1990

Scroll Compressor: Thrust Bearing Design Under Laminar Conditions

S. S. Kulkarni
Copeland Corporation

SCROLL COMPRESSOR: THRUST BEARING DESIGN UNDER LAMINAR CONDITIONS

Sunil S. Kulkarni
Senior Project Engineer
Research and Development Department
Copeland Corporation
Sidney, Ohio, U.S.A.

ABSTRACT

The paper presents the results of a theoretical study of the scroll compressor orbiting thrust bearing. The boundary conditions have been derived for the orbiting motion at the bearing surfaces. A Reynolds equation under incompressible laminar condition with constant viscosity is derived from the Navier-Stokes equations. The effects of the squeeze film velocity is considered. The equation is solved by finite difference methods. For a given average film thickness and attitude angle of the scroll plate a pressure profile in the bearing film is generated. The resultant force and tilting moment developed in the bearing are then computed.

NOMENCLATURE

F = oil film restoring force

F_f = friction force

h = local film thickness

h_1, h_2, h_3, h_4 = film thickness at mean radius

p = pressure in clearance space

p_1 = pressure at the inner radius of the bearing

p_2 = pressure at the outer radius of the bearing

Q = lubricant flow

r = radial coordinate

R_1 = inside radius of bearing

R_2 = outside radius of bearing

R_B = radial coordinate with shaft center as reference

u = radial velocity component

v = tangential velocity component

w = squeeze velocity component (z direction)

w_s = scroll plate local squeeze velocity component

X, Y, Z = Cartesian coordinate system

ω = angular velocity

θ = tangential coordinate

μ = lubricant viscosity

ϕ = velocity phase angle

INTRODUCTION

Scroll compressors are being increasingly adopted in air conditioning and refrigeration applications. The compression in the scroll compressor is achieved through the orbiting motion of the lower scroll member. Gas pressure inside the scroll chambers (formed between the fixed upper scroll member and the orbiting lower scroll member) produces axial and tangential force components. The tangential component of the gas force creates an overturning moment on the orbiting scroll member. The thrust bearing in the compressor must be designed to support the axial gas force and at the same time resist the overturning moment. Since the inner diameter of the thrust bearing is determined by the crank-offset, the design problem left is to find the minimum outside diameter of the thrust bearing.

Morishita and Sugihara [1] had considered this specific problem by a modification of Rumbarger's theory [2]. In their work the restoring force and its location are essentially assumed. In this paper the Reynolds equation is solved for the orbiting thrust bearing to obtain the pressure developed in the oil film for a given film thickness distribution. The restoring force and moment are then computed numerically.

GOVERNING EQUATIONS

For the laminar flow and constant viscosity with the usual assumptions in lubrication theory, the Navier-Stokes equations (1) and (2) and the Continuity equation (3) for the orbiting thrust bearing in Figure 1 are:

$$\mu \frac{\partial^2 v}{\partial z^2} = \frac{1}{r}\frac{\partial p}{\partial \theta} \tag{1}$$

$$\mu \frac{\partial^2 u}{\partial z^2} = \frac{\partial p}{\partial r} \tag{2}$$

$$\frac{\partial (ru)}{\partial r} + \frac{\partial v}{\partial \theta} + r\frac{\partial w}{\partial z} = 0 \tag{3}$$

The boundary conditions for orbiting motion (refer to Figure 1) are:

$$w = w_s,\ v = r\omega,\ u = 0 \text{ at } z = 0$$

$$w = 0,\ v = R_B\omega\cos\phi,\ u = R_B\omega\sin\phi \text{ at } z = h \tag{4}$$

Integrating equations (1) and (2) with boundary conditions (4) yields for the v and u velocities:

$$v = \frac{z(z-h)}{2\mu r}\frac{\partial p}{\partial \theta} + \frac{1}{h}(zR_B\omega\cos\phi + r\omega(h-z)) \tag{5}$$

$$u = \frac{z(z-h)}{2\mu}\frac{\partial p}{\partial r} + zR_B\frac{\omega\sin\phi}{h} \tag{6}$$

The integrated form of the continuity equation for this case is:

$$\int_0^h \frac{\partial (ur)}{\partial r}dz + \int_0^h \frac{\partial v}{\partial \theta}dz + \int_0^h r\frac{\partial w}{\partial z}dz = 0 \tag{7}$$

Above, $v = v(z, \theta)$, $u = u(z, r)$ and since $h = h(\theta, r)$, with the aid of Leibnitz's rule, equation (7) can be written as follows:

$$\frac{\partial}{\partial r}\int_0^h ru\,dz + \frac{\partial}{\partial \theta}\int_0^h v\,dz - v(h,\theta)\frac{\partial h}{\partial \theta} - ru(h,r)\frac{\partial h}{\partial r} - rw(0) = 0 \qquad (8)$$

