ASIATRIB 2002International Conference
JEJU ISLAND, KOREA

Dynamic Analysis of the Piston Slap Motion in Reciprocating Compressors

Tae-Jong Kim [1]

[1]*School of Mechanical Engineering, Pusan National University, San 30, Jangjeon-Dong, Kumjung-Ku, Pusan 609-735, KOREA*

Piston-cylinder systems are widely used in power engineering applications. In reciprocating refrigeration compressors, where extremely low friction losses are required, ringless pistons are being used to diminish the friction between piston rings and cylinder wall. Since the ringless piston has the freedom of lateral motion there is a potential danger that it will occasionally hit the cylinder wall while moving up and down along it's axis. A good design must therefore provide a smooth and stable reciprocating motion of the piston and ensure that the fluid film separating the piston from the cylinder wall is maintained all times. And the compromise between refrigerant gas leakage through the piston-cylinder clearance and the friction losses is required utilizing a dynamic analysis of the secondary motion for the high efficiency compressor. To this end, the computer program is developed for calculating the entire piston trajectory and the lubrication characteristics as functions of crank angle under compressor running conditions. The results explored the effects of some design parameters and operating conditions on the stability of the piston, the oil leakage, and friction losses.

Keywords: Reciprocating Compressors, Piston Secondary Dynamics, Hydrodynamic Lubrication, Piston Trajectory, Oil leakage, Friction Loss

1. INTRODUCTION

In a reciprocating compressor used in domestic refrigerators, where extremely low friction loss is required, ringless pistons are being used to eliminate the friction between piston and cylinder wall. And, the length of cylinder in this class of compressors is shortened to diminish the frictional losses of the piston-cylinder system. So, the contacting length between piston and cylinder wall is in variable with the rotating crank angle around the BDC of the reciprocating piston. Early studies on this lubrication model by Li et al.[1] and Zhu et al.[2] solved the problem for reciprocating pistons in automotive engines. Parata et al.[3] performed a dynamic analysis for the oil film between piston and cylinder in small refrigerating compressors. The piston oscillatory motion in the cylinder bore is directly related to the lubrication characteristics of the piston-cylinder clearance. It is certainly necessary to develop and improve piston-cylinder lubrication analysis for better understanding of piston dynamics and reliable prediction of friction characteristics. In this paper, a problem formulation for the piston dynamics is presented considering the variation in bearing length of the piston and hydrodynamic forces and moments between piston and cylinder wall. The corresponding computer program is developed for calculating the entire piston trajectory and the hydrodynamic lubrication characteristics as functions of crank angle under compressor running conditions. And the results explored the effects of the radial clearance, lubricant viscosity, length of the cylinder wall, and pin location on the stability of the piston, the oil leakage, and friction losses.

2. PISTON SECONDARY DYNAMICS

2.1 Equations of Motion

A piston-cylinder system in the reciprocating compressors is shown in Fig. 1 and the equations describing the piston secondary dynamics can be written in dimensionless form as.

$$T_x + F_h = m_p \ddot{x}_0 \tag{1}$$

$$M_f + M_h = I_p \ddot{\alpha} \tag{2}$$

And, T_x is the radial component of the connecting rod force, F_h is the hydrodynamic force due to the pressure in the oil film. M_h and M_f are, respectively, the moments about the wrist-pin due to the F_h and viscous frictional force F_f. The acceleration of piston center of mass in the x direction is $\ddot{x}_0$ and the angular acceleration is $\ddot{\alpha}$. The boundary position B of the piston is not in fixed if the sum of piston length and stroke exceeds the length of the cylinder wall. The nondimensional accelerations $\ddot{x}_0$, $\ddot{\alpha}$ can be found from the lateral acceleration $\ddot{\varepsilon}_b$, $\ddot{\varepsilon}_t$ of the two ends A, B of the reciprocating piston as shown in Fig.1.

