1996

A Study on Compression Characteristic of Wet Vapor Refrigerant

A. K. Dutta  
*Shizuoka University*

T. Yanagisawa  
*Shizuoka University*

M. Fukuta  
*Shizuoka University*

Follow this and additional works at: [http://docs.lib.purdue.edu/icec](http://docs.lib.purdue.edu/icec)

[http://docs.lib.purdue.edu/icec/1112](http://docs.lib.purdue.edu/icec/1112)
A STUDY ON COMPRESSION CHARACTERISTIC OF WET VAPOUR REFRIGERANT

Asit Kumar DUTTA, Tadashi YANAGISAWA, and Mitsuhiro FUKUTA
Department of Energy and Mechanical Engineering, Shizuoka University
3-5-1, Johoku, Hamamatsu, 432, JAPAN

ABSTRACT

In this study a compression characteristic of vapor-liquid two-phase refrigerant mixture is analyzed theoretically by using the mathematical models of vapor-liquid mixture condition in which the droplet size of the liquid, homogeneous mixture of vapor and liquid, and the slugging concept of the liquid are considered. Influences of the parameters, such as quality of wet vapor and the cylinder wall heat transfer, on the pressure as well as on the temperature are investigated. A model experiment has been done by using the components of the reciprocating compressor without connecting to the refrigeration cycle. Different condition of the liquid refrigerant such as liquid slug has been created by injecting liquid refrigerant into the cylinder prior to the compression start, and the first compression process has been considered for investigation. In this case, the result of the model considering slugging concept matches well with the experimental result.

Further, a refrigeration cycle experiment has been done by using a reciprocating compressor under liquid suctioning, and the cylinder pressure change at steady state operation with different suction quality are recorded for investigation. In this case, the result of the model considering homogeneous vapor-liquid mixture or the smallest drop size (1 μm) of liquid matches well with the experimental result.

1. INTRODUCTION

In the refrigeration fields, wet compression is a common practice to decrease the discharge gas temperature. Beside this, several researches have been done concerning with slugging [1,2], leakage and the performances of the compressor during wet compression [3,4,5]. However, it remains the question of how the compression characteristics behave under wet compression.

In this study, we have analyzed the compression pressure characteristic, temperature and pressure change under wet compression by considering three different models of vapor and liquid mixture inside the compressor cylinder. Two different experiments have been done to check the validity of these models i.e. refrigeration cycle experiment with liquid suctioning, and the model experiment by injecting liquid refrigerant inside the compressor cylinder without connecting to the refrigeration cycle.

2. ANALYTICAL MODEL

The model is presented schematically in Figure 1. In this figure, the compressor cylinder of volume \( V \) is considered as a control volume. The wet vapor refrigerant enters into the control volume and within the control volume owing to the small increment of piston as well as the heat from the outside, the temperature and pressure of the vapor increase and exchange heat to the existing liquid refrigerant. The process will be repeated throughout the whole compression period. Finally the refrigerant leaves the control volume.

2.1 Governing Equation

From the physical laws of mass and energy conservation, the basic energy equations for vapor and liquid refrigerant are expressed as:

\[
m_g \frac{dh_g}{dt} = G_{g} (h_{g} - h_{g0}) + G_{g0} (h_{g0} - h_{go}) + Q_{wg} + m_g \rho_g \frac{dP}{dt} - Q
\]  

(1)
and volume equation for control volume is:

\[ \frac{dV}{dt} = \frac{dV}{dt} = G_v (v_v - v_t) + m_v \frac{dv_v}{dt} + v_v (G_{gh} - G_{go}) + v_t (G_{gh} - G_{go}) \]  

Where, \( G \): refrigerant mass flow rate, \( v \) and \( h \): specific volume and enthalpy, \( m \): mass of refrigerant inside the control volume, \( P \): pressure, \( Q_{wg} \) and \( Q_{w} \): heat flow from the wall to the vapor and liquid, \( P \): pressure, \( Q \): heat flow from the vapor to liquid. The subscripts \( e \): evaporation, \( g \) and \( l \): vapor and liquid, \( g_1, g_0 \) and \( l_1, l_0 \): incoming and outgoing vapor and liquid.

