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ABSTRACT

CFCs, HCFCs are stipulated to phase out, and other alternative refrigerants like R134a, are also constrained as a result from their high GWP values. CO_2 as a natural working fluid now is used in reciprocating compressors due to its uniquely beneficial properties (ozone depletion potential, ODP=0; global warming potential, GWP =1). However, one remarkable characteristics of the CO_2 reciprocating compressor operated in trans-critical region is the high operating pressure, which may impose high pressure loads on the connecting–rod big end bearing, further on the crankshaft journal bearings through the slider-crank mechanism. The work presented here is trying to study the friction losses of crankshaft journal bearings in a CO_2 reciprocating compressor, by using a multiple regression method developed by Stachowiak and Batchelor(2001). The bearing performances depend mostly on the oil viscosity. In this case, the solubility of the CO_2 into the lubricating oil is taken into consideration, which affects the oil viscosity.

1. INTRODUCTION

Relatively high operating pressure is one of the characteristics of carbon dioxide reciprocating compressors, compared with compressors using other refrigerants as working fluids. High pressure may impose high force loads on the friction conjunctions in the compressor. Hence, analysis of the friction losses in a CO_2 reciprocating compressor should be taken into consideration. The places in which friction losses mainly occur in a reciprocating compressor could be divided into three parts, between the piston and the cylinder, crankshaft journal bearings and connecting-rod big end journal bearings. The performances and reliabilities of both crankshaft journal bearings and connecting-rod big end bearings have much influence on the performance of the whole machine. Numerous researches have been concerned on this topic for decades of years.

Paranjpe (1994) studied on the performance of a steadily loaded journal bearing. In his work, a mass conserving cavitation, coupled with a full 3-D energy equations were solved, by considering the heat conduction in the bush, oil mixing in the groove. And an optimum clearance was found, with which the largest minimum oil film thickness could be obtained.

Khonsari (1996) generalized the thermohydrodynamic analysis, and two temperature-rise parameters were found to be crucial for the temperature filed. The first one incorporates oil properties, and the second one is a function of the shaft velocity with specified lubricants properties.

Wilson (1998) provided a framework for the thermohydrodynamic(THD) lubrication analysis. THD analysis is very complicated due to the multiplicity of the heat transfer paths from the sources to sinks. By making use of a set of non-dimensionless parameters used to measure the relative importance of different heat transfer paths, the heat transfer paths are simplified. Method of lumped variables, and consequently an equivalent resistance network are applied. This framework may be useful in the interpretation of some THD analysis results obtained by other methods.

In actual reciprocating machinery (engines, reciprocating compressors), both the crankshaft journal bearings and connect rod big end journal bearings are dynamically loaded. Paranjpe (1995) proposed that the key points of the transformation from steadily loaded analysis to dynamically loaded analysis are different time scales of thermal effects for diverse components and the moving grids. The time scale of thermal effects in the oil film is the same order of the magnitude of dynamic load. However, time scale for the thermal effects of shaft is orders of magnitude of the dynamic load. Thus, the temperature of shaft journal in the analysis could be assumed to be a constant.

The analysis of both steadily and dynamically loaded journal bearings require the solution of the Reynolds equation, coupled with energy equation and heat transfer equation. The procedure to analyze the dynamically loaded journal bearings is much more complicated. In this case, Booker (1965) developed a mobility method. Goenka (1984) used a curve fitting in this mobility method. Although, the curve fitting method is conceptually only applicable for ideal bearings, it truly gets satisfactory results for non-ideal bearings. The work presented here is trying to use another method to study the performance of journal bearings, developed by Stachowiak and Batchelor (2001), which is so-called multiple regression method. The friction loss of the crankshaft journal bearings is the main concern.

