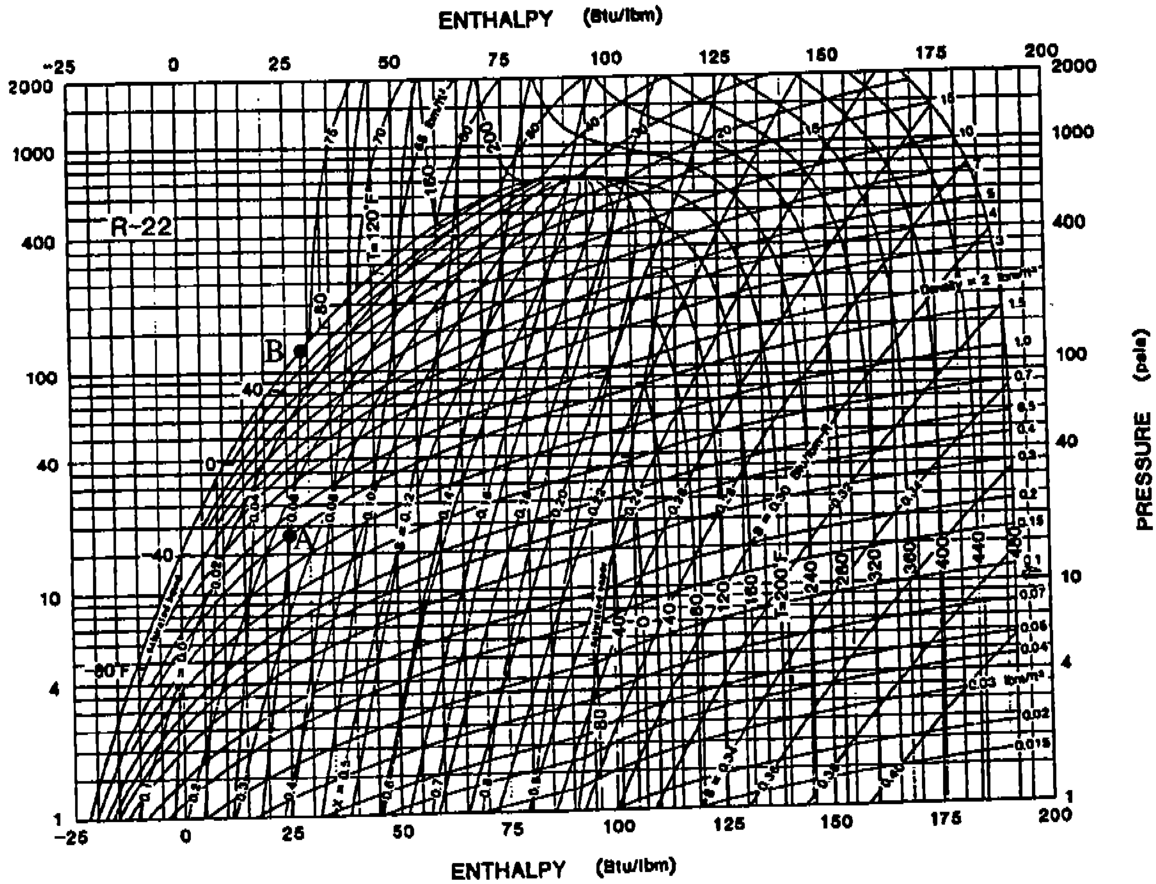

Figure 7 Illustration of the isentropic compression process of saturated refrigerant two phase mixture on the pressure - enthalpy diagram of R22 [8]