Modeling Temperatures in High Speed Compressors for the Purpose of Gas Pulsation and Valve Loss Modelling

Z. Liu
Purdue University

W. Soedel
Purdue University

Modeling Temperatures in High Speed Compressors for the Purpose of Gas Pulsation and Valve Loss Modelling

Zheji Liu, Graduate Research Assistant
Werner Soedel, Professor of Mechanical Engineering
Ray W. Herrick Laboratories
School of Mechanical Engineering
Purdue University
West Lafayette, IN 47907

ABSTRACT

The effects of compressor speed variation on the heat transfer inside a speed controlled compressor were investigated in this study. Based on the lumped capacitance method, the temperatures of the compressor components, the oil, and the refrigerant gas within the shell were modeled and simulated on a relative scale. A different heat transfer coefficient correlation was developed to evaluate the convective heat transfer rate between the compressor cylinder wall and the gas inside it. The refrigerant gas mass loss at different compressor speeds and how the compressor speed affects the suction gas heating were discussed.

1. Introduction

Power is consumed in each compression cycle to increase the enthalpy of gas through the compression performed by the piston inside the compressor cylinder. Cool gas enters the cylinder, and then leaves the cylinder at a much higher temperature. Hot gas circulates inside the hermetic shell and warms up other components in the shell. In addition, because of friction losses and motor losses, mechanical energy and electrical energy are converted into thermal energy, which can be considered as heat sources supplying energy to compressor parts. According to the heat transfer modes, the overall heat transfer inside the compressor system can be classified as: convection due to gas and oil circulation, conduction between the connected components, and radiation from internal components to the shell and from the shell to the surroundings.

Of the overall heat exchange of the compressor system, the heat transfer of most concern happens in the suction side of the compressor. From the moment when gas enters the suction pipe until the gas is trapped in the cylinder volume and compression is about to start, the gas continuously picks up heat from the suction passage and the cylinder wall, which is called suction gas heating. Because of suction gas heating, the gas expands and its density is decreased. Since the volume of the suction chamber is fixed, the lighter gas means less mass is pumped, and, therefore, volumetric losses occur without reducing the amount of energy required for compression.

The heat transfer rate between the compressor cylinder wall and the gas is one of the factors which affects the thermodynamic processes inside the cylinder. Therefore, the heat transfer between the cylinder wall and the gas during both the expansion and the compression process needs to be studied. The heat transferred between the cylinder wall and the gas has effects on the compressor power consumption and discharge temperature.

However, the study of the heat transfer in the compressor cylinder and the suction passage can not be isolated. The temperatures of the cylinder wall and the suction passage are needed to determine the heat flow rate in these regions. Since heat flows from one compressor component to another, the temperatures of these components are interactively related and have to be solved simultaneously. So, the overall heat transfer model is indispensable.

The compressor shaft rotation speed also affects the heat transfer both in the cylinder and in the suction passage. Intuitively, the faster compressor speed would mean less time for heat transferring to the gas from the suction passage and the cylinder wall, and therefore reducing the effects of suction gas heating. However, the increase of compressor speed also causes more power wasted into heat through friction and motor losses per unit time, and in the same time, causes larger convective heat transfer coefficients due to faster gas flow, so the heat transfer is elevated from this point of view. The net effect of the speed variation on heat transfer is very complicated and can only be determined by resorting to the computer simulation.

2. The Lumped Capacitance Method

The temperature distribution inside the compressor system is very complicated. Even in a single component, for example, the cylinder case, the temperature could be different from one place to another. It is beyond the scope of this research to study the spatial temperature distributions of the compressor components.

Based on the consideration that the spatial fluctuations of the temperatures are relatively small, comparing with the average temperatures of the solids, it is assumed that the temperature in each solid and oil is spatially uniform at any instant during the transient process. This is called the lumped capacitance method [1]. It implies that temperature gradients within the solid and the oil are negligible.

