A method of varying bulk modulus in journal bearings to allow for highly cavitied regions to be solved using realistic bulk modulus values

Benjamin Rothwell[1)*], Seamus Garvey[1)] and John Webster[1)]

[1)] Department of Mechanical Materials and Manufacturing Engineering, University of Nottingham, Nottingham, NG7 2RD, UK
[*]Corresponding author: benjamin.rothwell@nottingham.ac.uk

Presented at World Tribology conference 2017 in Beijing, China.

1. Introduction

Journal bearings are used to support a rotating shaft. Hydrodynamic journal bearings operate by using a self-generating hydrodynamic oil film pressure to support the shaft, preventing contact between the shaft and bearing surface (also known as a bush).

The clearance space between the shaft and bush is generally assumed to be full of fluid and the shaft and bush assumed to be cylindrical with their axes parallel. As the shaft is rotated and a load applied, the shaft will move closer to one side of the bush, decreasing the gap and forming a wedge between the shaft and bush. As more fluid is forced into the wedged area, a pressure force is generated, which in turn lifts and supports the shaft off the bush's surface[1]. This concept is shown in Figure 1.

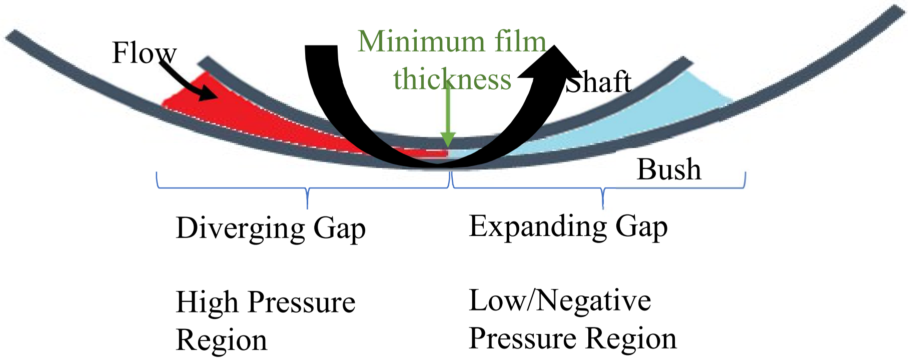

Figure 1 Area of interest around minimum film thickness

Hydrodynamic journal bearings have been understood and operated for a long time, and the advantages and disadvantages of the bearings compared traditional ball bearing/ anti-friction bearing are listed below [2];

Advantages-

- Infinite operating life span as no contact or wear should ever occur
- Low noise compared to anti-friction bearing
- Low transmitted vibration
- Low heat generation

Disadvantages -

- High power loss due to fluid friction
- Requires a constant lubricating fluid with a constant supply
- Cannot operate if fluid is not present
- Radial position of shaft can vary

Current computational models of journal bearings solve for the pressure force generated by the fluid film being force through a wedge by solving the Reynolds equations [3], shown in equation 1.

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right)=6\mu U\frac{\partial h}{\partial x} \qquad \textbf{(1)}$$

In equation 1 p is pressure, h is fluid film thickness, μ is viscosity and U is velocity of the fluid. Z is in axial direction and X is in the circumferential direction.

These models are used to aid the design of bearings allowing approximate geometry and operating conditions to be found. This provides a significant reduction in development effort and time to achieve a practical bearing. Several assumptions are made to allow for these models to be solved;

- There is no pressure change through the radial thickness of the fluid film

- Fluid is supplied into the bearing at the same rate it leaves the side of the bearing
- The fluid is Newtonian
- The fluid film thickness is of several orders of magnitude lower than the width and length of the film, so curvature effects can be ignored
- Fluid body forces are negligible
- Fluid viscous forces dominate the fluid inertia force

At high loads, the pressure force produced by the wedge can be large enough to cause elastic deformations on the shaft and bush surface, increasing the fluid film thickness. Modeling these elastic effects becomes difficult at high loads as the magnitude of the deformation can be in the same order of magnitude as the fluid film thickness (below 10 μm) [3], and a minor change in the fluid film thickness can have a dramatic effect on the pressure profile. This coupling of the pressure and stress-strain relationships is called elastic-hydrodynamic lubrication theory (EHL) and was put forward originally by Dowson & Higginson [1].

