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Computational Analysis of Pressure Variation of Squeeze Film Damper in Aero-Gas Turbine Engine

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1. Introduction

1.1. Background

A vibratory system generally, involves a means for storing potential energy alongside the kinetic energy, and a way by which energy is progressively lost (damper). The vibration of a system involves the transfer of its potential energy to kinetic energy and kinetic energy to potential energy and alternatively. Considering a damped system, some amount of energy is lost in each cycle of vibration which must be replaced by an external source if a state of steady vibration is to be preserved.

Damping, in physics, refers to restraining the vibratory motions, such as mechanical oscillations, noise, and so on, by the dissipation of energy. It could be classified into three categories: under-damped, critically-damped, and over-damped. Critical damping just averts vibration or is just adequate to allow the object to return to its rest position in the shortest time possible. Supplementary damping causes the system to be over damped, which may be appropriate, as in some door closers. The vibrations of an under damped system progressively taper off to naught.

Bearing Supports ensure damping by the oil film in the hydrodynamic bearings, also known as 'Squeeze Film Dampers (SFD)', designed specially by elastomeric dampers, or by Internal friction in the bearing and its housing structure assembly. A brief description about SFD is as follows:
1) The Influence of Inertia is Fundamental to the Operation of 'Squeeze Film Dampers'.
2) A Squeeze film damper is a journal bearing which is fitted around a rolling contact bearing.
3) These systems are used in Aircraft engines, where rolling bearing prevents bearing failure in the event of interruption to the lubricant supply.
4) The journal Bearing or Squeeze film damper is used to provide damping of the shaft vibrations, which are completely not affected by the presence of Rolling Bearings.
5) The stability of this bearing is analyzed by applying the Reynolds equation to a Journal bearing operating at a low eccentricity ratio.
6) If the limit of Journal Bearing stability against vibration is exceeded the squeeze damper will vibrate, negating its intended purpose.

A squeeze film damper could hence be defined as a filmy layer between the bearing and the housing that relaxes the bearing support to rise damping effectiveness. This film is in addition to the hydrodynamic film between the bearing and the shaft.

The advantages of using a SFD includes: (a) increases the stability, (b) decreases the rotor response, (c) escalates the separation margin between operating and critical speeds, (d) reduced the force transmission from rotor to the ground, (e) reduces pedestal vibration, and bearing wear, and (f) significantly used on high-speed machinery and/or with a flexible rotor.

1.2 Statement of Work

This project covers an analytic approach for squeeze-film damper performance prediction. Squeeze Film Damper (SFD) is used primarily in aircraft turbine engines to provide hydrodynamic damping. A brief study is undergone about the Squeeze Film damper and its nature. SFD have been used to overcome stability and vibration problems and also used to provide damping of the shaft vibrations. The damping in bearing supports is produced by the oil film in squeeze film dampers. The damping in the system is determined almost by the bearing damping properties. The pressure profile of the squeeze film damper is computationally analysed by using Finite Difference Method (FDM) in MATLAB. Discretization of the domain by solving Reynolds equation and prediction of pressure variation is analysed.

Squeeze Film Dampers (SFDs) are excellent means to upgrade rotor vibration amplitudes and to subdue the instabilities in rotor-bearing systems. SFD is not an off-the-shelf mechanical element but tailored to a particular rotor-bearing system as its design must satisfy a desired damping ratio. Industry requires well-engineered SFDs to lessen cost, maintenance, weight, and space although working for higher operating shaft speeds to surge power output. A manufacturer, as part of a business plan to develop and commercialize energy efficient aircraft gas turbine engines, supported a multiple–year project to test novel SFD design spaces.

2. Roots of Rotor Dynamics

Rotor dynamics is a specialized branch of applied mechanics concerned with the behaviour and diagnosis of rotating structures. The present trend in rotating equipment is toward increasing design speeds and this leads to an increase in operational problems from vibration. A thorough appreciation of vibration analysis will aid in the diagnosis of rotor dynamics problems. Rotor dynamics is a specialized branch of applied mechanical vibration concerned with the behaviour and diagnosis of rotating structures. It is commonly used to analyse the behaviour of structures ranging from jet engines and steam turbines to auto engines and computer disk storage. The purpose of rotordynamics is to establish methodologies of vibration reduction/ suppression. This enables design, appropriate operation and maintenance procedures of rotating machinery by elucidating causes of vibration from the viewpoints of excitation force and mechanisms of excitation, natural frequencies of modes excited by external influences on the rotor.