2. ANALYSIS OF THE MAIN JOURNAL BEARINGS

2.1 Load

Forces and moments of the crank-connecting rod system are analyzed to determine the force loads on the crankshaft bearings, crank journal bearings and piston-cylinder assembly. Due to the direction of the friction force between the piston ring and cylinder wall changing with the direction of piston velocity, the analysis of forces and moments of the crank-connecting rod system is proceeded in re-expansion process, suction process, compression process and discharge process respectively. The piston velocity has the same direction in re-expansion and suction process, as well as in the compression and discharge process. Figure 1 is a schema of the forces and moments in the compressor.

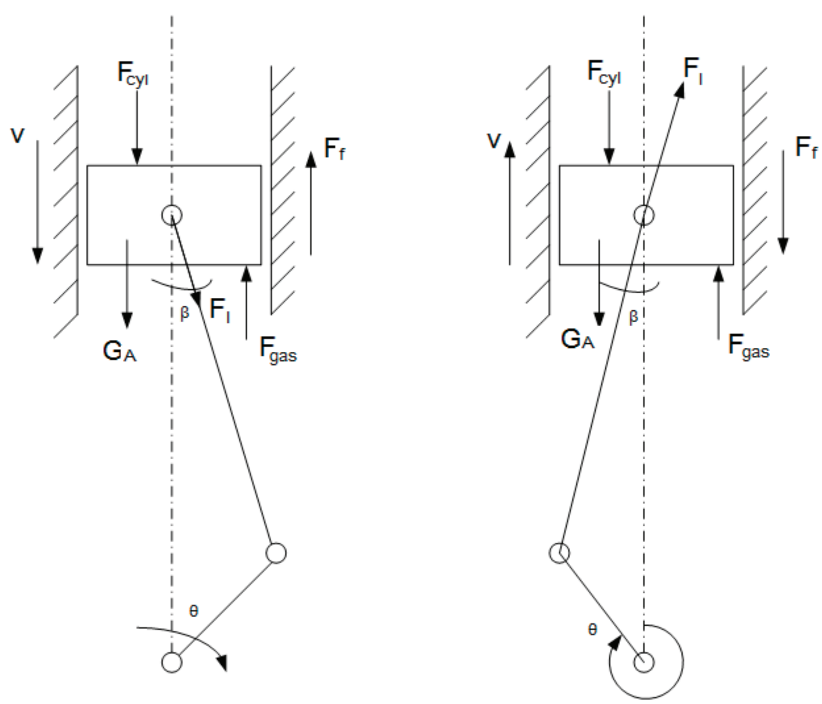

Figure 1: Force and moment analysis of the crank-connecting rod system

During the re-expansion and suction processes:

$$F_l = (m_A a(\theta) - G_A - F_{cyl} + F_f + F_{gas})/cos\beta \quad (1)$$

$$sin\beta = \lambda sin\theta, \qquad cos\beta = \sqrt{1-(\lambda sin\theta)^2} \quad (2)$$

During the compression and discharge processes:

$$F_l = (m_A a(\theta) - G_A - F_{cyl} - F_f + F_{gas})/(-cos\beta) \quad (3)$$

$$sin\beta = -\lambda sin\theta, \qquad cos\beta = \sqrt{1-(\lambda sin\theta)^2} \quad (4)$$

Where, F_l is the connecting rod force; $a(\theta)$ is the piston acceleration, the downward direction is set to be positive; m_A is the piston mass; G_A is the force acting on the piston due to gravity; F_{cyl} is the gas force in the cylinder; F_{gas} is the force caused by the gas pressure under the piston; F_f is the friction force between the piston and the cylinder liner; β is the angle between the axis of connecting rod and the center line of cylinder; λ is the ratio of length of crank to that of the connecting rod.

For the crankshaft system, it is assumed that the crank shaft is rotating at a constant speed. Based on the force and momentum balances of the whole crankshaft system, force loads on the crankshaft bearing D and E (in Figure 3), are determined respectively. The corresponding force analysis is depicted in Figure 2. The dimension of the crankshaft is indicated in Figure 3. Two situations are considered: (1) cylinder1 upstroke while cylinder2 down-stroke; (2) cylinder1 down-stroke while cylinder2 upstroke.