Due to the fluid being forced through a wedge shape via the pressure and rotational force, a temperature rise will occur within the fluid and shaft and bush surface. This temperature rise is a function of pressure and can be accounted for via the energy equation (shown in equation 2) [1]. This temperature rise can also cause a change in material properties, affecting the pressure and deformation solutions, as well as changing the fluid film thickness via thermal expansion [4].

$$\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+\mu\left(\left(\frac{\partial U}{\partial y}\right)^2+\left(\frac{\partial W}{\partial y}\right)^2\right)=0 \qquad \textbf{(2)}$$

In equation 2, k equals the thermal conductivity, U and W are the X and Z velocity components and y is in the fluid film height direction.

The point at which the shaft is closest to the bush, is called the minimum film thickness. If both materials are very stiff and the axes constrained to be parallel, this minimum film thickness will run along the entire length of the bearing, with the peak pressure being just upstream of the minimum film thickness. After the minimum film thickness, the fluid pressure will drop dramatically and due to this sharp change in pressure, if the Reynolds equation are only considered a region of negative pressure region would be predicted.

1.1. Solving Cavitation

In a real-world journal bearing, the sudden change in pressure across the minimum film thickness would result in cavitation of the fluid, and the fluid pressure would tend to zero, not negative. Elrod [5] put forward a cavitation boundary condition in 1974 which allowed computational models to account for cavitation by introducing a parameter ϕ (shown in equation 3) and a switch function g [6, 7].

In equation 3, ρ is the density of the fluid and ρ_c is the density of the lubricant at cavitation pressure:

$$\phi=\frac{\rho}{\rho_c} \qquad \textbf{(3)}$$

The pressure within the cavitated film can be solved by considering equation 4, where β is the bulk modulus of the fluid.

$$p=p_c+\beta\ln(\phi) \qquad \textbf{(4)}$$

The Elrod switch function, (g), allows the separation of cavitated and non-cavitated regions by applying the condition presented in equation 5. A modified Reynolds equation was put forward, to account for the cavitated fluid as shown in equation 6.

$$g=\begin{cases}1, for\phi\geq 1\\ 0, for\phi<1\end{cases} \qquad \textbf{(5)}$$

$$\frac{\partial}{\partial x}\left(h^3 g\beta\frac{\partial\phi}{\partial x}\right)+\frac{\partial}{\partial z}\Bigg| \quad \textbf{(6)}$$

This modification to the Reynolds equation meant that when cavitation is present within the fluid, the pressure is set to 0, due to g=0 and when there is no cavitation, g=1 and pressure is solved for. This change is shown in Figure 2.