At its grass root level, rotor dynamics talks about one or more mechanical arrangements (rotor) supported by bearings and controlled by internal occurrences that rotate about a single axis. The supporting arrangement is known as a stator. With increase in the speed of rotation, the amplitude of vibration passes over a maximum called a critical speed. This amplitude is usually excited by the rotating structure's unbalance. The examples include engine balance, and so on. However, if the amplitude turns excessive at these critical speeds, then catastrophic failure occurs. In addition, turbo machinery develops instabilities related to its internal makeup that must be rectified, which is the significant concern of engineers who design large rotors or Aero Gas Turbine Engines.

Rotating machinery depends upon the structure of the mechanism involved in the method. Any errors lead to the increased excitation of the vibration signatures. Vibration behaviour of the machine caused by the unbalance is a significant aspect studied in detail and considered while designing a rotating machinery.

Hence, diminishing the rotational unbalance and other avoidable external forces are significant in reducing the forces that results in resonance. When the vibration reaches resonance, it generates a destructive energy that is the most major concern in the design.

3. Experimental Procedure

The Reynolds Equation is a widely known partial differential equation that governs the pressure distribution of shrill viscous fluid films. Reynolds delivered the foremost analytical proof that a viscous liquid can separate two sliding surfaces physically, using hydrodynamic pressure that results in low friction and negligible wear.

General Term

The general Reynolds equation (3-Dimension) is:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\cdot\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{h^3}{\mu}\cdot\frac{\partial p}{\partial y}\right)=6\left(U\frac{dh}{dx}+V\frac{dh}{dy}\right)+12\left(\omega_h-\omega_o\right)$$

Where:

p is fluid film pressure,

x and y corresponds to the bearing width and length coordinates,

z is fluid film thickness coordinate,

h is oil film thickness,

μ is fluid viscosity,

U is the Surface velocity,

$\frac{\partial p}{\partial x}, \frac{\partial p}{\partial y}$ are the pressure gradient acting along x and y directions.

The equation can either be used with consistent units or non-dimensionalized.

Assumptions:

- The fluid is Newtonian.
- Fluid viscous forces dominate over fluid inertia forces. This is the principal of the Reynolds number.
- Fluid body forces are negligible.
- The variation of pressure across the fluid film is negligibly small (i.e. $\frac{\partial p}{\partial z}=0$)
- The fluid film thickness is much less than the width and length and thus curvature effects are negligible. (i.e. $h \ll l$ and $h \ll \omega$)

Solution of Reynolds Equation

In general, numerical solution to Reynolds equation is more common, using numerical methods such as finite difference, or finite element. In certain simplified cases, however, analytical or approximate solutions can be obtained. For the case of rigid sphere on flat geometry, steady-state case and half-Sommerfeld cavitation boundary condition, the 2-D Reynolds equation can be solved analytically. This solution was proposed by a Nobel Prize winner Pyotr Kapitsa. Half-Sommerfeld boundary condition was shown to be inaccurate and this solution has to be used with care.

In 1916 Martin obtained a closed form solution for a minimum film thickness and pressure for a rigid cylinder and plane geometry. This solution is not accurate for the cases when the elastic deformation of the surfaces contributes considerably to the film thickness.

In this solution it was assumed that the pressure profile follows Hertz solution. The model is therefore accurate at high loads, when the hydrodynamic pressure tends to be close to the Hertz contact pressure.

Applications

The Reynolds equation is used in many applications such as:

- Ball bearings
- Air bearings
- Journal bearings
- Squeeze film dampers in aircraft gas turbines
- Human hip and knee joints
- Lubricated gear contacts

Reynolds Equation for Squeeze term

1) Load capacity under a time dependent film thickness, $\frac{\partial h}{\partial t} \neq 0$.
2) Squeeze film is a term denoting a hydrodynamic film that sustains a negative $\left(\frac{\partial h}{\partial t}\right)$. i.e., when opposing surfaces are being squeezed together.
3) An extremely useful characteristics of squeeze films is that they provide increased load capacity when a bearing is subjected to an abnormally high load.