The properties of the refrigerant vapor are represented as:

\[ v_g = f_g(T_g, P) \]  
\[ h_g = f_g(T_g, P) \]  

and the refrigerant liquid are represented as:

\[ v_l = f_l(T_l) \]  
\[ h_l = f_l(T_l) \]  

Therefore, the main unknown variables in the above equations are: \( T_g, T_l \), \( P \), \( G_v \) and \( Q \), provided the incoming and outgoing mass flow rate and the heat flow from the wall to the control volume are known. Therefore, in order to solve those unknown variables, other two equations become necessary except the above stated three equations, and to do so, proper modeling of vapor-liquid mixture should be needed. In this study, the following models are considered: (1) Droplet model, (2) Homogeneous model and (3) Slugging model.

(1) **Droplet Model:**

In this model it is assumed that, the vapor and liquid refrigerant within the control volume exist separately and have different temperatures. The heat flow \( (Q) \) from the vapor to the liquid inside the control volume can be calculated by considering the size of liquid drop, heat transfer coefficient \[6\] and temperature difference between liquid and vapor inside the control volume.

The main unknown variables are: \( T_g, T_l, P \) and \( G_v \). The step-by-step solution procedure of the equations are obtained as follows: first to solve the equations without considering \( G_v \), and compare \( T_l \) with the saturation temperature corresponding to the cylinder pressure.

(2) **Homogeneous Model:**

In this model, it is assumed that, the vapor and liquid refrigerant inside the control volume possess the same temperature at any moment. Then, the energy equations \( (1) \) and \( (2) \) for vapor and liquid can be modified, and written together as follows:

\[ m_v \frac{dh_v}{dt} + m_l \frac{dh_l}{dt} = G_{gh} (h_g - h_g) + G_{go} (h_g - h_go) + G_{hi} (h_l - h_l) + G_{io} (h_l - h_io) - G_{sh} (h_g - h_g) + Q \]  

The main unknown variables are: \( T \) \( (T_g = T_l) \), \( P \) and \( G_v \). Under wet compression pressure \( P \) is the function of \( T \), and the other two variables are solved by the appropriate equations.

(3) **Slugging Model:**

In this model, it is assumed that, the liquid refrigerant in the control volume is the same temperature as initial at any moment, and the gas is always the saturation gas under wet compression. Therefore, volume equation for the control volume is same as \( (3) \) except \( v_i \) is replaced by \( v_{line} \), and the energy equations \( (1) \) and \( (2) \) can be modified, and written together as follows:

\[ m_v \frac{dh_v}{dt} + m_l \frac{dh_l}{dt} = G_{gh} (h_g - h_g) + G_{go} (h_g - h_go) + G_{hi} (h_l - h_l) + G_{io} (h_l - h_io) - G_{sh} (h_g - h_g) + Q \frac{VdP}{dt} \]  

(4)
\[ \frac{d(h_g)}{dt} = G_{s}(h_{g} - h_{g}) + G_{w}(h_{g} - h_{g}) + G_{i}(h_{iw} - h_{iw}) + G_{l}(h_{iw} - h_{l}) + Q_{w} + \frac{VdP}{dt} \]  

(5)

Where, \( h_{iw} \) : specific enthalpy of the liquid refrigerant at the initial condition.

The main unknown variables are: \( T_{g}, P \) and \( G_{e} \), and under wet compression pressure \( P \) is the function of temperature. Therefore, the remaining variables are solved easily by the appropriate equations.

The heat transfer \( (Q_{w}) \) from the wall surface to the control volume can be expressed as follows:

\[ Q_{w} = A_{w} \alpha (T_{w} - T) \]  

(6)

where, \( A_{w} \) : heat transfer area, \( \alpha \) : convective heat transfer coefficient which can be obtained from the Adair correlation [7] based on the reciprocating compressor. The property values which are used in that correlation, are considered in terms of the vapor quality within the saturated as well as the superheated zone corresponding to the average cylinder pressure.