Using the lumped capacitance method, uniform temperatures are assigned to the compressor components, as shown in Figure 1, and are described as follows:

- T_{comp} = Bulk temperature of the compressor and the rotor, combining the cylinder case, the vane, the roller, the shaft, and the motor rotor as one lumped mass.
- T_{oil} = Bulk temperature of the oil.
- T_{shelld} = Bulk temperature of the lower part of the shell, or of the shell section below the oil level.
- T_{shellu} = Bulk temperature of the upper shell and the stator, combining the shell section above the oil level and the motor stator as one lumped mass.
- T_{pipe2} = Bulk temperature of the suction pipe segment exposed to the outside, located between the compressor shell and the accumulator.
- T_{accu} = Bulk temperature of the accumulator and the suction pipe segment inside the accumulator, which are considered as one lumped mass.

3. Heat Transfer in the Suction Passage

The suction passage of the rolling piston rotary compressor is actually a circular pipe, as shown in Figure 2. The convection heat transfer coefficient for turbulent flow in a uniformly circular pipe is [1]

$$h_p = 0.023 \frac{k}{D} Re_D^{0.8} Pr^{0.4} \tag{1}$$

where k is the thermal conductivity of the gas inside the pipe, D is the diameter of the pipe, Re_D is the Reynolds number of the gas, and Pr is the Prandtl number of the gas.

The average value of the Reynolds number Re_D for flow in the circular pipe is [1]

$$Re_D = \frac{4\,\dot{m}_{ave}}{\pi D \mu} \tag{2}$$

where $\dot{m}_{ave}$ is the average mass flow rate of the suction gas during the suction process, and μ is the dynamic viscosity of the gas.

If the whole suction pipe is divided into three segments as shown in Figure 2, then, according to Section 2, the temperatures of the three segments of the pipe are respectively T_{accu}, T_{pipe2}, and T_{comp}. Applying the principle of energy conservation to each of the gas control volume inside the three segments of the pipe yields

$$h_p \pi D L_1 \left(T_{accu} - \frac{T_s + T_{s1}}{2} \right) = \dot{m}_{ave}\, c_p\, (T_{s1} - T_s) \tag{3}$$

$$h_p \pi D L_2 \left(T_{pipe2} - \frac{T_{s1} + T_{s2}}{2} \right) = \dot{m}_{ave}\, c_p\, (T_{s2} - T_{s1}) \tag{4}$$

$$h_p \pi D L_3 \left(T_{comp} - \frac{T_{s3} + T_{s2}}{2} \right) = \dot{m}_{ave}\, c_p\, (T_{s3} - T_{s2}) \tag{5}$$

where L_1, L_2, and L_3 are the lengths of the three segments of the pipe, c_p is the constant pressure specific heat of the gas, T_s is the temperature of the gas entering the suction pipe, T_{s1} and T_{s2} are the down stream gas temperatures of the control volume 1 and the control volume 2, as shown in Figure 2, and T_{s3} is the temperature of the gas entering the cylinder volume.

The component temperature T_{accu}, T_{pipe2}, and T_{comp} will be calculated in Section 5. When they are known, the gas temperature T_{s1}, T_{s2}, and T_{s3} can be solved from Equations (3)-(5).

4. Heat Transfer Between the Cylinder and the Gas Within it

The instantaneous heat transfer between the compressor cylinder walls and the gas within the cylinder is due to the turbulent motion of the gas relative to the walls. The velocity of the gas consists of two components [2]. The first component is due to the motion of the compressor cylinder volumes relative to the cylinder walls. The second one is the squishing velocity caused by the cylinder volume geometry change. However, the squishing velocity is very difficult, if not impossible, to predict, because of the complex configuration of the cylinder volume and its variation from one type of compressor to another. Therefore, modeling the heat transfer inside the cylinder is much more difficult than that inside the suction pipe. Not like the heat transfer inside the pipe, there is no general and accurate formula or correlation for the convection heat transfer coefficient between the cylinder walls and the gas in the cylinder of a rolling piston rotary compressor.

A fair amount of work has been done on the prediction of the heat transfer between cylinder walls and the gas inside a cylinder. Adair [3] is the first one who investigated the instantaneous heat transfer between the cylinder and the gas within a reciprocating compressor. He presented his heat transfer coefficient correlation as follows

$$h_c = 0.053 \frac{k}{D_e} Re^{0.8} Pr^{0.6} \tag{6}$$

$$D_e = \frac{6 \text{ cylinder volume}}{\text{cylinder surface area}} \tag{7}$$

$$Re = \frac{\rho D_e \left(\frac{D_e}{2} \omega_g \right)}{\mu} \tag{8}$$

where ρ is the density of gas, ω_g is the characteristic angular velocity. However, Recktenwald's results [4] cast considerable doubt on the validity of the heat transfer correlation obtained by Adair. By using the so called multinode model, he found that the heat transfer rate between the cylinder walls and the gas within the cylinder of a reciprocating compressor is an order of magnitude larger than that predicted by the Adair correlation.

