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Research Article

Analysis of the Journal Bearing Performance of a Reciprocating Compressor Using Methane

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Abstract: The performance of the journal bearing of a reciprocating compressor is crutial in industrial applications utilizing reciprocating compressors systems. This study analyzes a system utilizing EXXCOLUB SLG 190 as lubricant and methane, as the compressed gas. Numerical method, specifically the finite element method, is utilized in this study. Reynolds equation is analyzed using the half sommerfield boundary condition. The analysis encompasses a study of the effect of viscosity thinning with solubility of the methane at various operating pressures on the pressure profile and the load capacity of the bearing. The effect of vibration on the performance of the journal bearing is also analyzed by considering the effect of vertical flow. It was observed that the maximum pressure reduced by 50% as well as the load capacity of the journal bearing, when the operating pressure was increased from 0 to 5000 psi (3.447×10^7 N/m^2). The pressure profile flattens out as the operating pressure increases. Increase in vibration (vertical flow effect) results in the reduction of load capacity of the journal bearing.

Keywords: Finite element method, load capacity, lubrication, pressure profile, viscosity

INTRODUCTION

Journal bearing performance is important as it determines the load capacity for efficient operation of machinery. The lubricant viscosity used is a very important factor in its performance. Factors that affect the lubricant indirectly affect the journal bearing performance. The analysis of journal bearing has been undertaking for some decades.

The mobility method provides a simplified analytical, graphical and numerical solution to dynamically loaded journal bearing problems (Booker, 1965). Goenka (1984) improved on it by incoporating curve fit solution. New curve fits were presented which included two components of mobility vectors, the location and magnitude of the maximum film pressure and starting and finishing angles of the pressure curve. In addition, through the thermohydrodynamic analysis of the journal bearing, two temperature-rise parameters important for the temperature filed were discovered. One is related to the oil properties and the second is a function of shaft velocity with stated lubricant properties (Khonsari *et al*., 1996). Refrigeration oil with lower refrigerant solubility yields better characteristics in the bearing lubrication zone (Kobayashi *et al*., 1998). Distinct behaviour was observed, for the different lubricant to carbon dioxide and the information is useful in the design and uses of system utilizing carbon dioxide (Li and Rajewski, 2000). The vapour pressure of carbon dioxide/lubricant mixture provide measurement for solubility, liquid viscosities provides information for lubricity, falex testing provides information for tribology wear and boundary lubrication and sealed glass tube testing enables miscibility to be detected (Seeton *et al*., 2000).

Laboratory test and field trials indicate that progress is being made to increase the performance of carbon dioxide system by compressor/system manufacturers and lubricant suppliers (Li *et al*., 2002). Using the multiple regression method to analyse journal bearing of a reciprocating compressor, It was observed that both the solubility of carbon dioxide into the oil and the oil temperature have significant effect on the viscosity of the oil and friction power loss of the system (Yang *et al*., 2010).

The present study is to analysed the mathematical model proposed by Yang *et al*. (2010). Finite element analysis is used to determine the pressure profile and load capacity of the journal bearing of a reciprocating Compressor used to compress methane. The lubricant used was EXXOCOLUB SLG 190. Vertical flow was also used to analyse the effect of vibration in the system.

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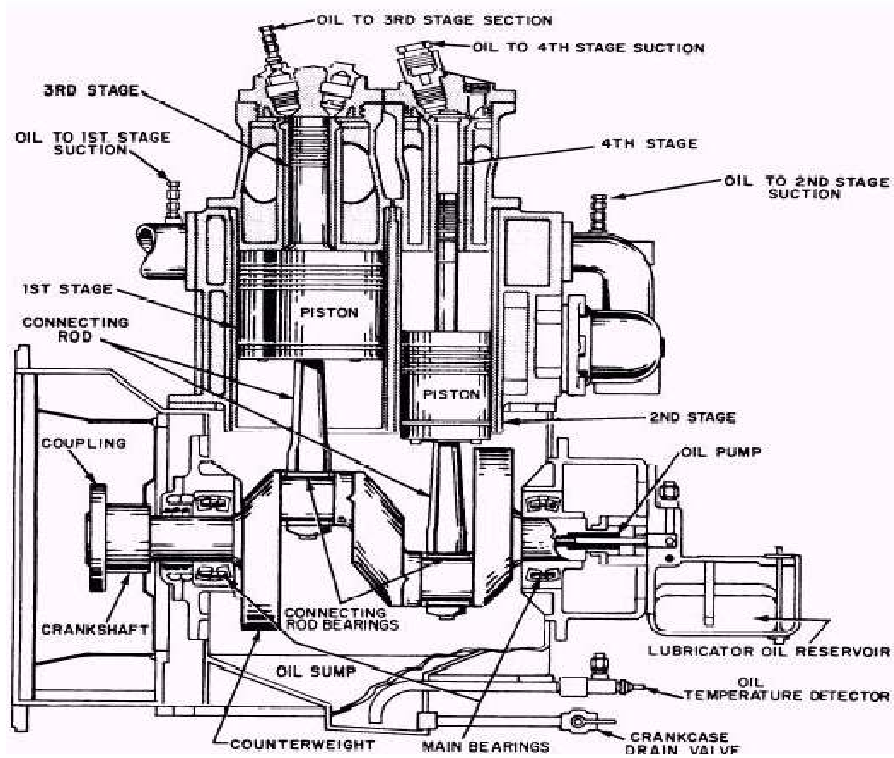