4) The aspect of squeeze films is that the squeeze film force is always opposite in direction to the motion of either bearing surface.
5) This is a form of damping and squeeze film forces contribute to the vibrational stability of a bearing.
6) To analyse squeeze film forces, the term $\frac{\partial h}{\partial t}$ is kept in the Reynolds equation and is given precedence over the film geometry term $\frac{\partial h}{\partial x}$.

The Reynolds equation using the squeeze term could be expressed as:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\cdot\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{h^3}{\mu}\cdot\frac{\partial p}{\partial y}\right)=6\,U\frac{dh}{dx}+12\frac{dh}{dt}$$

With an assumption of an iso-viscous lubricant, the equation becomes,

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\cdot\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{h^3}{\mu}\cdot\frac{\partial p}{\partial y}\right)=12\,\mu\frac{dh}{dt}$$

It can be stated that the same basic analytical method is applied to the analysis of all hydrodynamic bearings regardless of their geometry. Initially the bearing geometry, h = f(x,y), must be defined and then substituted into the Reynolds equation. The Reynolds equation is then integrated to find the pressure distribution, load capacity, friction force and oil flow. The solution to the Reynolds equation becomes more complicated if other effects such as heating, locally varying viscosity, elastic deformation, cavitation, etc., are introduced to the analysis. The basic method of analysis, however, remains unchanged. It is always necessary to start with a definition of the bearing geometry and to perform the integration procedure, taking into account extra terms and equations describing the additional effects that we wish to consider. This approach is very useful as a method of rapid estimation and is widely applied in engineering analysis. The solution of the 2-D Reynolds equation requires the application of numerical method.

4. Computational Procedure

4.1 Introduction

The differential equations which arose from the theories of Reynolds and exceeded the capacity of analytical solution. Before numerical methods were developed, analogue methods, were experimented with as a means of determining hydrodynamic pressure fields. These methods became largely obsolete with the advancement of numerical methods to solve differential equations. This change radically affected the general understanding and approach to hydrodynamic lubrication. Numerical solutions obtained to hydrodynamic lubrication problems are now accepted as per the most engineering requirements for predicting the bearing features. A popular numerical technique, the 'finite difference method is introduced and its application to the analysis of hydrodynamic lubrication is demonstrated. The steps involved to obtain solutions includes:

- Solving Reynolds Equation for SFD
- Finite difference equivalent of the Reynolds equation.
- Definition of solution domain and boundary conditions.
- Coding of MATLAB program with required data.
- Calculation of pressure field.

4.2. Non-Dimensionalization of the Reynolds Equation

Non-dimensionalization refers to the substitution of all real variables in an equation. This process extends the generality of a numerical solution. Analytical expressions are not limited to any specific values and are suited for providing data for general use. A computer program would have to provide a comprehensive coverage of all the controlling parameters.

The major advantage of non-dimensionalization is that it results in a decrease in the number of controlling parameters and an arrow data set provides the vital information.
The Reynolds equation is expressed in terms of film thickness *'h'*, pressure *'p'*, entraining velocity *'U'*, and dynamic viscosity *'q'*. Non-dimensional forms of the equations variables are following:

$$h^* = \frac{h}{c}, x^* = \frac{x}{R}, \; y^* = \frac{y}{L}, p^* = \frac{pc^2}{6U\mu R}$$

where:
h is the hydrodynamic film thickness (m);
c is the bearing radial clearance (m);
R is the bearing radius (m);
L is the bearing axial length (m);
p is the pressure (Pal);
U is the bearing entraining velocity (m/s);
μ is the dynamic viscosity of the bearing (Pas);
x, y are hydrodynamic film co-ordinates (m).