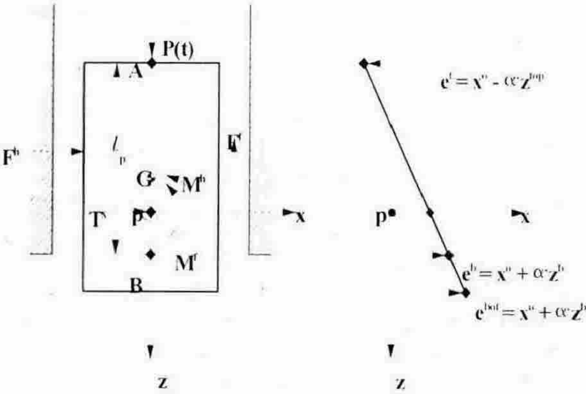

Fig. 1 FBD of the reciprocating piston

$$\ddot{x}_0 = \ddot{\varepsilon}_b - \left(\frac{\ddot{\varepsilon}_b - \ddot{\varepsilon}_t}{\ell_p}\right) z_b \, , \; \ddot{\alpha} = (\ddot{\varepsilon}_b - \ddot{\varepsilon}_t) / \ell_p \tag{3}$$

Equations (1), (2) can be transformed to the following nonlinear equation system for the application of Newton-Raphson method.

$$f_1(\ddot{\varepsilon}_b, \ddot{\varepsilon}_t) = T_x + F_h - m_p \left[\ddot{\varepsilon}_b - (\ddot{\varepsilon}_b - \ddot{\varepsilon}_t)\frac{z_b}{\ell_p}\right] = 0 \tag{4}$$

$$f_2(\ddot{\varepsilon}_b, \ddot{\varepsilon}_t) = M_f + M_h - I_p(\ddot{\varepsilon}_b - \ddot{\varepsilon}_t) / \ell_p = 0 \tag{5}$$

2.2 Hydrodynamic Analysis

The force and moment acting on the piston skirt, F_h and M_h, are due to the hydrodynamic pressure developed in the oil film, can be obtained from the Reynolds equation. The Reynolds equation for incompressible and laminar flow can be written in nondimensional form as

$$\frac{\partial}{\partial\theta}(h^3\frac{\partial p}{\partial\theta}) + \frac{\partial}{\partial z}(h^3\frac{\partial p}{\partial z}) = 6V_p\frac{\partial h}{\partial z} + 12\frac{\partial h}{\partial t} \quad (6)$$

This equation can be discretized with the conventional finite difference scheme. The hydrodynamic pressure can be expressed by using SOR (Successive Over Relaxation) method with over-relaxation parameter. The pressure in the oil film is known, the hydrodynamic force and moment, the viscous frictional force and moment can be obtained, respectively. In integrating above terms, whenever cavitation occurred, the oil pressure was replaced by a gas pressure interpolated between dynamic cylinder pressure P_{cyl} and static suction pressure P_{sus}, depending on the cavitation axial location. Therefore, cyclically averaged power consumption and instantaneous volumetric oil leakage through the clearance between piston and cylinder can be determined.

3. RESRULTS AND DISCUSSIONS

Results for the x direction orbits of the piston center located at the piston top, piston-pin, piston boundary location, and piston bottom are shown in Fig. 2. This figure shows the converged periodic solution for the piston trajectory inside the cylinder within about 20~30 cycles depending on the values of design parameters. It can be shown that there are different trajectories between the boundary and bottom locations during the crank angle of 80.9° ~ 278.8° . The trajectory of the piston top center is closing on the thrust side of the cylinder wall, but the piston bottom center is lying on the center line of the cylinder. The influence of the radial clearance on the secondary oscillatory motion of the piston is presented in Fig. 3 for an entire cycle. From this figure, it is seen that small values of radial clearance results in increasing damping of the oil film which, in turn, tend to stabilize the secondary motion of the piston. As seen in Fig. 4, the averaged power consumption per cycle for the fixed value of clearance of $4\,\mu m$ increases almost linearly with the oil viscosity. In the same condition, the variation of the averaged oil leakage per cycle with lubricant viscosity parameter is shown. And, the averaged power consumption per cycle for the fixed value of viscosity $5\,mPa\cdot s$ is shown with the various values of the side clearance of piston. Also, the averaged oil leakage per cycle as a function of clearance of the piston is shown in the same figure.

