Leakage Performance of a Novel Turbomachinery Shaft Seal

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***Abstract*-Advanced sealing systems are needed to control parasitic leakage flows to achieve high turbine engine efficiency and low emissions. Typical extreme turbomachinery engine operating conditions when combined rotor excursions do not lend simple sealing solutions. This work presents an in-depth analysis of a novel robust yet simple sealing system that is capable of maintaining long life under high speed and high temperature operating conditions. The proposed seal design is actually a gas bearing that is carefully tailored, analyzed, and designed to function as a differential pressure seal. The design involves a simple rigid/semi-flexible seal ring that is attached to a stationary support plate via flexible metal cloth structure. The seal body is capable of moving under the effect of hydrodynamic lift force. Therefore, above a certain clearance limit, which ensures that asperity contact is avoided, the seal follows shaft excursions to avoid damaging hard rubs.**

***Keywords*- Seal, Reynolds Equation, turbomachine.**

I. INTRODUCTION

The invention of brush seal has become a breakthrough in sealing industry. The compliance of individual bristles gives brush seals the superiority over a labyrinth seal [1]. Ferguson [2] reports that leakage improvements an order of magnitude over labyrinth seals are possible in the brush seal applications. Similarly, Carlile [3], and Chupp [4] present experimental leakage data for brush seals illustrating the increase in leakage performance compared to annular and labyrinth seals. Parallel to the leakage improvements, there is also increase in thrust of gas turbine engine equipped with brush seals.

Contrary to the advantages, there are also reported problems for brush seal applications. When coupled with frictional effects, loads created by the leakage flow and pressure difference become the main sources for the challenges encountered in application of brush seals [1]. Bristle stiffening, hysteresis and pressure closure are the three main phenomena which affect the clearance and wear rates of the brush seal materials [1], [5].

Flexibility in sealing components has been extended with the introduction of metal cloth structure into sealing applications. It is a challenging issue for turbomachinery seal designers to maintain flexibility with extended service life in high-temperature, high pressure sealing applications. Integration of flexible metal cloth to the sealing applications provides a means to tolerate thermal misalignments and shows the promise to extend the service life at high temperatures [6]. Metal cloth seals are utilized in various applications in industrial gas and steam turbines [7], [8]. Application areas are generally the junctions between stationary components where the vibratory motion, thermal growth and misalignment are possible to occur. Up to now, the application of metal cloth seal remains limited between stationary components. In this study, a design of seal ring coupled with flexible metal cloth is developed to provide dynamic sealing around rotating shafts.

II. SOLUTION PROCEDURE

Nonlinear partial differential elliptic equations for gas lubricated journal bearings are originated from continuity and Navier-Stokes equations for compressible Newtonian fluids. The velocity gradient in Newtonian fluids is linearly related to the shearing stress. Almost all lubricating fluids are Newtonian.

Continuity and Navier-Stokes equations for compressible Newtonian fluids are expressed in vector form as:

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho V) = 0 \qquad (1)$$

$$\rho\left(\frac{\partial V}{\partial t} + V.\nabla V\right) = -\nabla P + \mu \nabla^2 V + F + (\mu + \mu^b)\nabla(\nabla.V) \qquad (2)$$

where μ and μ^b stand for dynamic viscosity and bulk viscosity respectively and **F** represents body forces. Mass density is ρ, absolute pressure is P, time is t, and velocity is $\mathbf{V}=\mathbf{i}u + \mathbf{j}v + \boldsymbol{k}w$. Last term on the right hand side of equation (2) is an addition for compressible fluids to the Navier Stokes equation of incompressible flow (3).

$$\rho\left(\frac{\partial V}{\partial t} + V.\nabla V\right) = -\nabla P + \mu \nabla^2 V + F \qquad (3)$$

Stokes makes an idealization for compressible fluids and assumes that pressure is independent of dilation. As a consequence, a relation between dynamic viscosity and bulk viscosity can be stated as follows:

$$2\mu + 3\mu^b = 0 \qquad (4)$$

Considering that ideal gas law holds for the lubricating gas film, density of gas can be expressed in terms of film pressure as follows:

$$PV = NRT \rightarrow PV = (M / M_a)RT \rightarrow \rho = \frac{M}{V} = P\left(\frac{M_a}{RT}\right) \qquad (5)$$

