1996

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International Compressor Engineering Conference. Paper 1091.  
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Lubrication Analysis on the profile of slider in reciprocating compressor

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Abstract

Theoretical Analysis on the profile of slider in reciprocating compressor has been presented based on the hydrodynamic lubrication. Theoretical Analysis calculates fluid film pressure, attitude angle, frictional force, and minimum film thickness in normal condition for comparing two different profiles of slider. The calculated results show that the profile of slider plays an important role in the determination of the lubrication characteristics and the barrel type profile is more efficient from the viewpoint of lubrication.

Introduction

The slider in a reciprocating compressor undergoes an oscillatory motion in the shell when compressor operates and piston moves up and down along the axis of the cylinder. This oscillatory motion is usually small but very important to compressor performance and reliability. All the major concerns in design, such as frictional loss, noise and anti-wear life, are significantly related to the mutual dependence between the slide motion and lubrication. It is certainly needed, therefore, to develop and improve numerical analysis for better understanding and reliable prediction of slider motion, friction and lubrication at the slider-shell interface.

In this paper, theoretical analysis on the profile of slider in reciprocating compressor has been presented based on the hydrodynamic lubrication. And the approach analyses shows fluid film pressure, attitude angle, frictional force, and minimum film thickness in normal condition to compare two different profiles of slider.

Nomenclature

<table>
<thead>
<tr>
<th>F</th>
<th>: Force</th>
<th>φ</th>
<th>: Attitude Angle</th>
</tr>
</thead>
<tbody>
<tr>
<td>M</td>
<td>: Moment</td>
<td>μ</td>
<td>: Viscosity</td>
</tr>
<tr>
<td>h</td>
<td>: Film Thickness</td>
<td>ρ</td>
<td>: Pressure</td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
<td></td>
<td></td>
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<tr>
<td>--------</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>I</td>
<td>Inertia Moment</td>
<td></td>
<td></td>
</tr>
<tr>
<td>U</td>
<td>Speed</td>
<td></td>
<td></td>
</tr>
<tr>
<td>l_s</td>
<td>Length of slider</td>
<td></td>
<td></td>
</tr>
<tr>
<td>x,z</td>
<td>Cartesian Coordinate</td>
<td></td>
<td></td>
</tr>
<tr>
<td>α</td>
<td>Thermal expansion coefficient</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Governing Equation**

**Equation of Motion**

A schematic drawing of shell/slider system is presented in Fig 1. The external forces acting on the surface of slider are depicted as \( F_{\text{ext1}}, F_{\text{ext2}} \). The descriptive equation of motion for the slider can be simply expressed as following:

\[
0 = F_h + F_{\text{ext1}} + F_{\text{ext2}} \tag{1}
\]

\[
l_s \varphi'' = M_h + M_{\text{ext1}} + M_{\text{ext2}} \tag{2}
\]

where \( F_h, M_h \) are due to the hydrodynamic action and \( M_{\text{ext1}}, M_{\text{ext2}} \) are due to the external forces.

**Equation of Lubrication**

The following steady incompressible Reynolds Equation is employed in the present work to include the effect of profile on the hydrodynamic lubrication.

\[
\frac{\partial}{\partial x} \left( \frac{h^3}{12 \mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( \frac{h^3}{12 \mu} \frac{\partial p}{\partial z} \right) = \frac{U}{2} \frac{\partial h}{\partial x} \tag{3}
\]

And lubricant film thickness equation can be well approximated by the following;

\[
h(x, \varphi) = h_o + h_i(x) \cos \varphi + x \sin \varphi \tag{4}
\]

where \( h_o \) means minimum film thickness and \( h_i \) is due to thermal expansion which can be evaluated by following equation;

\[
h_i(x) = R_i^{\text{max}} - R_i(x) \tag{5}
\]

where

\[
R_i(x) = R_i(x) \left( 1 + \alpha \cdot \Delta T(x) \right) \tag{6}
\]
\( R_{\text{R}}^{\text{max}} \) denotes the maximum radial deformation due to thermal expansion, \( R_{\text{To}} \) the radial profile in atmosphere, \( \Delta T \) the difference of temperature between actual operation and atmosphere. Substituting (5),(6) into (4) gives the following lubricant film thickness equation.