Yanagisawa [5] applied both the Adair correlation and the McAdams correlation to the heat transfer between the cylinder walls and the gas in the cylinder of a rolling piston rotary compressor. The McAdams correlation was first used by Atesmen [2] on the rotary combustion engine. It is expressed as

$$h_c = 0.023 \frac{k}{D_h} Re^{0.8} Pr^{0.4} \tag{9}$$

where D_h is the instantaneous hydraulic diameter of the combustion chamber. In the combustion chamber of the rotary engine, the fuel-air mixture velocity relative to the housing is more complicated than the gas velocity in the compressor cylinder. Besides the velocity component due to the motion of the chamber relative to the housing and the squishing velocity component caused by the changing chamber geometry, there is a third velocity component induced by the expanding combustion products in between the flame fronts. Therefore, the effective fuel-air mixture velocity Atesmen used in the McAdams correlation is three times the velocity of the chamber relative to the rotary combustion engine housing. Without any modification, Yanagisawa borrowed the above correlation for the rolling piston rotary compressor, although the gas velocity in the compressor cylinder does not consist of the component of flame front velocity as in the rotary combustion engine. He adopted

$$h_c = 0.023 \frac{k}{D_h} \left(\frac{u D_h}{\nu} \right)^{0.8} \left(\frac{\nu}{\kappa} \right)^{0.4} \tag{10}$$

$$D_h = r_1 = \text{cylinder radius} \tag{11}$$

$$u = 3\, \omega\, r_1 \tag{12}$$

where ν is the kinematic viscosity, κ is the thermal diffusivity, ω is the angular velocity of the compressor shaft, and u is the effective velocity of the gas.

The Adair correlation was developed for the reciprocating compressor whose geometry is quite different from that of the rolling piston rotary compressor. The McAdams correlation is primarily for the heat transfer between a pipe and the fluid within it, and does not reflect the effect of the curved geometry of the rolling piston rotary compressor cylinder volume on the heat transfer. Therefore, a different heat transfer coefficient correlation is used in this study. Since the crescent shaped volume inside the rolling piston rotary compressor is similar to a spiral tube with rectangular cross section, the convection heat transfer correlation for curved channels of rectangular cross section [6] is applied to the heat transfer inside the rolling piston rotary compressor cylinder. The correlation is

$$h_c = 0.025 \frac{k}{D_h} Re^{0.8} Pr^{0.4} \left(1.0 + 1.77 \frac{D_h}{r_{ave}} \right) \tag{13}$$

where r_{ave} is the average value of the cylinder radius r_1 and the roller radius r_2. As the compressor shaft rotates, the cylinder volume geometry is changing, so is the hydraulic diameter D_h. Since the physical properties of the gas, such as density, thermal conductivity, and dynamical viscosity, are all functions of the pressure and the temperature in the cylinder, the instantaneous value of Reynolds number Re and that of Prandtl number Pr are all changing from one location of the cylinder volume to another.

The gas velocity is a very important factor of the heat transfer coefficient. As mentioned in the beginning of this section, the velocity of the gas inside the rolling piston rotary compressor cylinder is made up of two components. As predicted by Atesmen, the relative magnitude of the velocity component due to the motion of the cylinder volume relative to the cylinder is of the same order as the squishing velocity component caused by the cylinder volume geometry change. So, It is assumed that the characteristic velocity of the gas is

$$u = 2\,\omega\, r_1 \tag{14}$$

The instantaneous value of the hydraulic diameter D_h of the crescent shaped cylinder volume is

$$D_h = \frac{4\,V(\alpha)}{A_c(\alpha)} \tag{15}$$

where the cylinder volume $V(\alpha)$ and the volume surface area $A_c(\alpha)$ can be calculated by using the geometrical relations of the compressor.