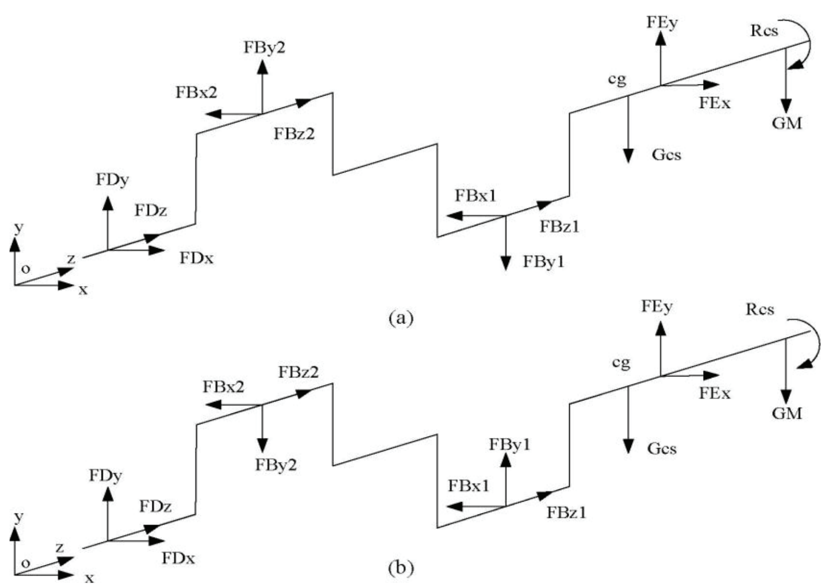

Figure 2: Force and moment analysis of the crankshaft
(a) Left cylinder upstroke, right cylinder down-stroke
(b) Left cylinder down-stroke, right cylinder upstroke

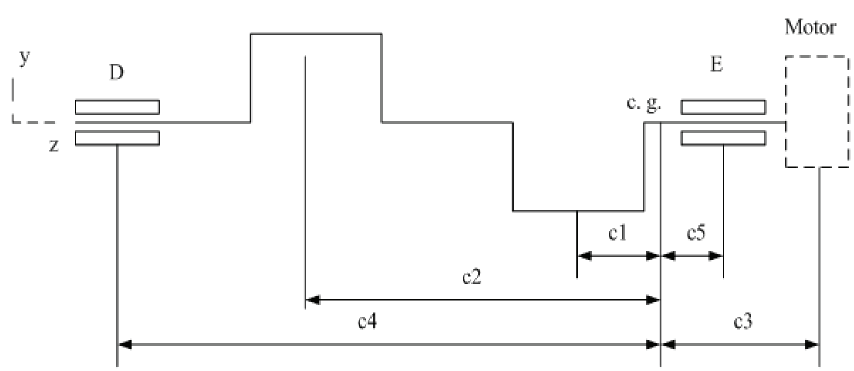

Figure 3: Crankshaft illustrations with dimensions

The force loads on the journal bearing D and E both in the x-direction and y-direction, are determined by the formula as listed in equation (5) and (6).

$$F_{(D,E)X} = A(B * F_{BX2} + C * F_{BX1}) \quad (5)$$

$$F_{(D,E)y} = A(B * F_{By1} + C * F_{By2} + D * G_{CS} + E * G_M) \quad (6)$$

Where, F_{BX1}, F_{BX2}, F_{By1}, F_{By2} are force loads on the connect big end bearing in x- and y- direction in cylinder 1 and cylinder 2. G_{CS} and G_M are forces due to gravity acting on the crankshaft and motor, respectively. The coefficients are listed as in Table 1.