From equations (5) and (6):

$$v(h,\theta) = R_B\omega\cos\phi, \quad u(h,r) = R_B\omega\sin\phi \qquad (9)$$

Substitution of equations (4), (5), (6) and (9) into equation (8), result in a partial differential equation for the pressure distribution as follows:

$$\frac{\partial}{\partial r}\left(\frac{rh^3}{12\mu}\frac{\partial p}{\partial r}\right) + \frac{1}{r}\frac{\partial}{\partial \theta}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial \theta}\right)$$

$$= \frac{1}{2}\frac{\partial}{\partial \theta}(\omega\, rh + \omega\, R_B h\cos\phi) + \frac{\omega}{2}\frac{\partial}{\partial r}(rR_B\sin\phi\; h)$$

$$- R_B\omega\cos\phi\,\frac{\partial h}{\partial \theta} - rR_B\omega\sin\phi\,\frac{\partial h}{\partial r} - rw_s \qquad (10)$$

The above is the Reynolds equation with $p = p(r, \theta)$. The pressure boundary conditions are:

$$p = p_1 \text{ at } r = R_1, \quad p = p_2 \text{ at } r = R_2 \qquad (11)$$

EXPRESSIONS FOR BEARING PERFORMANCE

The restoring force and its location, using the Sommerfield boundary condition, can be calculated as follows:

$$F = \int_0^{2\pi}\int_{R_1}^{R_2} pr\,dr\,d\theta \qquad (12)$$

$$\overline{X} = \frac{1}{F}\int_0^{2\pi}\int_{R_1}^{R_2} r^2 p\sin(\theta-\pi)\,dr\,d\theta$$

$$\overline{Y} = \frac{1}{F}\int_0^{2\pi}\int_{R_1}^{R_2} r^2 p\cos(\theta-\pi)\,dr\,d\theta \qquad (13)$$

The oil flow and the friction force can be calculated as:

$$Q = \int_0^{2\pi}\int_0^{h} ur\,dz\,d\theta \qquad (14)$$

$$F_f = \mu\int_0^{2\pi}\int_{R_1}^{R_2} \frac{\partial v}{\partial z}\, r\,dr\,d\theta \qquad (15)$$

SOLUTION

The pressure field is obtained by integration of Reynolds equation (10) with boundary conditions (11) using the finite difference technique. Once the pressure is computed numerical integration is used to determine restoring force and its location.

NUMERICAL EXAMPLE

The following data will be assumed for the purposes of a sample calculation:

Orbit radius = 0.1 inch,

R_1 = 1.0 inch, R_2 = 2.5 inch, $\mu = 10^{-7} \dfrac{lb_f\text{-sec}}{inch^2}$,

P_1 = 1.0 PSI, P_2 = 0.0 PSI, Shaft rpm = 3600.0,

h_1 = 0.001 inch, h_2 = 0.0005 inch, h_3 = 0.001 inch,

h_4 = 0.0015 inch

The assumed film thickness profile nomenclature is shown in Figure 1.

The integration of the Reynolds equation results into a pressure distribution as shown in Figure 2. A maximum pressure of 135 PSI is produced in the film. This pressure distribution produces 378 lb_f separating force acting at (-.93,-1.29). This restoring force counteracts the axial gas force component and the overturning moment on the orbiting lower scroll member.

SUMMARY AND CONCLUSIONS

The Reynolds equation is derived and integrated for the orbiting thrust bearing using finite difference methods. The pressure generated in the oil film is used to calculate axial load carrying capacity and the tilting moment resistance capability of the thrust bearing. To support the required axial load and the tilting moment for a given inner radius of the bearing an outer radius can be estimated.

ACKNOWLEDGMENT

Discussions with, and help from B. Anderson, M. Bass, J-L Caillat, G. Fain, K. Topp, T. Ubach, and F. Wallis of Copeland Corporation, as well as comments from Dr. C. Maday of North Carolina State University, were very helpful in the development of this work. The author wishes to thank them all for their help.

REFERENCES

[1] Morishita E., Sugihara M., "Some Design Problems of Scroll Compressors," Bulletin of JSME, Vol. 29, No. 258, December 1986.

[2] Rumbarger, J.H., Machine Design, 34-4 (1962), 1972.