Fig. 1: High pressure compressor showing the lubrication system

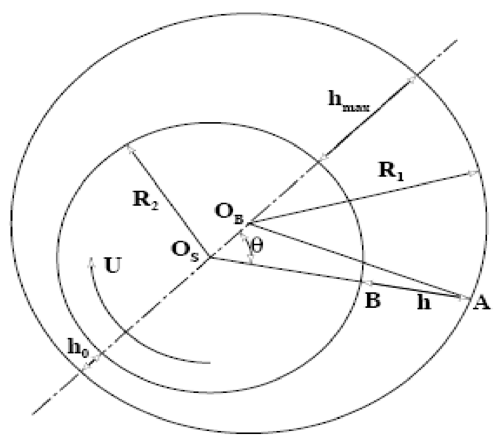


Fig. 2: Geometry of the journal bearing

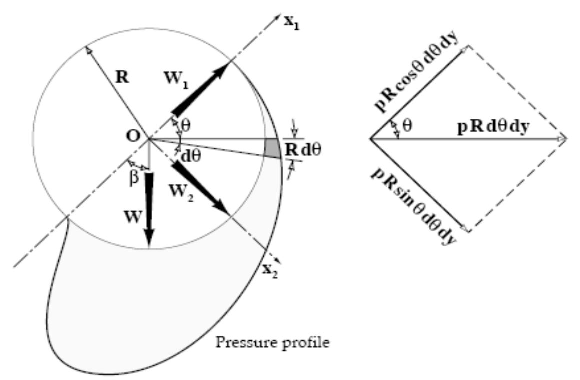


Fig. 3: Load components and pressure field acting in a journal bearing

MATERIALS AND METHODS

High pressure compressor cylinders are generally lubricated by an adjustable mechanical force-feed lubricator (Fig. 1). Oil is fed into each cylinder through separate lines from the cylinder lubricator. Each feed line has an oil flow indicator and a check valve located at the end of each line to prevent the compressed gas from forcing the oil back to the lubricator.

The bearing is lubricated by an oil pump driven from the compressor shaft. The pump moves oil from the reservoir (oil sump) in the base of the compressor, delivers it through filters and oil cooler. From the cooler the oil is distributed to the top of each bearing, to spray nozzles for other mechanical components. The crank shaft is drilled in a way that oil fed to the main bearing is picked up from the main bearing journal and carried to the crank journal.

The connecting rods contain passages that conduct lubricating oil from the crank bearings up to the piston pin bushings. As oil is forced out from the various bearings, it drips back into the oil sump (in the base of the compressor) and is re-circulated. Oil from the outboard bearings is carried back to the sump by drain lines (Lubrication Systems. 2013) (Fig. 2 and 3).

From the EXXOCLUB SLG 190 the operating pressure and the associated viscosity is given below.

The journal bearing properties used for this study are shown in Table 1.

Table 1: Journal bearing properties

Journal bearing length	25.40 mm
Journal bearing diameter	63.50 mm
Radial clearance	35.56 µm
Angular velocity	2000 rpm
Eccentricity ratio ε	0.10

Table 2: Viscosity of EXXOCOLUB SLG 190 at different operating pressure

Pressure (psig)	Percentage of initial viscosity (%)	Viscosity (pas)
1000	84	0.1084
2000	68	0.0877
3000	60	0.0774
4000	55	0.0710
5000	50	0.0645

Table 2 highlights the viscosity of EXXOCOLUB SLG 190 at different operating pressures. Source: EXXOCOLUB SLG 190 data sheet (2010).

Methodology: According to Yang *et al*. (2010), the journal bearing of a reciprocating compressor can be analyzed using the equation below:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(h^3\frac{\partial p}{\partial y}\right)=12\mu\eta\frac{dh}{dx} \tag{1}$$

where,
h = Oil film thickness in m
η = The dynamic viscosity of oil in N.s/m^2
u = Linear velocity of shaft in m/s

The pressure distribution is obtained with the aid of the finite element method. The equation for pressure in the finite element form is given below.