Table 1: Coefficients in the formula of journal bearing force loads

Coefficients	A	B	C	D	E
		Cylinder 1 upstroke while cylinder 2 down-stroke			
F_{DX}	$\frac{1}{c_4 + c_5}$	$c_5 + c_2$	$c_5 + c_1$	-	-
F_{EX}		$c_4 - c_2$	$c_4 - c_1$	-	-
F_{Dy}		$c_5 + c_1$	$-(c_5 + c_2)$	c_5	$c_5 - c_3$
F_{Ey}		$c_4 - c_1$	$c_2 - c_4$	c_4	$c_4 + c_3$
		Cylinder 1 down-stroke while cylinder 2 upstroke			
F_{DX}	$\frac{1}{c_4 + c_5}$	$c_5 + c_2$	$c_5 + c_1$	-	-
F_{EX}		$c_4 - c_2$	$c_4 - c_1$	-	-
F_{Dy}		$c_5 + c_2$	$-(c_5 + c_1)$	c_5	$c_5 - c_3$
F_{Ey}		$c_4 - c_2$	$-(c_4 - c_1)$	c_4	$c_4 + c_3$

The coefficients c_m $(m = 1,2,3,4,5)$ in the Table 1 represent the dimensions indicated in the Figure 3, where, c_1 is the distance between the gravity center (c.g.) of crankshaft and cylinder1 centerline; c_2, the distance between the c.g. and cylinder2 centerline; c_3, the distance between the c.g. and motor; c_4, the distance between the c.g. and the center plane of the bearing D; c_5, the distance between the c.g. and the center plane of the bearing E.

2.2 Oil Viscosity

The viscosity of the oil decreases with a rise in oil temperature, which may lead to a failure in the journal bearing. Besides, the solubility of CO_2 into the lubricant oil would also reduce the oil viscosity. Li et al. (2000, 2002) evaluated lubricants for the CO_2 refrigerant. Compared to other oils (mineral oils, PAOs, PAGs, POEs), AN(alkyl naphthalene) oil had improved solvency, better stability and less affinity for water. AN oil showed miscibility with CO_2 at low concentrations and was stable in CO_2 environment. Seeton et al. (2000) researched on the solubility, viscosity, boundary lubrication and miscibility of CO_2 and synthetic lubricants. A correlation between the temperature and absolute viscosity was obtained under the condition of 90% lubricant and10% refrigerant mixtures. An AN oil of a certain grade is used in the compressor.

The oil temperature in the journal bearing is generally in the range of 45°C~65°C, therefore, it is reasonable to assume the effective oil temperature to be 55°C. With this temperature and under the working condition (suction pressure is 4.18MPa), the solubility of CO_2 into the oil could be computed as 15%. Li and Rajewski (2000) presented a graph, which indicates the relation between the oil kinematic viscosity and temperature at different solubilities (0%, 10%, and 20%). Thus, by interpolation, the oil kinematic viscosity with solubility of 15% at 55°C could be determined. Consequently, the dynamic viscosity can be obtained as follows,

$$\eta = \rho \upsilon \quad (7)$$

Where, η is the dynamic viscosity, in Pa s; ν is the kinematic viscosity, in m^2/s; ρ is the density of the oil, in kg/m^3. The units of the kinematic viscosity, m^2/s, equals to $10^6 cs$, which is often used in industry. The density of the oil is a function of the oil temperature.

2.3 Multiple Regression Method

Basically, the power loss of journal bearings could be calculated by solving the Navier-Stokes and continuity equations. For the hydrodynamic lubrication in the crankshaft journal bearings analyzed in this work, the fluid properties assumed to be constant, and the Navier-Stokes equation can be simplified into the standard reduced form of Reynolds equation, as follows,

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(h^3\frac{\partial p}{\partial y}\right)=12u\eta_0\frac{\partial h}{\partial x} \tag{8}$$