where P is the absolute pressure of the gas, V is the volume of the gas, N is the number of moles of gas, M is the mass of gas, M_a is the molar mass of the gas, R is the universal gas constant, T is the absolute temperature. Mass, volume and mole number of gas are related to the quantity of the gas. However, for a specific gas in a steady environment, M_a, R, and T are independent of gas quantity and can be taken as constant. Therefore, for a specific gas, such as air, a direct relationship between the density and the pressure can be defined. This relation reduces the number of unknowns in momentum equations to four, which are P, u, v and w. As a result, we have four equations including the continuity equation, and four unknowns and thus the problem is well posed in mathematical terms. Although the problem is well posed, nonlinear, second order, partial differential structure of Navier-Stokes equations makes it very difficult to get exact solutions except in a few special instances. A series of simplifications to these Navier-Stokes equations must be performed to solve the gas film lubrication problem. Simplification of the equations requires order of magnitude analysis on the variables. In order to start the order of magnitude analysis, we must first note that the vertical component of the velocity, w, is very small when compared with the horizontal components, u, v. The ratio w/u has the same magnitude as the inclination between the bearing surfaces. Due to the fact that there is a slight inclination between the bearing surfaces, vertical component of the velocity is expected to be very low and can be ignored from the equations. After removing the terms related to w from the explicit Navier-Stokes and continuity equations for 3 spatial dimensions can be written in a more compact and simple form as:

$$\rho\left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\partial P}{\partial x} + \mu\left(\frac{1}{3}\frac{\partial^2 v}{\partial x \partial y} + \frac{4}{3}\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) \qquad (6)$$

$$\rho\left(\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial P}{\partial y} + \mu\left(\frac{1}{3}\frac{\partial^2 u}{\partial x \partial y} + \frac{4}{3}\frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial z^2}\right) \qquad (7)$$

$$0 = -\frac{\partial P}{\partial z} + \mu\frac{1}{3}\left(\frac{\partial^2 u}{\partial x \partial z} + \frac{\partial^2 v}{\partial y \partial z}\right) \qquad (8)$$

.

For a typical finite journal bearing, the dimensions of bearing geometry can be scaled as follows:

$$L, B >> h \tag{9}$$

where L is the circumferential length and B is the width/axial length of bearing. A scaling between L and B for a journal bearing is not obvious because of the fact that infinitely many design combinations can be created varying L and B values. However, for a finite journal bearing, these two variables are almost in the same order of magnitude. An order of magnitude difference between L and B does not create a dilemma for the simplification of the momentum equations because of the fact that determining factor for the order of magnitude test is the ratio of fluid film gap distance, h, to the other variables. Magnitude of fluid film gap, h, is extremely small with respect to other dimensional variables. Typical scale factors of h with respect to other variables are:

$$\frac{h}{L} \sim 10^{-4}, \frac{h}{B} \sim 10^{-3} \tag{10}$$

If we apply order of magnitude analysis on the second order differential operators, following scaling relations are obtained. Thus, the momentum equations reduce to:

$$\rho\left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\partial P}{\partial x} + \mu\frac{\partial^2 u}{\partial z^2} \tag{11}$$

$$\rho\left(\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial P}{\partial y} + \mu\frac{\partial^2 v}{\partial z^2} \tag{12}$$

$$0 = -\frac{\partial P}{\partial z} + \mu\frac{1}{3}\left(\frac{\partial^2 u}{\partial x \partial z} + \frac{\partial^2 v}{\partial y \partial z}\right) \tag{13}$$

The equations above can be expressed in non-dimensional form by normalizing film velocities with respect to surface velocities along the corresponding directions, distances with respect to corresponding dimensions, pressure with respect to $\mu\frac{UL}{h^2}$ and time with respect to L/U such that:

$$u \sim U \quad \rightarrow \quad u^* = \frac{u}{U} \text{ and } u = Uu^*$$

$$v \sim U\frac{B}{L} \quad \rightarrow \quad v^* = \frac{v}{U} \cdot \frac{L}{B} \text{ and } v = \frac{UB}{L}v^*$$

$$x^* = \frac{x}{L} \quad \rightarrow \quad x = Lx^*$$

$$y^* = \frac{y}{B} \quad \rightarrow \quad y = By^*$$

$$P^* = \frac{P}{\mu\frac{UL}{h^2}} \quad \rightarrow \quad P = \mu\frac{UL}{h^2}P^*$$

$$t^* = \frac{t}{L/U} \quad \rightarrow \quad t = \frac{L}{U}t^*$$

Applying the normalization factors to Equations (11) to (13) the following equations in non-dimensional form are obtained:

$$\rho\frac{U^2}{L}\left(\frac{\partial u^*}{\partial t^*} + u^*\frac{\partial u^*}{\partial x^*} + v^*\frac{\partial u^*}{\partial y^*}\right) = \mu\frac{U}{h^2}\left(-\frac{\partial P^*}{\partial x^*} + \frac{\partial^2 u^*}{\partial z^{*2}}\right) \tag{14}$$

$$\rho\frac{U^2B}{L^2}\left(\frac{\partial v^*}{\partial t^*}+u^*\frac{\partial v^*}{\partial x^*}+v^*\frac{\partial v^*}{\partial y^*}\right)=-\mu\frac{UL}{h^2B}\frac{\partial P^*}{\partial y^*}+\mu\frac{UB}{h^2L}\frac{\partial^2 v^*}{\partial z^{*2}} \quad (15)$$

$$0=-\mu\frac{UL}{h^3}\frac{\partial P^*}{\partial z^*}+\frac{1}{3}\mu\frac{U}{Lh}\left(\frac{\partial^2 u^*}{\partial x^*\partial z^*}+\frac{\partial^2 v^*}{\partial y^*\partial z^*}\right) \quad (16)$$