Using the above characteristic velocity and the hydraulic diameter, the Reynolds number is obtained as

$$Re = \frac{\rho\, u\, D_h}{\mu} \tag{16}$$

and the Prandtl number can be calculated by the relation

$$Pr = \frac{c_p\,\mu}{k} \tag{17}$$

Finally, the heat transfer rate from the cylinder walls to the gas inside the cylinder can be expressed as

$$\dot{Q} = h_c\, A_c(\alpha)\,(T_{comp} - T)$$

$$= 0.025\,\frac{k}{D_h}\,Re^{0.8}\,Pr^{0.4}\left(1.0 + 1.77\,\frac{D_h}{r_{ave}}\right)A_c(\alpha)\,(T_{comp} - T) \tag{18}$$

5. Governing Equations of the Compressor System Temperatures

The temperatures of some compressor components, such as the cylinder temperature, and the temperatures of the three segments of the suction pipe, are needed in determining the heat transfer both in the cylinder and in the suction passage. As mentioned in Section 1, all the compressor component temperatures must be solved simultaneously, because they are coupled to each other through the heat flowing between them.

Thus, the presented results are based on the lumped capacitance method introduced in Section 2 and the following assumptions:

(1) The heat transfer induced by oil circulation throughout the compressor system is neglected.
(2) The motor losses is considered as a heat source inside the stator, or the upper shell-stator lumped mass.
(3) Friction losses are considered as a heat source inside the compressor cylinder, or the compressor-rotor lumped mass.
(4) An average heat transfer coefficient is used for the overall convection heat transfer between the gas inside the shell and the compressor components.
(5) The properties of the gas inside the shell, such as temperature, pressure, etc., are uniform, as they are inside the cylinder.
(6) Ideal gas relations are applicable to the gas inside the shell.

The heat flow diagram [7] of the compressor system is shown in Figure 2. The bulk temperatures of the lumped mass were defined in Section 2, and the rest of the symbols used in Figure 1 are described below:

T_{amb} = Temperature of the ambient surrounding the compressor shell.
T_{gas} = Bulk temperature of the refrigerant gas inside the shell.
R_1 = Average thermal resistance between the oil and the lower part of the shell.
R_2 = Average thermal resistance between the oil and the compressor.
R_3 = Average thermal resistance between the lower shell and the compressor.
R_4 = Average thermal resistance between the upper shell and lower shell.
R_5 = Average thermal radiative resistance between the upper shell-stator lumped mass and the compressor-rotor lumped mass.
R_6 = Average thermal convective resistance between the gas inside the shell and the compressor-rotor lumped mass.

R_7 = Average thermal resistance between the oil and the upper shell-stator lumped mass due to the fact that part of the windings in the stator is immersed in the oil.
R_8 = Average thermal convective resistance between the gas inside the shell and the oil.
R_9 = Average thermal convective resistance between the gas inside the shell and the upper shell-stator lumped mass.
R_{10} = Average thermal resistance between the upper shell and the accumulator.
R_{11} = Average thermal resistance between the outside surface of the lower shell and the surrounding medium.
R_{12} = Average thermal resistance between the outside surface of the upper shell and the surrounding medium.
R_{13} = Average thermal resistance between the lower shell and the suction pipe segment exposed to the outside, located between the compressor shell and the accumulator.
R_{14} = Average thermal resistance between the compressor and the suction pipe segment exposed to the outside.
R_{15} = Average thermal resistance between the suction pipe segment exposed to the outside and the surrounding medium.
R_{16} = Average thermal resistance between the suction pipe segment exposed to the outside and the accumulator.
R_{17} = Average thermal resistance between the accumulator and the surrounding medium.

$\dot{Q}_m$ = Rate of heat released by the compressor motor.

$\dot{Q}_{avc}$ = Average heat flow rate from the compressor cylinder walls to the gas inside the cylinder.

$\dot{Q}_{friction}$ = Rate of heat generated by friction in the compressor.

$\dot{Q}_{pipe1}$ = Rate of heat picked up by the gas from the pipe segment 1, as shown in Figure 2, when the gas flows through the suction passage.