Assuming pressure gradient across the y axis to be zero, the Reynolds Eq. (1) reduces to:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=12\mu\eta\frac{dh}{dx} \tag{2}$$

In this analysis, due to the effect of vibration, a modification of the above equation is presented as:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=12\mu\eta\frac{dh}{dx}+12\eta(w_2-w_1) \tag{3}$$

where,
w_2 = The velocity at which the top of the column moves down
w_1 = The velocity at which the bottom of the column of lubricant moves up

The residual, $R(x,p)$, of the governing equation is first obtained by taking all terms on the right hand side to the left hand side to obtain:

$$R(x,p)\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)-12\mu\eta\frac{dh}{dx} \tag{4}$$

Multiplying Eq. (3) by a weight function w_i and integrating over a typical element having nodes x_1 and x_2 results in:

$$\int_{x_1}^{x_2} w_i\left[\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)-12\mu\eta\frac{dh}{dx}\right]dx=0 \tag{5}$$

Integrating the first term of (5) result in (6), thus:

$$\int_{x_1}^{x_2} w_i\left[\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)\right]dx=\int_{x_1}^{x_2}-\frac{\partial w_i}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\left[w_i h^3\frac{\partial p}{\partial x}\right]_{x_1}^{x_n} \tag{6}$$

Substituting (6) into (5) gives (7), as:

$$\int_{x_1}^{x_2}-\frac{\partial w_i}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\left[w_i h^3\frac{\partial p}{\partial x}\right]_{x_1}^{x_n}+\int_{x_1}^{x_n}-12\mu\eta\frac{dh}{dx}w_i=0 \tag{7}$$

In an instance of vibration resulting in vertical flow across the film Eq. (2) changes to (8):

$$\int_{x_1}^{x_2}-\frac{\partial w_i}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\left[w_i h^3\frac{\partial p}{\partial x}\right]_{x_1}^{x_n}+\int_{x_1}^{x_n}-12\mu\eta\frac{dh}{dx}w_i-\int_{x_1}^{x_2}12\eta(w_2-w_1)w_i=0 \tag{8}$$

A trail function is assume of the form:

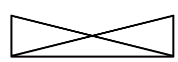
$$p=ax(L-x)$$

which satisfies the boundary conditions:

$$0<x<L \; and \; p(0)=0, P(L)=0$$

$H_1(x)$ $H_2(x)$

1.0



The shape functions has the following properties:

- The shape function associated with node i has a unit value at node i and vanishes at the other node:

  $H_1(x_i)=1, H_1(x_{i+1})=0, H_2(x_i)=0,$
  $H_2(x_{i+1})=1$

- The sum of all shape functions is unity:

  $\sum_{i=1}^{2}H_i(x)=1$

Since the Garlekin method is used, the test function comes from the chosen trial function. That is:

$$w_i=\frac{\partial p}{\partial a_i}$$

If the shape is used, the elemental matrix of the i^{th} element is:

$$\int_{x_i}^{x_{i+1}} -h^3 \begin{Bmatrix} H'_1 \\ H'_2 \end{Bmatrix} [H'_1 H'_2]$$
$$= -\frac{h^3}{h_i}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \quad k_1 = \begin{pmatrix} k_{i1} & k_{i2} \\ k_{i3} & k_{i4} \end{pmatrix}$$

where, i is the element under consideration.

The finite element analysis leads to the formation of elemental matrix and vector which are further assembled. The assembled matrix is iterated by the Gauss Seidel's method and boundary conditions are applied to the system of equations. The size of the system matrix is equal to the total number of degrees of freedom in the system.

Fsol (finite element solution) for nodal pressures $= kk\backslash ff$.
where,

$$kk = \begin{pmatrix} k_{11} & k_{12} & 0 & 0 & 0 & 0 \\ k_{13} & k_{14}+k_{21} & k_{22} & 0 & 0 & 0 \\ 0 & k_{23} & k_{24}+k_{31} & k_{32} & 0 & 0 \\ 0 & 0 & k_{33} & k_{34}+k_{41} & k_{42} & 0 \\ 0 & 0 & 0 & k_{43} & k_{44}+k_{51} & k_{52} \\ 0 & 0 & 0 & 0 & k_{53} & k_{54} \end{pmatrix}$$

$$The\ elemental\ vector\ (F) =$$
$$\int_{x_i}^{x_{i+1}} 12\mu\eta \frac{dh}{dx}\begin{Bmatrix} H_1 \\ H_2 \end{Bmatrix} dx = 12\mu\eta \frac{dh}{dx}\frac{h_i}{2}\begin{bmatrix} 1 \\ 1 \end{bmatrix}$$

The assembled vector ff is given by:

$$ff = \begin{pmatrix} f_{11} \\ f_{12}+f_{21} \\ f_{22}+f_{31} \\ f_{32}+f_{41} \\ f_{42}+f_{51} \\ f_{52} \end{pmatrix}$$

$$where\ f_i = \begin{pmatrix} f_{i1} \\ f_{i2} \end{pmatrix}$$

The load capacity is obtained by integrating the pressure distribution over the bearing area. This is given as:

$$w = \int_0^L \left(\int_0^B p dx\right) dy$$

But since the infinitely long bearing approximation is used for the analysis, the load capacity per unit length will be obtained by the mathematical expression given below:

$$\int_0^B p dx$$

RESULTS AND DISCUSSION

The pressure profile shown in Fig. 4a is of the journal bearing operating at the different operating pressures 0 psi, 1000 psi (6.895×10^6 N/m^2), 2000 psi (1.379×10^7 N/m^2), 3000 psi (2.068×10^7 N/m^2).

