Hydrodynamic Optimization of the Tribological System Piston-Ring-Liner

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ABSTRACT

Mechanical friction losses in piston type machines amount to approximately 60% in the tribological system "piston-ring-liner". Ring surface improvement by proper design of the ring profile can contribute to optimize this tribological system. Hereby hydrodynamical load capacity should be optimized and caused friction losses are to be minimized.

Simulation programs are available to calculate the conditions in the lubricant between piston ring and liner. Frictional forces, bearing capacities and the thickness of lubrication film can be estimated by use of these programs. Uniform velocity of the piston can be described by these simulation programs as well as variable velocities.

By use of these programs calculations have been made for different profiles of piston rings. Investigations were made on symmetrical wedge shaped, barrel shaped and extreme barrel shape profiles. Additionally an unsymmetrical barrel shaped ring was examined.

Experimental results are required for comparison with computed results. Therefore a special test rig exists. Band shaped piston rings can be studied on a model slider. All required data to describe the oil film characteristics can be measured with this experimental set up. By use of a special crankshaft simulator uniform velocities can be simulated as well as the cinematics of crank mechanism. The different profiles have been examined under uniform and variable conditions. Especially at TDC and LDC and at the point of maximum speed the experimental results must be analysed. At these points the influence of the different ring profiles is of special interest with regard to film thickness and friction losses.

INTRODUCTION

In a reciprocating compressor the system of piston-ring-liner is one of the most important assemblies. Approximately 60% of the frictional forces caused in reciprocating machines result from this tribological system. Regarding the high amount of piston and piston ring friction it is of great importance to optimize the ring profile in the sence of minimized losses.

A further aim of piston ring lubrication is to separate the sliding surfaces by the lubricant specially at LDC and TDC. To achieve both aims it is necessary to improve the profile of piston rings in order to attain as well low frictional losses at the point of maximum speed as sufficient oil film thickness at the ends of stroke.

ANALYSIS

Only some theoretical approaches have been made to predict the lubrication conditions of the piston ring liner system. At the Purdue Compressor Technology Conferences in 1974, 1976 and 1978 Kruse /1,2/ and Wrede /4/ presented a mathematical solution of this problem. The most important equations are given in these papers.

Postulating the well known assumptions for incompressible Newtonian fluids the mathematical model can be described by the Reynolds' equation as follows:

\[ \frac{\partial}{\partial x} (h^3 \frac{\partial p}{\partial x}) + \frac{\partial}{\partial z} (h^3 \frac{\partial p}{\partial z}) = 6\eta (U_1 - U_2) \frac{\partial h}{\partial x} + 6h \frac{\partial}{\partial x} (U_1 + U_2) + 12nV \]

Eq. 1

This equation cannot be solved with an acceptable amount of computation time. It can be simplified in the following ways:

For a non-uniform lubricant distribution in the region of piston ring liner the problem can be described by:

\[ \frac{\partial}{\partial x} (h^3 \frac{\partial p}{\partial x}) + \frac{\partial}{\partial z} (h^3 \frac{\partial p}{\partial z}) = 6\eta \frac{\partial h}{\partial x} \]

Eq. 2

neglecting radial motion of piston or ring as introduced by non uniform velocity. The lubricant is described hereby in the most simple three dimensional steady state form of Reynold's equation. It has to be changed to a system of finite difference expressions for solving.

Distributing a centric position of piston and ring in the cylinder and an uniform lubrication film but taking into account a certain radial motion can be simplified in the following way:

\[ \frac{\partial}{\partial x} (h^3 \frac{\partial p}{\partial x}) = 6\eta \frac{\partial h}{\partial x} + 12nV \]

Eq. 3
This two dimensional model describes nonsteady events including variable piston velocities and squeeze effects within the fluid.

The simulation program where this model is installed therefore can describe a real type reciprocating machine with velocities according to the connecting rod assembly:

\[ u = r \cdot \omega \cdot (\sin(\omega t) + \frac{\lambda}{2} \cdot \sin(2\omega t)) \]

Eq. 4

This special simulation model also enables to calculate lubricant film thickness at TDC and LDC. This is one of the most important points for estimating the behaviour of piston rings regarding prevention of wear. Additional frictional losses which are cause in the middle of the stroke at the point of highest piston velocities can be analysed.