In the final form above, the coefficients are still dimensional. To get rid of these dimensional terms, equations (14), (15) and (16) are divided respectively by $\mu\frac{U}{h^2}$, $\mu\frac{UB}{h^2L}$ and $\mu\frac{UL}{h^3}$. The resultant equations are fully non-dimensional:

$$\frac{\rho Uh}{\mu}\frac{h}{L}\left(\frac{\partial u^*}{\partial t^*}+u^*\frac{\partial u^*}{\partial x^*}+v^*\frac{\partial u^*}{\partial y^*}\right)=-\frac{\partial P^*}{\partial x^*}+\frac{\partial^2 u^*}{\partial z^{*2}} \quad (17)$$

$$\frac{\rho Uh}{\mu}\frac{h}{L}\left(\frac{\partial v^*}{\partial t^*}+u^*\frac{\partial v^*}{\partial x^*}+v^*\frac{\partial v^*}{\partial y^*}\right)=-\frac{L^2}{B^2}\frac{\partial P^*}{\partial y^*}+\frac{\partial^2 v^*}{\partial z^{*2}} \quad (18)$$

$$0=-\frac{\partial P^*}{\partial z^*}+\frac{1}{3}\frac{h^2}{L^2}\left(\frac{\partial^2 u^*}{\partial x^*\partial z^*}+\frac{\partial^2 v^*}{\partial y^*\partial z^*}\right) \quad (19)$$

Since the scaled Reynolds number is very small for a typical case with respect to unity, inertial terms on the left side of the equations (17) and (18) can be neglected. Also, $\frac{h^2}{L^2}$,which is located at the right hand side of (19) is very small, and the term multiplied with it can also be neglected. Therefore, the remaining simplified non-dimensional equations are:

$$0=-\frac{\partial P^*}{\partial x^*}+\frac{\partial^2 u^*}{\partial z^{*2}} \quad (20)$$

$$0=-\frac{L^2}{B^2}\frac{\partial P^*}{\partial y^*}+\frac{\partial^2 v^*}{\partial z^{*2}} \quad (21)$$

$$0=\frac{\partial P^*}{\partial z^*} \quad (22)$$

Finally, reduced Navier-Stokes equations can be expressed in dimensional form as:

$$0=-\frac{\partial P}{\partial x}+\mu\frac{\partial^2 u}{\partial z^2} \quad (23)$$

$$0=-\frac{\partial P}{\partial y}+\mu\frac{\partial^2 v}{\partial z^2} \quad (24)$$

$$0=\frac{\partial P}{\partial z} \quad (25)$$

One last step is needed to derive the governing equations for gas film lubrication. Velocities on the bearing boundaries are determined according to the results of flow regime analysis.

Assuming no-slip conditions and laminar flow, boundary conditions for equations (23) and (24) can be defined as:

$$0 = -\frac{\partial P}{\partial x} + \mu \frac{\partial^2 u}{\partial z^2}$$
$$z = 0,\ u = U \qquad (26)$$
$$z = h(x),\ u = 0$$

$$0 = -\frac{\partial P}{\partial y} + \mu \frac{\partial^2 v}{\partial z^2}$$
$$z = 0,\ v = 0 \qquad (27)$$
$$z = h(x),\ v = 0$$

By solving the equations in the sets (26) and (27) with the defined boundary conditions, velocity profiles in x and y directions are obtained as:

$$u = \frac{1}{2\mu}\frac{\partial P}{\partial x}\left(z^2 - zh\right) + U\left(1 - \frac{z}{h}\right) \qquad (28)$$

$$v = \frac{1}{2\mu}\frac{\partial P}{\partial y}\left(z^2 - zh\right) \qquad (29)$$

Now, we can plug the resulting equations into the continuity equation (3). Integration of continuity equation with respect to z yields:

$$\int_0^{h(x)} \left[\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) + \frac{\partial}{\partial z}(\rho w) \right] = 0 \qquad (30)$$

In steady state conditions, plugging equations (28) and (29) into (30) yields the Reynolds equations for our compressible flow:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\mu}\frac{\partial P}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h^3}{\mu}\frac{\partial P}{\partial y}\right) = 6U\frac{\partial(\rho h)}{\partial x} \qquad (31)$$

The above equations governs the characteristics of compressible, laminar, steady state continuum flow with no-slip assumption on the boundaries.