\[
h(x, \varphi) = h_0 + R_{\text{R}}^{\text{max}} - [R_{\text{R}}^{\text{max}} - h_r(x)] [1 + x \cdot \Delta T(x)]
\]

---7)

In order to solve the Reynolds Equation, Half-Sommerfeld boundary condition is used. After obtaining the hydrodynamic pressure distribution by solving the Reynolds Equation, the normal force due to the hydrodynamic pressure and its moment about slider hole can be calculated by the following integration;

\[
F_n = \int \int_A p(x, y) \, dx \, dy
\]

---8)

\[
M_n = \int \int_A p(x, y) \left( y - 0.5 l_x \right) \, dx \, dy
\]

---9)

in which \( A \) is the total bearing area on both thrust and antithrust sides.

And hydrodynamic friction force can also be calculated based on the shear stress;

\[
F_s = \int \int_A \left( \frac{\mu U}{h} \pm \frac{h}{2} \frac{\partial p}{\partial \theta} \right) \, dx \, dy
\]

---10)

Results and Discussion

The two models of radial profile in slider is shown Fig.2, where the profile of slider(b) has the barrel type of characteristics compared with the profile of slider(a).

Calculated velocity and resultant force acting on the slider based on the given gas pressure with time is plotted in Fig 3 and Fig 4 as a function of crank angle. Note that suction stroke is from 0 to 180 deg crank angle, discharge stroke from 180 to 360 deg.

Figure 5 shows the calculated result of attitude angle corresponding to each of two profiles. The thin line is the attitude angle of slider (b), where \( \phi / \phi^{\text{max}}(b) \) means the relative value to maximum attitude angle of slider (b). It can be seen that slider (a) contacts with thrust side in the whole range, but slider (b) contacts occasionally with antithrust side, which is probably due to the improvement of lubrication characteristics.

Figure 6 shows the minimum film thickness The bold line is the minimum film thickness of slider (a), where
h / h_{min} (a) means the relative value to minimum film thickness of slider (a). It is found that in this case minimum film thickness of slider(a) is nearly close to the arithmetic sum of surface roughness. On the other hand, slider(b) has occasionally thick film thickness.

Asso, the frictional force is shown by Fig.7. The thin line is the frictional force of slider (b), where F / F_{max} (b) means the relative value to maximum frictional force of slider (b). The magnitude of frictional force in slider(b) is much smaller than that in slider(a). This probably comes from enough film thickness, i.e. good lubrication.

In any cases above the slider has two unavoidable dead points, where the velocity is zero, due to its typical compression mechanism. This means that film thickness is minimum and frictional force is close to zero at these points theoretically.

Conclusion

Theoretical analysis on the profile of slider in reciprocating compressor has been developed based on the hydrodynamic lubrication. Some conclusion drawn from this study are as follows;

(1) A computer program is constructed based on the profile of slider. It can be used to predict the entire trajectory of slider and hydrodynamic friction as a function of crank angle.

(2) It is shown that the profile plays an important role in the determination of slider motion and friction in a point of the lubrication characteristics.

(3) Good hydrodynamic lubrication with the barrel type of profile can minimize the slider wear against the thrust side and reduce the frictional loss.

Aknowledgement

The authors thank Dr.Kwan-Sik Cho, the leader of compressor team of LG Electronics Inc. for his advice during the achievement of this study.

Reference

1.Kawahira, "Hermetic Compressor", pp.27, 1971
Fig 1. A schematic drawing of shell/slider

Fig 2. The 2 profiles of slider
Fig 3. Velocity of Slider

Fig 4. Resultant Force on Slider

Fig 5. Attitude Angle

Fig 6. Minimum Film Thickness

Fig 7. Frictional Force