$\dot{Q}_{pipe2}$ = Rate of heat picked up by the gas from the pipe segment 2, as shown in Figure 2, when the gas flows through the suction passage.

$\dot{Q}_{pipe3}$ = Rate of heat picked up by the gas from the pipe segment 3, as shown in Figure 2, when the gas flows through the suction passage.
C_{comp} = Thermal capacity of the compressor-rotor lumped mass.
C_{oil} = Thermal capacity of the oil.
C_{shelld} = Thermal capacity of the lower shell.
C_{shellu} = Thermal capacity of the upper shell-stator lumped mass.
C_{pipe2} = Thermal capacity of the suction pipe segment exposed to the outside.
C_{accu} = Thermal capacity of the accumulator and the suction pipe segment inside the accumulator.
C_{gas} = Thermal capacity of the gas inside the shell.

$\dot{m}_2$ = Mass flow rate of the gas exiting the shell.
h_2 = Enthalpy of the gas exiting the shell.
h_0 = Average enthalpy of the gas flowing out of the discharge port.

All the thermal capacities, except that of the gas, can be treated as constants. By summing up the heat flow rates at the node points of Figure 3, the energy balance equation of each lumped mass can be obtained, expressed as

$$\frac{d}{dt}(T_{comp}) = \frac{1}{c_{psteel} m_{comp}} \left(-\frac{T_{comp} - T_{shelld}}{R_3} + \frac{T_{gas} - T_{comp}}{R_6} - \frac{T_{comp} - T_{oil}}{R_2} - \frac{T_{comp} - T_{pipe2}}{R_{14}} + \frac{T_{shellu} - T_{comp}}{R_5} + \dot{Q}_{friction} - \dot{Q}_{ave} - \dot{Q}_{pipe3} \right) \quad (19)$$

$$\frac{d}{dt}(T_{oil}) = \frac{1}{c_{poil}\, m_{oil}} \left(\frac{T_{comp} - T_{oil}}{R_2} - \frac{T_{oil} - T_{shelld}}{R_1} + \frac{T_{shellu} - T_{oil}}{R_7} + \frac{T_{gas} - T_{oil}}{R_8} \right) \quad (20)$$

$$\frac{d}{dt}(T_{shelld}) = \frac{1}{c_{psteel}\, m_{shelld}}\left(\frac{T_{comp} - T_{shelld}}{R_3} + \frac{T_{oil} - T_{shelld}}{R_1} + \frac{T_{shellu} - T_{shelld}}{R_4} - \frac{T_{shelld} - T_{pipe2}}{R_{13}} - \frac{T_{shelld} - T_{amb}}{R_{11}}\right) \quad (21)$$

$$\frac{d}{dt}(T_{shellu}) = \frac{1}{c_{psteel}\, m_{shellu}}\left(\frac{T_{gas} - T_{shellu}}{R_9} - \frac{T_{shellu} - T_{accu}}{R_{10}} - \frac{T_{shellu} - T_{comp}}{R_5} - \frac{T_{shellu} - T_{oil}}{R_7} - \frac{T_{shellu} - T_{shelld}}{R_4} - \frac{T_{shellu} - T_{amb}}{R_{12}} + \dot{Q}_m\right) \quad (22)$$

$$\frac{d}{dt}(T_{pipe2}) = \frac{1}{c_{psteel}\, m_{pipe2}}\left(\frac{T_{comp} - T_{pipe2}}{R_{14}} + \frac{T_{shelld} - T_{pipe2}}{R_{13}} - \frac{T_{pipe2} - T_{amb}}{R_{15}} - \frac{T_{pipe2} - T_{accu}}{R_{16}} - \dot{Q}_{pipe2}\right) \quad (23)$$

$$\frac{d}{dt}(T_{accu}) = \frac{1}{c_{psteel}\, m_{accu}}\left(\frac{T_{shellu} - T_{accu}}{R_{10}} + \frac{T_{pipe2} - T_{accu}}{R_{16}} - \frac{T_{accu} - T_{amb}}{R_{17}} - \dot{Q}_{pipe1}\right) \quad (24)$$

where

c_{psteel} = Specific heat of steel
c_{poil} = Specific heat of the oil
m_{comp} = Mass of the compressor and the rotor
m_{oil} = Mass of the oil
m_{shelld} = Mass of the lower shell
m_{shellu} = Mass of the upper shell and the stator
m_{pipe2} = Mass of the suction pipe segment exposed to the outside
m_{accu} = Mass of the accumulator, including that of the suction pipe segment inside the accumulator