Where, h is the oil film thickness, in m; u is the shaft linear velocity, in m/s; η_0 is the dynamic viscosity, in $N\cdot s/m^2$; x, y indicates the journal circumferential direction and axial direction, respectively. However, solving this partial differential equation implies a complicated procedure and numerous numerical works. Here, the friction power loss of the crankshaft bearing is determined by using a multiple regression method, put forward by Stachowiak and Batchelor(2001). This multiple regression method is an empirical formula derived from theoretical data of hundreds of bearings. This regression formula is described like,

$$H=3.9307\cdot 10^3\cdot v_{1,oil}^{-0,706}\cdot v_{2,oil}^{1.577}\cdot L^{0.477}\cdot D_{journal}^{2.240}\cdot N_j^{1.287}\cdot c^{0.249}\cdot T_s^{-0.204}\cdot(1+lnW^*)^{1.324} \tag{9}$$

Where $v_{1,oil}, v_{2,oil}$ are oil kinematic viscosity at 37.8℃ and 93.3℃, respectively, in cS; L is the axial length of bearing, in m; $D_{journal}$ is the diameter of the crank shaft journal, in m; N_j is the rotational speed of the crank shaft journal, in rps; c is the bearing clearance, in m; T_s is the lubricant supply temperature, in F; W^* is the dimensionless load capacity, which is obtained by:

$$W^*=W\cdot c^2/(\eta\cdot U\cdot L\cdot R^2) \tag{10}$$

Where W is the total force load acted on the bearing, in N; U is the circumferential velocity, in m/s; R is the journal radius, in m; η is the oil dynamic viscosity, in $Pa\cdot s$. The total force load is defined as the square root of the sum of the squares of force components in three directions.

$$W=\sqrt{F_x^2+F_y^2+F_z^2} \tag{11}$$

In which, F_x, F_y, F_z are the load components in x, y, z direction respectively.

3. RESULTS AND DISCUSSIONS

The CO_2 reciprocating compressor is tested at several working conditions with suction pressure of 4.18MPa, suction temperature of 10.5℃, and discharge pressure of 7.57MPa, 8.26MPa, 9.67MPa, 12.41MPa, 13.87MPa. With the assumed oil temperature of 55℃ and solubility of 15%, the relative error of the compressor input power by this simulation model compared with experimental data, is in 0.05%~4.3%, compression work in 0.01%~0.42%. The working condition with discharge pressure of 7.57MPa is chosen as an example to determine the friction loss of the crankshaft journal bearings.

Both the force loads acting in x-direction and y-direction on the crankshaft journal bearings are calculated by analyzing the forces and moments of the crank-connecting rod system before-mentioned. This procedure is dealt with in detail in a future paper. With the calculated force loads and constant viscosity, the friction power loss of the crankshaft journal bearings under this working condition is obtained, 173W.

The cyclic period of the crank shaft is divided into 36 equal parts. In every 10 degrees, the journal bearing is assumed to be acted on by a steadily loaded force. Thus, the friction power loss during every equal 10 degree parts is obtained by using the multiple regression method. Afterwards, friction losses of 36 equal parts are summed by applying the Simpson method, in this way, the friction loss in a cyclic period is reached.

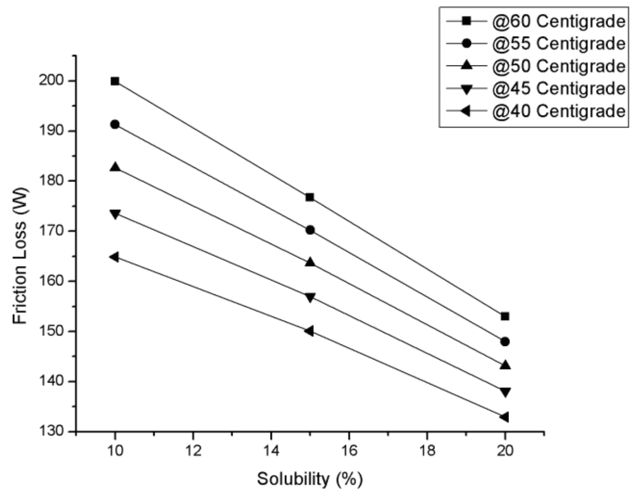

Figure 4: Solubility-Friction loss relation at different temperatures
(suction pressure, 4.18MPa; discharge pressure, 7.57MPa)

Figure 4 shows the relation between the solubility and friction power loss at different oil temperatures under same force loads. The friction loss increases with a rise in the oil temperature at a constant solubility, because of the oil dynamic viscosity reduction by the temperature rise.