The Reynolds equation in its non-dimensional form is:

$$\frac{\partial}{\partial x^*}\left(h^{*3}\cdot\frac{\partial p^*}{\partial x^*}\right)+\left(\frac{R}{L}\right)^\wedge 2\cdot\frac{\partial}{\partial y^*}\cdot\left(h^{*3}\cdot\frac{\partial p^*}{\partial y^*}\right)=\frac{\partial h^*}{\partial x^*}$$

All terms in equation are non-dimensional apart from ***'R'*** and ***'L'*** which are only present as a non-dimensional ratio.

4.3. The Vogelpohl Parameter

In order to improve the accuracy of the solution obtained for the Reynolds Equation, the Vogelpohl parameter was introduced. The Vogelpohl parameter ***'M'*** is defined as follows:

$$M_v = p^* h^{*1.5}$$

Substitution into the non-dimensional form of Reynolds equation yields the 'Vogelpohl equation:

$$\frac{\partial^2 M_v}{\partial x^{*2}}+\left(\frac{R}{L}\right)^2\cdot\frac{\partial^2 M_v}{\partial y^{*2}}=FM_v+G$$

where parameters 'F' and 'G' for journal bearings are as follows:

$$F=\frac{0.75\left[\left(\frac{\partial h^*}{\partial x^*}\right)^\wedge 2+\left(\frac{R}{L}\right)^2\cdot\left(\frac{\partial h^*}{\partial y^*}\right)^\wedge 2\right]}{h^{*2}}+1.5\frac{\left[\frac{\partial^2 h^*}{\partial x^{*2}}+\left(\frac{R}{L}\right)^2\cdot\frac{\partial^2 h^*}{\partial y^{*2}}\right]}{h^*}$$

$$G=\frac{\left(\frac{\partial h^*}{\partial x^*}\right)}{h_*^{1.5}}$$

The Vogelpohl parameter enables computation by the simplification of the differential operators in the equation. Moreover, it is to be noted that the higher derivatives are not included in the final solution. It can be seen that the introduction of the Vogelpohl parameter does not complicate the boundary conditions in the Reynolds equation. Numerical solutions of the Reynolds equation are obtained in terms of M_v and values of **p*** found from the definition $\frac{M_v}{h^{*1.5}} = p^*$.

4.4. Finite difference equivalent of the Reynolds equation

The finite difference method is based on approximating a differential quantity by the difference between function values at two or more adjacent nodes.

The finite difference equivalent of equation is found by considering the nodal variation of 'M_v' in two axes, i.e., the 'x' and 'y' axes. A second nodal position variable is introduced along the 'y' axis, the 'j' parameter. The expressions for 'F' and 'G' can be added along with the finite difference operator to obtain a comprehensive equivalent of the Reynolds equation. The equation can then be rearranged to provide an expression for 'Mv, I, j' and the following expression

$$M_{v,i,j} = \frac{C_1\left(M_{v,i+1,j} + M_{v,i-1,j}\right) + \left(\frac{R}{L}\right)^2 C_2\left(M_{v,i,j+1} + M_{v,i,j-1}\right) - G_{i,j}}{2C_1 + 2C_2 + F_{i,j}}$$

$$C_1 = \frac{1}{\delta x^{*2}}$$

$$C_2 = \frac{1}{\delta y^{*2}}$$

Where,
This above expression is the finite difference equivalent of the Reynolds equation. Its solution gives the required nodal values of M_v. The finite difference operator is convenient for computation and does not create any difficulties with boundary conditions.

Definition of Solution Domain and Boundary Conditions
After establishing the controlling equation, the next step in numerical analysis is to define the boundary conditions and range of values to be computed. For the journal or pad bearing, the boundary conditions require that 'p^*' or 'M_v' is zero at the edges of the bearing and also that cavitation can occur to prevent negative pressures occurring within the bearing.

The range of 'x*' lies between 0- 2π (360 degrees angle), while considering a complete bearing. Whereas the'y*' ranges from -0.5 to +0.5, such that the bearing mid-line is chosen as the reference datum. Zero value is assumed at the nodes on the edges of the bearing, whereas the solution has to be found using finite difference method at the other nodes.