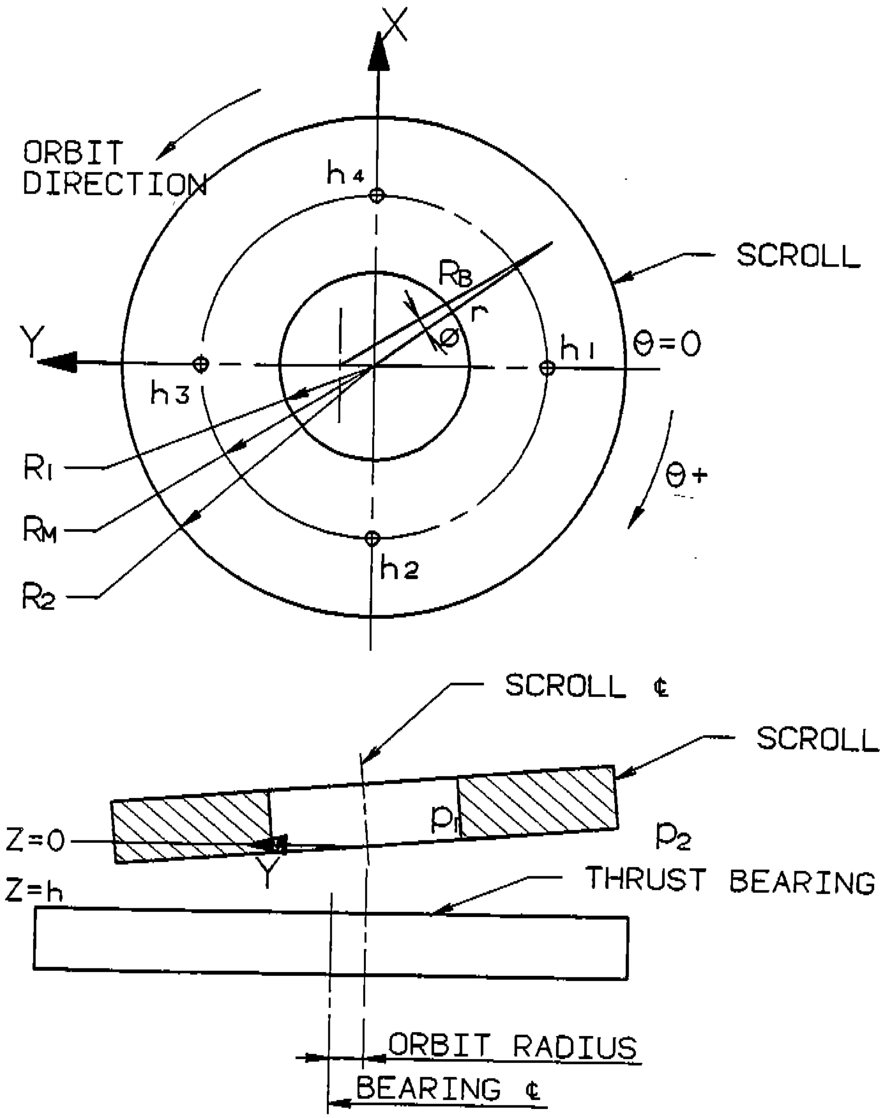
X
ORBIT
DIRECTION
h4
SCROLL
RB
r
Ø
Y
h1
θ=0
h3
R1
θ+
RM
h2
R2
SCROLL ¢
SCROLL
p1
Z=0
P2
Y
THRUST BEARING
Z=h
ORBIT RADIUS
BEARING ¢

FIGURE 1.

THRUST BEARING PRESSURE DISTRIBUTION

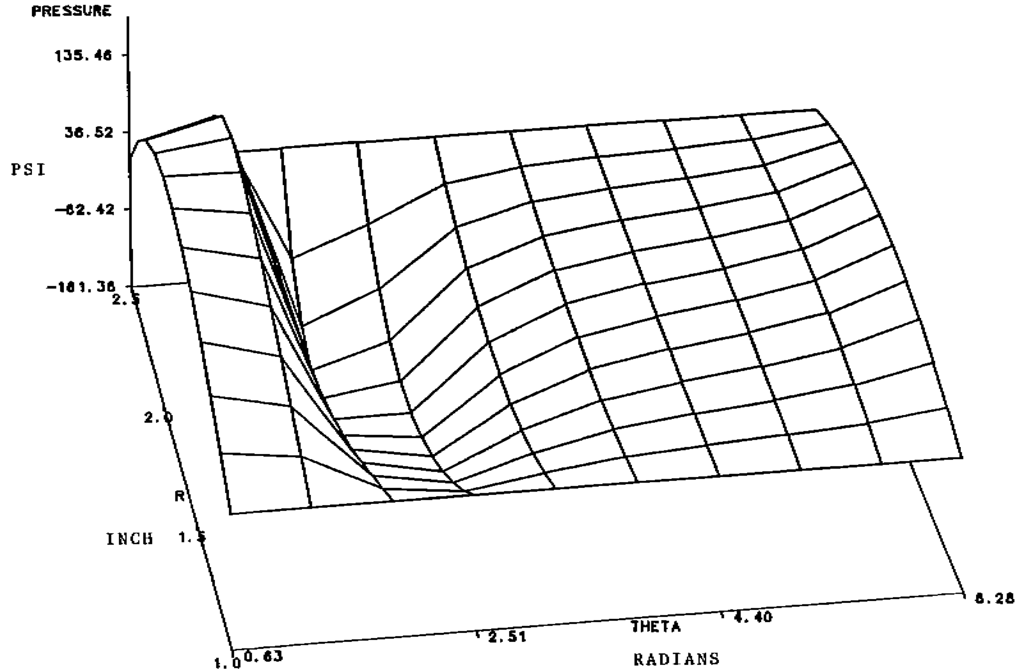

FIGURE 2