The mass of the gas inside the shell is not a constant during the transient process. Its rate of change is

$$\frac{d}{dt}(m_{gas}) = \dot{m}_{ave} - \dot{m}_2 \quad (25)$$

Applying the principle of energy conservation to the gas inside the shell yields

$$c_{vgas} m_{gas}\frac{d}{dt}(T_{gas}) + c_{vgas} T_{gas}(\dot{m}_{ave} - \dot{m}_2) = -\frac{T_{gas} - T_{shellu}}{R_9} - \frac{T_{gas} - T_{comp}}{R_6} - \frac{T_{gas} - T_{oil}}{R_8} + \dot{m}_{ave} h_0 - \dot{m}_2 h_2 \quad (26)$$

or

$$\frac{d}{dt}(T_{gas}) = \frac{1}{c_{vgas}\, m_{gas}}\left[-\frac{T_{gas} - T_{shellu}}{R_9} - \frac{T_{gas} - T_{comp}}{R_6} - \frac{T_{gas} - T_{oil}}{R_8} + \dot{m}_{ave}(h_0 - c_{vgas} T_{gas}) - \dot{m}_2 T_{gas}(c_{pgas} - c_{vgas})\right] \quad (27)$$

where c_{pgas} is the constant pressure specific heat of the gas inside the shell, c_{vgas} is constant volume specific heat of the gas inside the shell, and m_{gas} is instantaneous mass of the gas inside the shell.

The heat rate released by the motor, $\dot{Q}_m$, depends on the efficiency of the motor. It is expressed as

$$\dot{Q}_m = (1 - \eta_{motor})\, W_{in} \quad (28)$$

where η_{motor} is the motor efficiency, and W_{in} is the power input to the motor. Both of them are usually given.

Without studying the friction in the compressor in detail, the heat rate generated by friction can be approximated by the following relation

$$\dot{Q}_{friction} = (1 - \eta_{mech})\, \eta_{motor} W_{in} \tag{29}$$

where η_{mech} is the mechanical efficiency of the compressor. It can be empirically determined.

The heat transfer rates from the suction pipe segments to the suction gas, $\dot{Q}_{pipe1}$, $\dot{Q}_{pipe2}$, and $\dot{Q}_{pipe3}$, are respectively the terms on the left sides of Equations (3)-(5), which have been determined in Section 3.

Equation (19)-(25) and Equation (27) form a group of first order differential equations. The temperatures of the compressor system can be obtained by numerically solving those equations.

6. Results and Conclusions

Typical outputs are plotted in Figures 4-6. Three interpretations can be obtained from Figure 4. Firstly, the temperatures of the compressor-rotor lumped mass, the oil, the lower shell, the upper shell-stator lumped mass, and the shell gas go up with the increase of the compressor shaft rotation speed. Since the piston work required to complete one cycle does no change much, it follows that the faster the compressor shaft rotates, the more power is consumed, and the more energy is converted to heat per unit time for the given motor efficiency and mechanical efficiency factors. Secondly, the temperature of the upper shell-stator lumped mass is higher than that of the shell gas at high rotation speeds. This suggests that the refrigerant gas works as a kind of coolant for the motor. Thirdly, at higher speed, the transient process of the compressor system temperatures converges to steady state faster in time.

As discussed in the last paragraph of Section 1, the compressor speed has two direct effects on the heat transferred from the cylinder to the gas inside it. The higher speed reduces the time for heat to transfer from the cylinder to the gas per cycle, but enhances the convection heat transfer coefficient. Besides this, an indirect effect of higher speed on the heat transfer is through the fact that it causes the compressor metal temperature to increase, as shown in Figure 4. By carefully examining the cylinder gas temperature comparisons at different rotation speeds, as shown in Figure 5, we find that the cylinder gas temperature increases when the speed is increased. This can be further verified by checking the instantaneous mass curves of the cylinder gas, as shown in Figure 6. Less mass of gas is pumped into the cylinder per cycle at the higher speed. So, we conclude that at higher speed, suction gas heating is more severe, which causes a larger amount of mass loss.