To simulate the compression process, the incoming and outgoing flow rate are calculated considering flow through the valve ports as the nozzle flow [8]. The properties of the refrigerant HCFC22 can be obtained from the correlation equations [9]. The property subroutine was implemented in the simulation programs.

3. EXPERIMENTAL METHODS

3.1 Cycle Experiment

Refrigeration cycle is shown in Figure 2. An open type reciprocating compressor (stroke volume 56.3 cm³) which is generally used in the automobile refrigeration, was connected to the refrigeration cycle. In order to control the suction quality, a bypass line was connected to the compressor suction line from the condenser outlet. The main portion of the condensed liquid refrigerant was passed through the evaporator, and the bypass portion was added at the suction line through a control valve. Pressure in the cylinder and the plenums of the compressor were measured by the piezo-electric and the strain gauge type pressure transducer respectively. The compressor was driven by an electric motor, and the compressor speed along with the rotation angle was measured by an optical sensor. Temperature at different points of the cycle were measured by the C-C thermocouples. Refrigerant flow rates through the evaporator and the bypass were measured by a rotameter and an oval type flow meter respectively. Refrigerant HCFC22 was used as a working fluid, and the cycle was operated under suction and discharge absolute pressure of 0.59 MPa and 1.57 MPa respectively.

Data were recorded at steady state operation, with different suction quality and compressor speed. Further, the vapor quality at the suction port was obtained by revising the vapor quality at mixing point since refrigerant takes
heat from the plenum wall.

3.2 Model Experiment of Wet Compression with Liquid Slug

Model experiment has been done by using the component of the reciprocating compressor without connecting to the refrigeration cycle as shown in Figure 3. The upper part of the compressor, such as cylinder head, suction and discharge ports, valves etc., have been replaced by a metallic cylindrical cap, in which a sight glass, charging port, pressure gauge and a thermocouple were installed (the volume ratio of the top dead and the bottom dead center of the piston=1.8). To prevent the leakage through the piston ring, a groove has been created around the top surface of the piston and a rubber o-ring was installed. The compressor was driven by an electric motor, and the ON-OFF operation of the compressor was controlled by the clutch attached to the compressor pulley. The compressor speed along with rotation angle was measured by the magnetic sensor. Cylinder pressure was measured by the strain gauge pressure transducer and the output signal of that transducer was recorded by the computer. Liquid refrigerant injected to the compressor cylinder by the charging vessel and the charged mass of refrigerant was calculated as follows: the weight of the charging vessel was measured by the electronic measuring equipment before and after the charge. In the charging condition, the liquid slug was clearly found inside the cylinder through the sight glass fitted at the top of the cylinder. Compressor was started when the cylinder pressure reached to the saturation pressure corresponding to the room temperature.

Experiment was done by varying the liquid refrigerant quality with different speed, and only the first compression pressure change was considered for investigation. Refrigerant HCFC22 was used as a working fluid for this experiment. Furthermore, the experiment was also done with superheated refrigerant vapor, and recorded the cylinder pressure change by the computer as same as before.