Following chapters a theoretical and experimental analysis of the lubrication conditions in the region of piston ring liner will be given. Steady state conditions will be discussed as well as nonsteady ones.

PRESENTATION OF THEORETICAL RESULTS

Lubrication conditions of the system piston-ring-liner have been computed using the above described simulation models. Estimating uniform velocities the pressure distribution of the oil film has been calculated using model described first. Side flow of lubrication is taken into account here. Parameters which are assumed constant in these calculations are:

- relative velocity between sliding surfaces
- load on the piston ring
- available amount of lubricant

Figure 1. Pressure distribution under a wedge shaped ring

Calculating the pressure distribution under a wedge shaped ring shown in figure 1 a relative velocity of 3 m/s and a ring load of 2 bar was supposed. The pressure builds up in the lubricant starts at the inlet of the oil film gap. The highest value of 5,1 bar is reached under the front section of the piston ring. The length of the pressure profile amounts to about 63% of the length of the piston ring itself. By definition the length of pressure profile at the diverging wedge is limited to that point where the oil film pressure equals the surrounding one.

Figure 2. Pressure distribution under a barrel shaped ring

The pressure distribution caused under a barrel shaped ring (fig. 2) looks similar to that one described before. Under same conditions its highest value reaches 5,7 bar. The length of the profile is about 60% of the ring length in this case.

Figure 3. Pressure distribution under a extreme barrel shaped ring

Remarkable for the third ring, an extreme barrel shaped one, is that the pressure distribution is not as symmetrical as simulated for the first two profiles (fig. 3). The less gradually increasing film pressure reaches its highest value at a range of 8,8 bar. This even increase of pressure distribution in direction to its maximum value is followed by an abrupt pressure drop.
Calculated pressure profiles caused by an unsymmetrical barrel shaped ring profile are shown in figures 4 and 5. Two possibilities have to be discussed in this case:

1. the larger wedge is responsible for pressure built up,
2. the opposite side of the ring is the leading part.

Thinking of the first possibility (fig. 4) the maximum value of caused film pressure will be 3.7 bar. It is to be found under the trailing part of the ring. Length of pressure profile and piston ring are nearly the same.

In case of other sliding direction (fig. 5) the length of the generated pressure profile in the lubricant is limited to a minor section under the piston ring. The length of profile is only in the range of 20% of the ring length. The pressure increases with a steep slope to its maximum value of 18.2 bar. Since the length of distributed pressure profile is much smaller in this case higher peak pressures must be generated to bear the same piston ring load as before.

For all ring profiles discussed above can be stated the peak pressure generated in the lubricant depends on the length of pressure profile. Peak pressures in the lubricant will be found more and more in the leading part of the piston ring when its sliding velocity is increased. This was found for all ring profiles discussed in this paper.

Minimum oil film thickness obtained from these simulations are shown in fig. 6. Constant velocity between sliding partners and constant ring loading was assumed when the different values were calculated. The minimum film thickness increases with increasing velocities.

The difference between results obtained for the wedge shaped and the symmetrical barrel shaped ring are not significant. On the other hand smaller values are computed for the extreme barrel shaped ring. With the long leading wedge in the sliding direction very high values of film thickness are obtained for the unsymmetrical barrel shaped ring. In case of other sliding direction the calculated values of the oil film thickness are very poor.

DESCRIPTION OF INSTALLED TEST RIG

Fig. 7 shows a survey drawing of the experimental rig described by Wrede /3,4/ at the 1978 Compressor Technology Conference. The sliding system and the related hydraulic drive are arranged on a common foundation. Piston, piston ring and liner are modeled as a plain sliding pair.

In this rig the plane liner is operated by a special hydraulic drive whereas piston and piston ring model are fitted within a stationary support.
Fig. 8 shows the corresponding geometry of the piston model equipped with a double wedge type piston ring.