Acknowledgement

Support of this research by the Toshiba Corporation is gratefully acknowledged.

References

[1] Incropera, F.P., and Dewitt, D.P., "Introduction to Heat Transfer", John Wiley & Sons, 1990.

[2] Atesmen, K.M., Trans. ASME, Ser. C, Vol. 97, No. 2, P. 288, May, 1972

[3] Adair, R.P., M.S. Thesis, Purdue University, 1972.

[4] Recktenwald, G.W., Ramsey, J.W., and Patankar, S.V., "Prediction of Heat Transfer in Compressor Cylinders", Proceedings of the 1986 Purdue Compressor Technology Conference, pp. 159-174.

[5] Yanagisawa, Tadashi, et al., "A Study on Suction Gas Heating in a Rolling Piston Type Rotary Compressor", Bulletin of the JSME, Vol. 27, April, 1984, pp. 741-748.

[6] "Handbook of Single-Phase Convective Heat Transfer", Wiley, 1987

[7] Dhar, M., Ph.D Thesis (Major professor, W. Soedel), Purdue University, 1978.

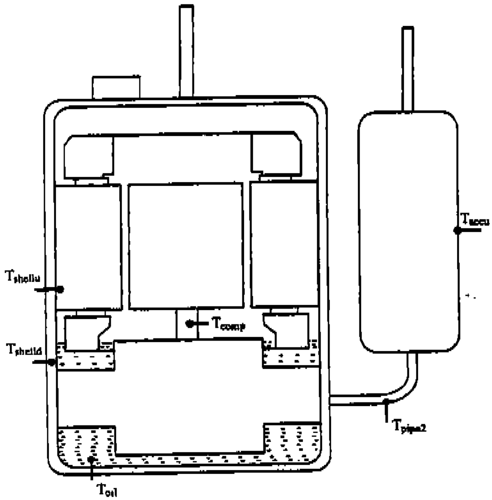


Fig.1 Bulk Temperatures of the compressor system

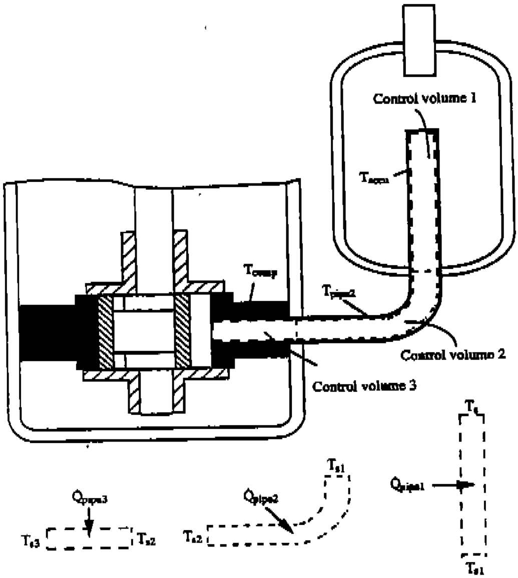
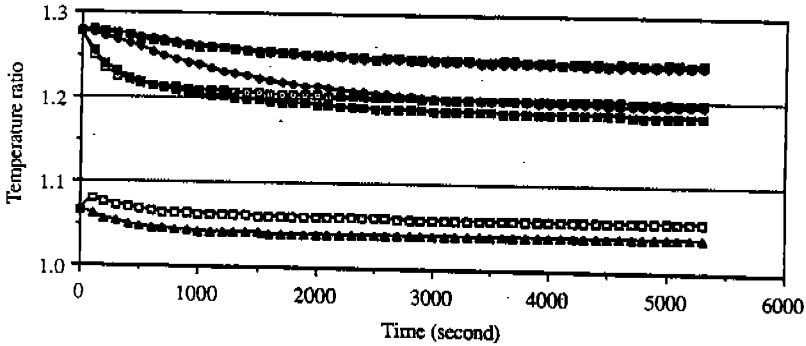
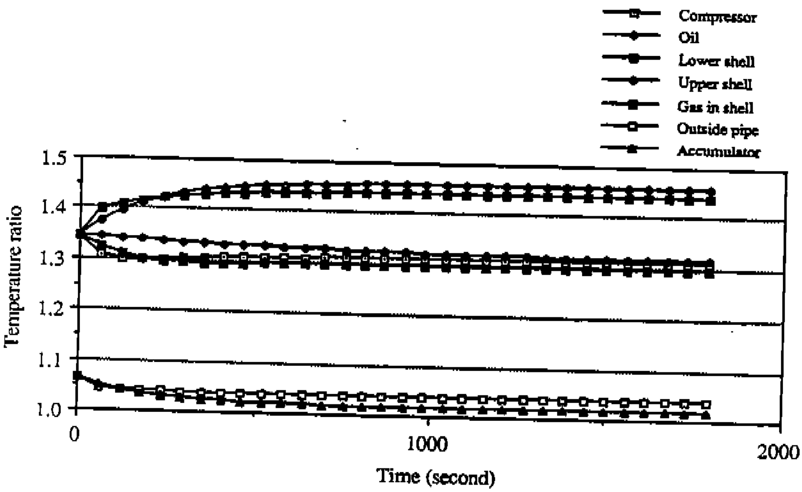