4. CONCLUSIONS

A mathematical model was developed for the secondary dynamic analysis of reciprocating pistons considering the hydrodynamic characteristics of the oil film between piston and cylinder. A computer program developed on this model can be used to analyze the secondary dynamic characteristics of reciprocating piston by calculating the trajectory of the piston, power consumption and oil leakage. The influence of the radial clearance, oil viscosity, pin location, and length of cylinder wall on the piston dynamics and lubrication characteristics are explored.

5. ACKNOWLEDGEMENT

This work was supported by grant No. R01-2001-00383 from the Korea Science & Engineering Foundation.

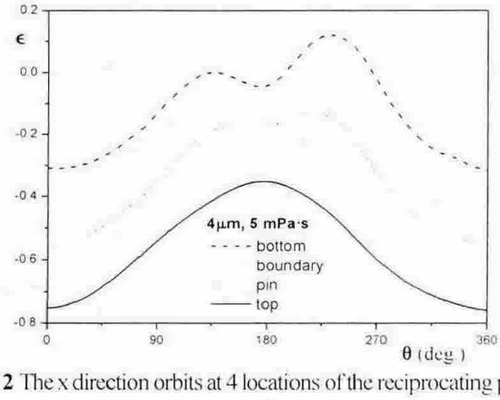

Fig. 2 The x direction orbits at 4 locations of the reciprocating piston

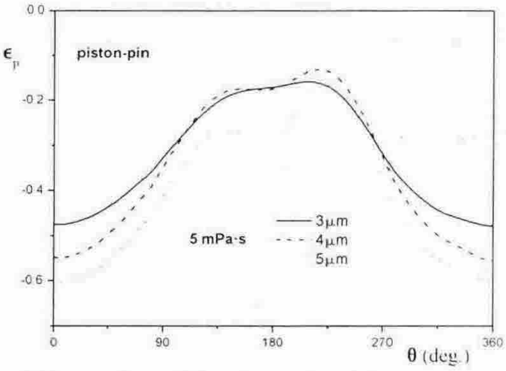

Fig. 3 Comparison of the piston pin orbits variation in the radial clearance between piston and cylinder

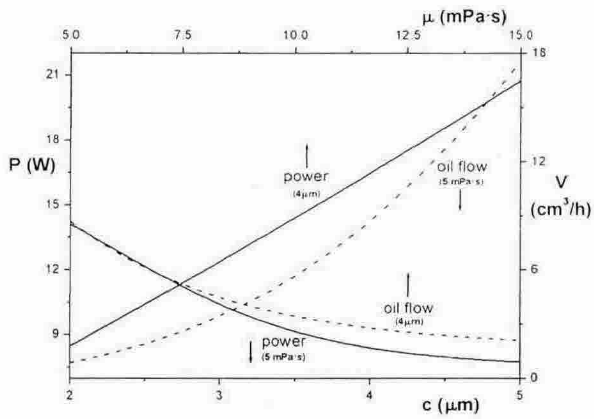

Fig. 4 Cycle averaged power consumption and oil leakage as a function of the radial clearance and a function of the lubricant viscosity

6. REFERENCES

[1] Li, D. F., Rohde, S. M., and Ezzat, H. A., , "An Automotive Piston Lubrication Model," ASLE Transactions, Vol. 26, No. 2, pp. 151~160, 1983.

[2] Zhu, D., Cheng, H. S., Arai, T., and Hamai, K., "A Numerical Analysis for Piston Skirts in Mixed Lubrication Part 1: Basic Modeling," Journal of Tribology, Vol. 114, pp. 553~562, 1992.

[3] Prata, A. T., Fernando Julio R. S., and Fagotti, F., "Dynamic Analysis of Piston Secondary Motion for Small Reciprocating Compressor," Trans ASME, Journal of Tribology, Vol. 122, pp. 752~760, 2000.