III. LEAKAGE ANALYSIS

For the presented seal system, leakage through seal-rotor clearance is nothing but the integration of axial velocity on the cross section of the channel gap.

It is also important to note that the density of gas film is not constant throughout the channel gap. Therefore, the amount of mass passing through the channel gap must be evaluated instead of the volumetric flow. Taking into account the compressibility of air, we can obtain the mass flow rate for leaking air as follows:

$$mf = \int_0^{2\pi r}\int_0^{h(\theta)} (\rho v) dz d\theta \qquad (32)$$

where ρ is the density of the air and dependent on the boundary location, and ν is the axial flow velocity of leaking air. Substituting Equation (28) and (29) into Equation (32), the following relation for the mass flow rate is obtained:

$$mf = \int_0^{2\pi r}\int_0^{h(\theta)} \rho \left[\frac{1}{2\mu}\frac{\partial P}{\partial y}\left(z^2 - zh(\theta)\right) \right] dz d\theta \qquad (33)$$

Pressure and density of air do not change along the z direction. Similarly, dynamic viscosity μ is constant throughout the fluid film. Film thickness, h varies only in the circumferential direction. Thus, we can easily take the integration with respect to z. As a result, Equation (33) reduces to:

$$mf = \int_{0}^{2\pi r} \frac{\rho\left(h(\theta)\right)^3}{12\mu}\frac{\partial P}{\partial y}d\theta \qquad (34)$$

III. RESULTS and DISCUSSION

Effective clearance is a measure of seal leakage performance that allows comparison of various seals. Note that the mounting clearance of our seal is roughly 2 mils, i.e. 50 µm. Obtained effective clearance values show that the leakage performance of the designed seal is comparable or better than brush seals. Figures (1) and (2) demonstrate the effect of eccentricity on the leakage rates of new seal where B is seal width.

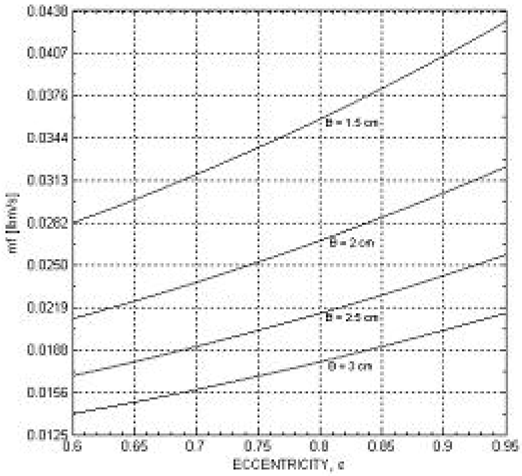

Figure (1): Mass flow rate vs. Eccentricity ratio [9]

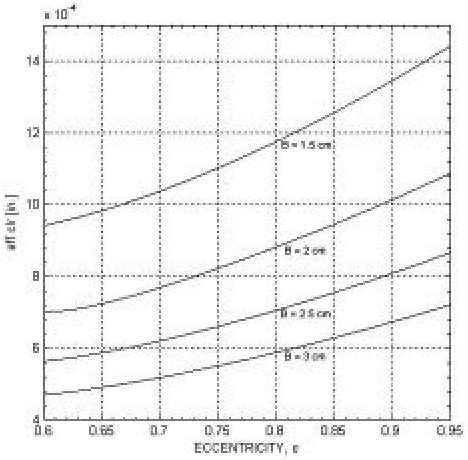

Figure (2): Effective clearance vs. Eccentricity ratio [9]

One drawback of the new seal is that leakage rate gradually increases with the increasing differential pressure. However, if axial space is available, this problem can be resolved increasing seal axial length. In brush seals, effective clearance level is stabilized by the effect of blow-down force [10]. Brush seals require higher clearances compared to the proposed seal design because of the fact that rubbing contact may create problems during transient conditions for critical locations like steam turbine applications. However, this is not a limitation for the proposed seal since a possible physical contact is avoided by the flexible cloth attachment.

IV. CONCLUSION

Overall, it is observed that under typical operating conditions all of the analyzed seal designs show satisfactory leakage performances that are similar to that of brush seals. Owing to the simple mechanical structure of the proposed seal design, manufacturing and maintenance costs are expected to be substantially low with respect to brush seals and other mechanical seals in the market. In addition to avoiding seal-rotor contact, during operation this new design provides longer seal life and higher operating temperature capabilities. As a result, the analyses indicate that the proposed new sealing system may provide a viable alternative to current engine seals.

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