CO_2 has a much smaller viscosity, compared with lubricant oil. When the more CO_2 is dissolved into the oil, the smaller the oil viscosity would be, leading to the worse lubricating condition in corresponding regions. Consequently friction loss increases due to the rise in solubility. However, it seems strange here, that the friction loss decreases with a rise in solubility at a constant temperature. According to the multiple regression method used in this work, the kinematic viscosity at 37.8℃ and 93.3℃ will also change when the solubility changes. And the value of the kinematic viscosity depends on the kind of the lubricating oil. Therefore, it is not easy to conclude that the friction loss will surely decrease or increase with change in the solubility. Different oils may have different results.

4. CONCLUSIONS

A full analysis of performances of the journal bearings requires the solution of Reynolds equation, coupled with energy equation, which is mainly used to determine the temperature distribution in the lubrication region. Solving these coupled partial differential equations is very complicated. Instead, a multiple regression method is used in this work, to determine the friction power loss. This empirical formula is derived based on study of several hundreds of bearings.

Both the solubility of CO_2 into the oil and the oil temperature have much influence on the oil viscosity, further, influence the friction power loss. Hence, it is essential to find a lubricating oil which has suitable solubility with CO_2 and proper temperature-viscosity relation.

NOMENCLATURE

$a(\theta)$	piston acceleration	(m/s^2)
A, B, C, D, E	coefficients	
c_1, c_2, c_3, c_4, c_5	dimensions	(m)
c	bearing clearance	(m)
$D_{journal}$	diameter of the crank shaft journal	(m)
F_{BX1}, F_{BX2}	force loads on the connect big end bearing. in x- direction in cylinder 1 and cylinder 2	(N)
F_{By1}, F_{By2}	force loads on the connect big end bearing in y- direction in cylinder 1 and cylinder 2.	(N)
F_{DX}, F_{EX}	force acting on bearing D and E in x direction	(N)
F_{Dy}, F_{Ey}	force acting on bearing D and E in y direction	(N)
F_{cyl}	gas force in the cylinder	(N)
F_f	friction force between the piston and the cylinder liner	(N)
F_{gas}	force caused by the gas pressure under the piston	(N)
F_l	connecting rod force	(N)
F_x, F_y, F_z	force load components in x, y, z direction	(N)
G_A	force acting on the piston due to gravity	(N)
G_{CS}, G_M	forces due to gravity acting on the crankshaft and motor	(N)
h	oil film thickness	(m)
H	bearing friction loss	(W)
L	axial length of bearing	(m)
m_A	piston mass	(kg)

N_j	rotational speed of the crank shaft journal	(rpm)
p	hydrodynamic pressure in the oil film	(Pa)
R	journal radius	(m)
T_s	lubricant supply temperature	(F)
u	shaft linear velocity	(m/s)
U	circumferential velocity	(m/s)
W	total force load acting on the bearing	(N)
W^*	dimensionless load capacity	
x, y	circumferential direction and axial direction of journal	
β	angle between the axis of connecting rod and the center line of cylinder	(rad)
λ	ratio of crank length to the connecting rod length	
θ	crankshaft angle	(rad)
ρ	density	(kg/m^3)
ν	kinematic viscosity	(m^2/s)
$\upsilon_{1,oil}$, $\upsilon_{2,oil}$	kinematic viscosity at 37.8°C and 93.3°C	(cS)
η, η_0	Dynamic viscosity	(Pa s)

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