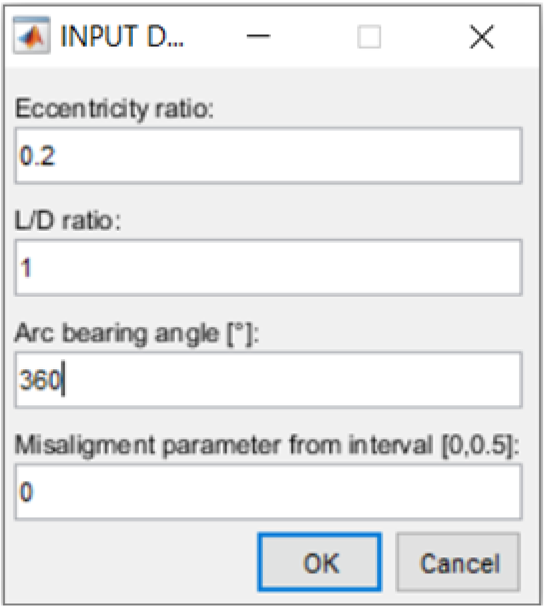
Numerical Solution Technique for Vogelpohl Equation
The nodal values of M_v are conveniently arranged in a matrix with 'i' and 'j' as the column and row ordinates. It is therefore possible to solve equation by matrix inversion but this requires elaborate computation. Programming is greatly simplified when iterative solution methods are applied. The Gauss-Seidel iterative method is used here. All node values are assigned an initial zero value and the finite difference equation is repeatedly applied until convergence is obtained.

A numerical solution to the Reynolds equation for the full operating system of Squeeze Film Damper is necessary for the calculation of pressure distribution, load capacity, lubricant flow rate and friction coefficient. This condition is valid for bearings with L/D ratio in the range l/3 < L/D < 3, where 'L' is the bearing length and 'D' is the bearing diameter.

Results and Discussion

5.1. Input Data

- Eccentricity ratio = 0.2
- L/D ratio = 1
- Bearing arc angle = 360°
- Misalignment parameter = 0



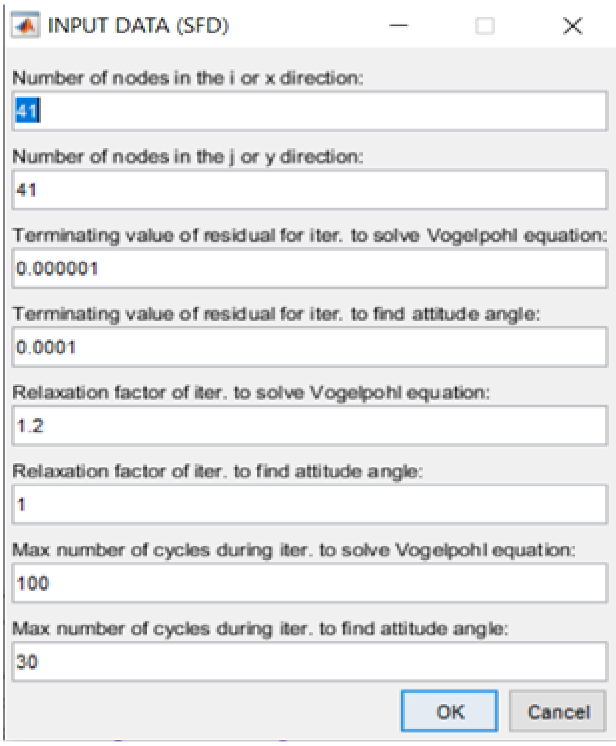


Figure: Input dialog box in MATLAB

5.2. Output Data

- Dimensionless load = 0.085026
- Attitude angle = 77.542°
- Dimensionless friction coefficient = 84.285
- Maximum dimensionless pressure = 0.080201