The piston model is suspended by pre-stepped three-component dynamometer in order to measure caused lubricant film forces. The pressure profile of the lubrication film can be measured by a piezoelectric pressure pick up what is installed within the liner. By means of these installations the following signals can be registrated:

- oil film pressure
- pressure distribution
- oil film thickness
- oil film temperatures
- film forces
  (frictional forces as well as load forces)

The crank assembly of the test rig is operated by a hydrostatic linear drive. This consists of a hydraulic circuit, a double acting high speed hydraulic cylinder, an electro-hydraulic servo valve and an electronic crank assembly simulator. Hereby the following variations are possible:

\[
\begin{align*}
& r = 0 \div 400 \text{ mm} \\
& \lambda = 0 \div 0.3 \\
& \omega = 0 \div 12 \text{ s}^{-1}
\end{align*}
\]

- r = connecting rod length
- \( \lambda \) = connecting rod ratio
- \( \omega \) = cyclic frequency

Additional constant speed upto 4.5 m/s can be simulated.

To measure the lubrication conditions in the piston, piston-ring-liner assembly tests were carried out using various shaped piston ring models. Since representative signals were required in order to compare theoretical and experimental results various ring models were used. Therefore the ring profiles had to be adjusted to those which have been discussed earlier in the theoretic part of this paper. The ring profiles are shown in fig. 10.

First the wedge shaped profile is shown (angle of wedge = 0.10 deg, height of wedge = 10 \( \cdot \) 10^{-6} m). The barrel shaped profile is symmetrical, the height of profile is 20 \( \cdot \) 10^{-6} m in this case. Thinking of the extreme barrel shaped ring this height amounts to 70 \( \cdot \) 10^{-6} m. The last profile shown in fig. 10 is the unsymmetrical barrel shaped one. The height...
at one side of the ring will be $90 \cdot 10^{-6}$ m and $20 \cdot 10^{-6}$ m at the other side.

![Figure 10. Sliding profiles](image)

**COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS**

Measured and calculated values of oil film thickness are compared in fig. 11.

![Figure 11. Minimum oil film under a barrel shaped ring](image)

The non dotted line represents computed results whereas the singular points show the measured values. A good correspondence is obvious and availability of the theoretic model (eq. 2) can be stated therefore. Comparisons shown up to now, only have been made with constant velocity distributions of the piston.

For non uniform velocities following the crank assembly formula (eq. 4) measurements also have been made. Fig. 12 shows such results of measured lubrication film thickness as caused under this velocity distribution.

![Figure 12. Minimum oil film thickness](image)

The upper three lines describe the film thicknesses at the point of maximum piston velocity next to the mid of the stroke ($\varphi = 90^\circ$ deg). The lower curves in the picture show values of measured film thickness at TDC and LDC ($\varphi = 0^\circ$ and $\varphi = 180^\circ$ deg). At cyclic frequencies higher than $\omega = 3.9$ s$^{-1}$ better lubrication conditions were obtain at the point of max. velocity with a barrel shaped ring than with a wedge shaped one.

Best squeeze film lubrication in the dead centers is obtained from a wedge shaped ring profile. In both points the measured film thickness under extreme barrel shaped ring profiles was smaller than those discussed before. Fig. 13 shows a comparison of theoretical and experimental results obtained under nonsteady conditions.

![Figure 13. Minimum oil film thickness under a wedge shaped ring](image)

The minimum film thickness has been calculated for different crank angles. The correspondence is dearly visible, indicating that this theoretical model described by eq. 3 is valid for this kind of lubrication problem as well as shown for the first model.
CONCLUSION

To improve hydrodynamic lubrication conditions in the piston piston-ring-liner assembly with regards to frictional losses and prevention of wear piston sliding profiles have to be optimized. Theoretical and experimental results have been compared for steady state and nonsteady state conditions. In both cases good correlation between theoretical and experimental results were stated. In order to optimize those piston ring profiles with regard to smallest friction losses and highest load capacity the application of these simulation programs can be a useful help. By this means a prediction of reduced energy losses and prevention of wear can be made.

REFERENCES


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