Fig.2 Heat transfer model of the gas in the suction pipe



(a) At the rotation speed of 1800 rpm



(b) At the rotation speed of 9000 rpm

Fig.4 The ratios of compressor system temperatures at different speeds to a reference temperature

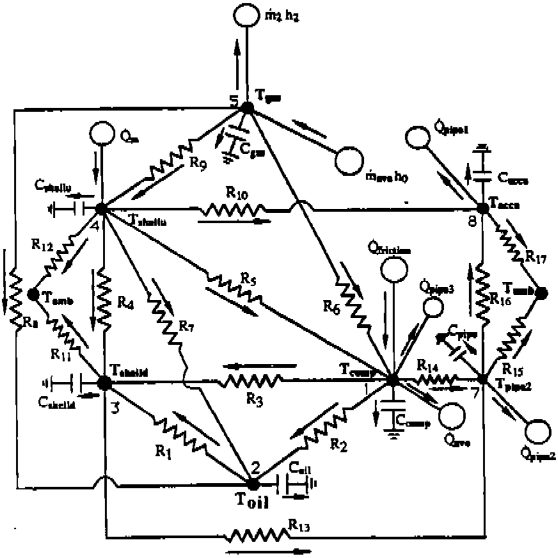


Fig.3 heat flow diagram of the compressor system

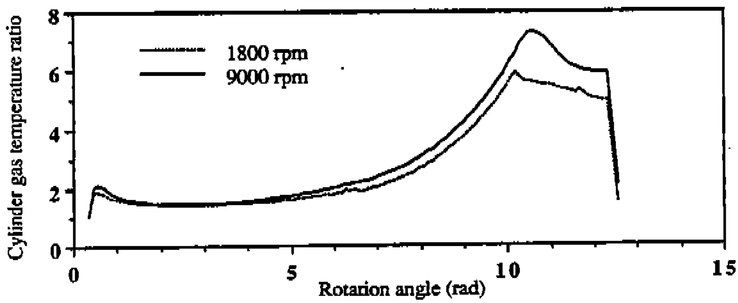


Fig. 5 The ratio of cylinder gas temperature at different speeds to a reference temperature

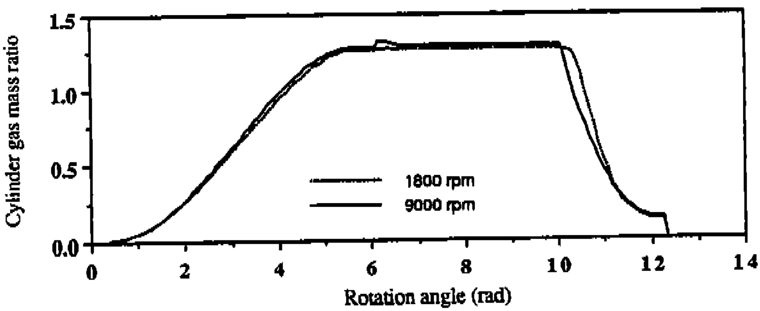


Fig. 6 The instantaneous cylinder gas mass ratio at different speeds