4. RESULTS AND DISCUSSIONS

4.1 Refrigeration Cycle Experiment

Figure 4 shows the suction and discharge plenum pressure and cylinder pressure change at each suction quality, and it indicates that cylinder suction pressure is affected by the suction plenum pressure and valve losses. Therefore, to investigate the compression characteristic at different suction quality, the cylinder pressure change at each suction quality is normalized with the pressure at crank angle 180 degree and the compression and discharge pressure change are shown in the Figure 5. Further, the pressure change calculated by considering the homogeneous and slugging models with same experimental condition are shown in the bottom two of the Figure 5. In the simulations, the measured discharge temperature at each suction quality is used as a wall temperature. By comparing model results with experimental one, it is clear that, the calculated results considering homogeneous model behave in the same way with the experimental results, i.e. at the latter half of the compression (crank angle 230~285 degree), the pressure rise become slower as compared with superheated vapor pressure rise and decreases with decreasing suction quality due
to the cooling effect of refrigerant liquid. While, at the beginning of the compression (crank angle 180°–230° degree) the pressure rise of the wet vapor at each suction quality becomes slightly earlier (though it is not clear in the figure) as compared with the superheated vapor pressure rise since the temperature difference between wall and refrigerant inside the cylinder enhance the evaporation of liquid. Further, the results of the slugging model shows that the pressure rise of the wet vapor becomes earlier with decreasing quality. The results of droplet model are not shown here since with smallest drop size the results of the droplet model are almost same with the homogeneous model, while at largest drop size the heat transfer between vapor and liquid decreases which causes to increase the pressure rise as compared with the superheated vapor.

Depending on the same behavior of the homogeneous model and experimental results, it can be said that, during liquid suctioning at steady state operation, the liquid enters into the cylinder as a fine drop and perfectly exchanges heat with the vapor inside the cylinder.

Further, the figure 6 shows the temperature and quality change of the refrigerant vapor with crank angle considering homogeneous model. The figure indicates that, the temperature of the vapor decreases in the process of compression due to the evaporation of the refrigerant liquid. The quality curve shows that at 86% quality the liquid refrigerant is evaporated in the process of compression, while at 60% quality, the condensation occurs in the process of compression since the liquid cool the vapor up to its saturation temperature corresponding to the vapor pressure, and further cooling is the cause of condensation.

Finally, the Figure 7 is a plot of calculated and experimental discharge vapor temperature, and it shows that, discharge temperature decreases with decreasing the suction vapor quality and below 90% quality, the discharge temperature becomes equal to the saturation temperature corresponding to the discharge pressure. Figure 7 further indicates that, the calculated result considering heat transfer from wall to refrigerant inside the cylinder matches well with the experimental result.

4.2 Model Experiment of Wet Compression with Liquid Slug

The model experimental results of variation of cylinder pressure with different starting quality are shown in the top of the Figure 8, and the calculated results from different models are shown in the bottom three of the same figure. Compared with experimental results with model results it can be seen that, the trend in pressure rise of slugging model at each starting quality behave the same as the experimental results, i.e. at 90% quality of the wet vapor, the trend in compression pressure rise becomes slower as compared with the superheated vapor (103%) pressure rise. While at 61% quality, the pressure rise becomes earlier and is almost the same as the superheated vapor pressure rise, because at low quality the volume of liquid refrigerant enhance the compression on the vapor. On the other hand, the trend in pressure rise of the homogeneous model at each starting quality becomes slower with decreasing quality due to the cooling effect of the refrigerant liquid. The results of the droplet model shows that at constant quality the compression pressure rise decreases with decreasing drop size since as the drop size decreases the surface area of the drop increases. Further, at smallest drop size (1 μm) the result of the droplet model is almost same with the homogeneous model.
5. CONCLUSIONS

The compression pressure characteristic of the wet vapor refrigerant was investigated experimentally and mathematically, and was drawn the following conclusions:

1. The slugging condition created by the model experiment was analyzed by the slugging model, and the results indicated that under slugging condition the compression pressure rise became earlier with decrease in starting quality of vapor below 90%.

2. The cylinder process of the compressor under liquid suctioning at steady state operation, was analyzed by the homogeneous model or the droplet model of smallest drop size (1μm). The results indicated that the pressure rise of the wet vapor became earlier at the beginning of the compression as compared with the superheated vapor pressure rise. However at latter half of the compression it became slower with decreasing suction quality.

3. From the analytical results it became clear that, the evaporation of the liquid refrigerant in the compressor cylinder decreased not only the vapor temperature but also decreased the compression pressure rise under liquid suctioning. Further, as compared with the results excluding the heat transfer, it can be said that heat transfer during wet compression must not be avoided.

6. REFERENCES