5.3. Results

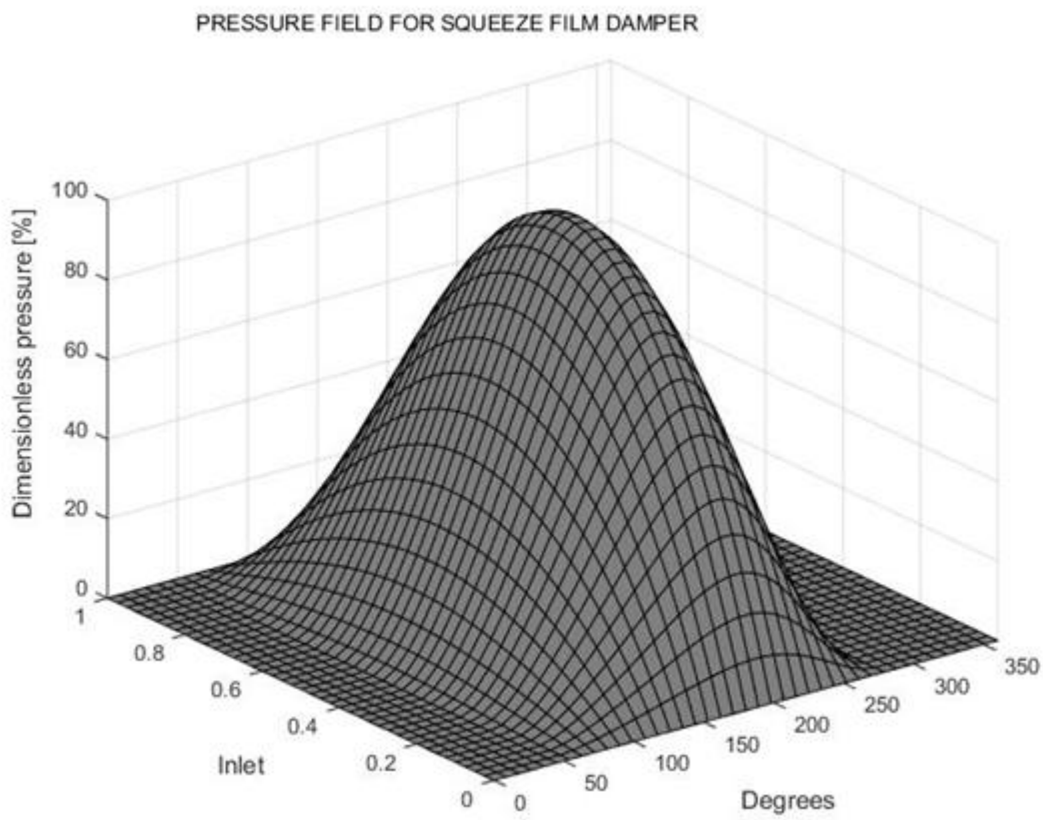


Figure: Final Output of Pressure Field of SFD

6. Conclusions

This report consolidates analytical findings that evaluate the dynamic forced performance of a simply configured closed-ends Squeeze Film Damper. In particular, this report presents identified force coefficients and recorded fluid film dynamic pressures for the SFD undergoing circular whirl orbits ranging with small static eccentricity. Additionally, the analysis in this report states the maximum pressure profile acquired form the whirling of squeeze film damper, it is also graphically represented in the output.

7. Major Findings

The fluid film pressure appears more sensitive to increases in orbit amplitude than to increases in static eccentricity.

However, at a small static eccentricity, the dynamic pressure at the mid plane i.e., location of minimum film thickness shows a sudden increase in the graph. At initial stage the pressure increases, until the cavitation region is reached. There will be gradual increase in the pressure and according to the static eccentricity it reaches a maximum value. Once it reaches the cavitation region there is a sudden drop in the pressure and it gradually decreases to the minimum value as shown in the graph.

8. Future Work

As mentioned, this report states the computational analysis of squeeze film damper. It is suggested that the analyses work of SFD can be carried out using Ansys software. The usage of SFD is widely used in many Mechanical and Aeronautical sectors and therefore the model structure of SFD can be designed and stimulated using Ansys. Thus, it will help in finding out the different specifications, without building test products or conducting crash tests.

9. Closure

This report thoroughly compiles the computational results obtained from squeeze film dampers undergoing circular whirl orbits. The work addresses to industry needs for simpler SFD configurations by means of computational analysis of the dynamic force performance of a very simply configured SFD with short-length (L/D=0.2) and closed-ends. This work also brings to light certain analysis characteristics in SFDs. Computational analysis illustrates the effects with a very low static eccentricity of SFD dynamic force performance.

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