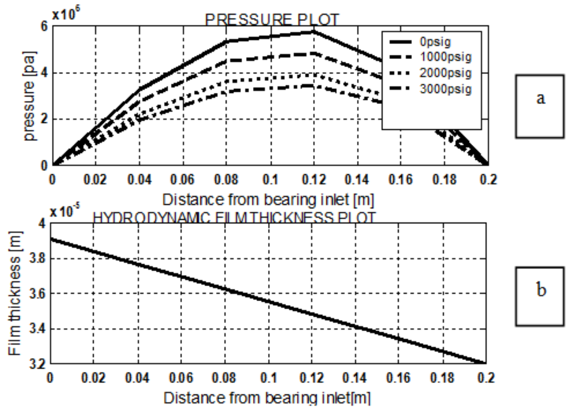


Fig. 4: A plot of pressure and film thickness across the unwrapped bearing length

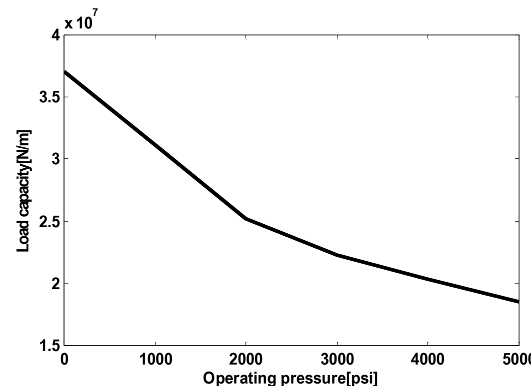


Fig. 5: A plot of load capacity against operating pressure

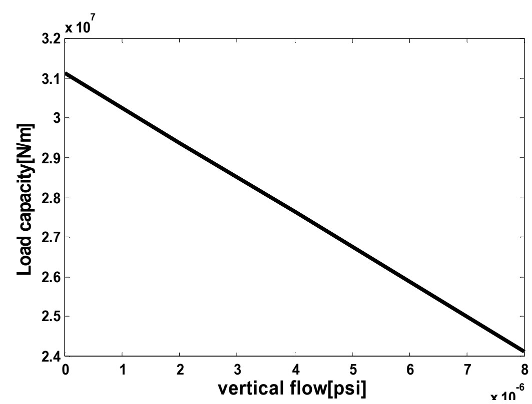


Fig. 6: A plot of load capacity against vertical flow of lubricant

Figure 4b is the film thickness across the film. It is observed that the pressure profile flattens out in an increasing form as the operating pressure increases.

As shown in Fig. 5, the load capacity of the journal bearing decreases as the operating pressure of the reciprocating compressor increases. As the operating pressure increases, more methane dissolves into the EXXOCOLUB SLG 190 lubricant, leading to viscosity thinning of the lubricant.

From the Fig. 5, it is also observed that the load capacity per unit length at 5000 psi (3.447×10^7 N/m^2) is reduced by 50% of the load capacity per unit length at ambient pressure. It is therefore imperative for the effective performance of the journal bearing, to take cognizance of the operating pressure of the compressor in order to sustain required load upon the bearing without failure of the bearing.

From Fig. 6, it is evident that as the vertical flow of lubricant increases, the load capacity decreases for a given operating pressure.

The vertical flow is due to wear of parts such as, pistons, cylinders which leads to increased clearance in these parts. Vibrations arising from the frame, rod drop, rod run out, cross head vibration (as the clearance between the cross head and the babbitt surface increases) leads to vertical flow in the journal bearing lubricant.

CONCLUSION

From the results of this study, it is seen that the operating pressure has a significant effect on the performance of the journal bearing of a reciprocating compressor. It increases the compressed gas (in this case methane) solubility in the lubricant which results in viscosity thinning of the lubricant and reduction in the pressure profile and load capacity of the bearing. It can be deduced from the analysis that, an excessive operating pressure can lead to bearing failure. Vibration in the system should also be taken into consideration as it can also lead to bearing failure. From this analysis, it can be seen that vibration leads to vertical flow and the load capacity decreases as the vertical flow (vibration) increases.

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