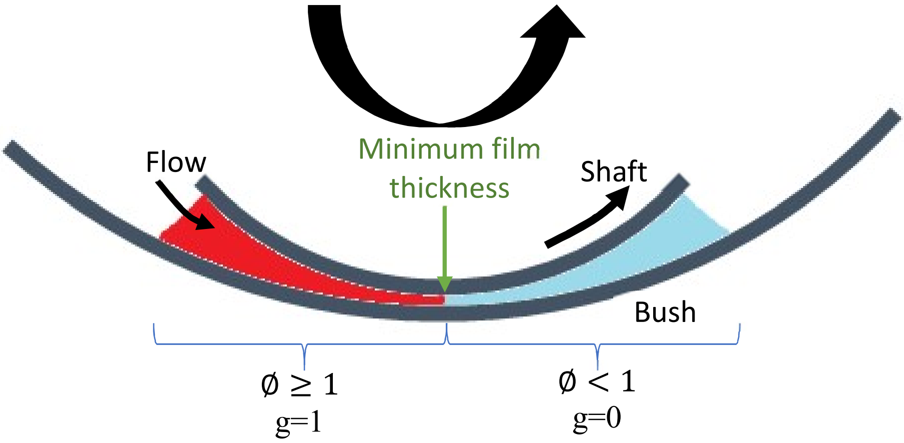

Figure 2 Diagram showing Elrod cavitation condition

The Elrod cavitation condition allows for cavitated solutions to be solved for, but uses a constant bulk modulus throughout the whole fluid domain, meaning the cavitated and un- cavitated regions have the same bulk modulus. This is unrealistic for a realistic fluid, as the cavitated region will contain air bubbles and there the bulk modulus will be reduced. Using a realistic bulk modulus for the fluid throughout the whole domain, will mean the fluid becomes too stiff to solve for in the cavitated region, so Elrod proposed a solution where a lower then realistic bulk modulus was used. Typically computational models use a bulk modulus of around $1\text{x}10^7 \text{N/m}^2$ [9] whilst a typical engine oil, at room temperature, has a bulk modulus of $1.996\text{x}10^9 \text{ N/m}^2$[8]. The draw back to this approach is that using lower bulk modulus reduces the pressure profile seen by bush and shaft. This effect is shown in Figure 3.

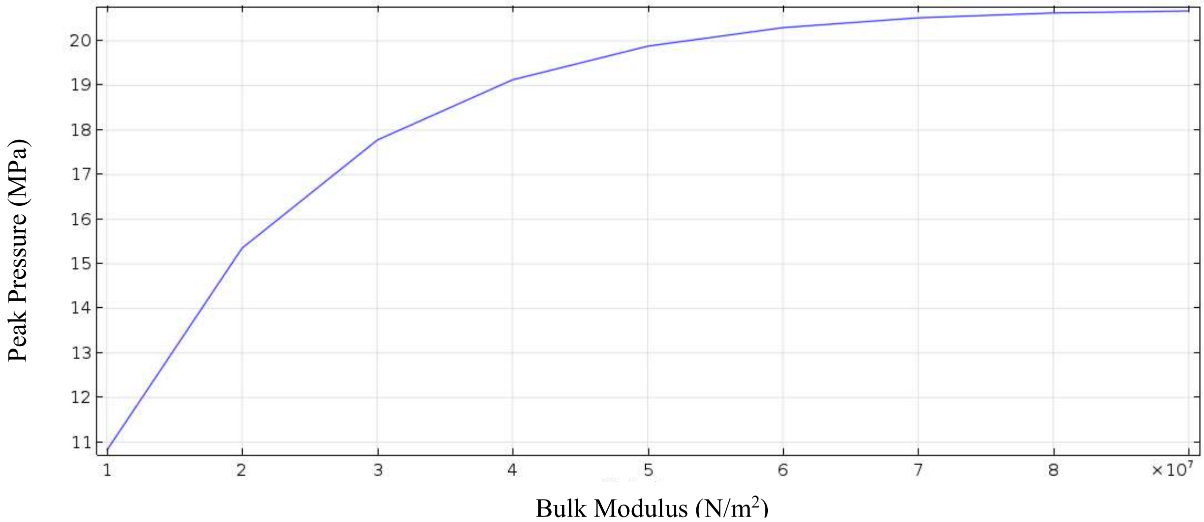

Figure 3 The effect of changing bulk modulus on Peak Pressure

The dimensions of the bearing used in Figure 3 were as followed; Bearing radius= 198.5mm, length= 200mm, radial clearance= 237µm, rotational speed=1000RPM and eccentricity ratio= 0.75, and the data was obtained by using the model described later in this paper.

2. Aims of Study

The aim of this investigation was to find a method to increase the value used for bulk modulus and achieve realistic values, when solving highly loaded cases (eccentricities of 0.95 and above). The model uses Comsol Multi-physics software.

The initial idea put forward was to incrementally increase the value for bulk modulus, using the previous solution as initial conditions for the current step (called the continuous solving method). Using this method, it was hoped that high load cases could be solved, but would increase solving time dramatically for any given load case. This method was later discounted as the model would start to produce unrealistic results, as shown in Figure 4, where zero-pressure regions (cavitation regions) would developed upstream of the minimum film thickness, as well as high pressure regions downstream of the minimum film thickness. From literature [5] it can be seen that this is an unrealistic result, as downstream of the minimum film thickness the pressure tends to zero.

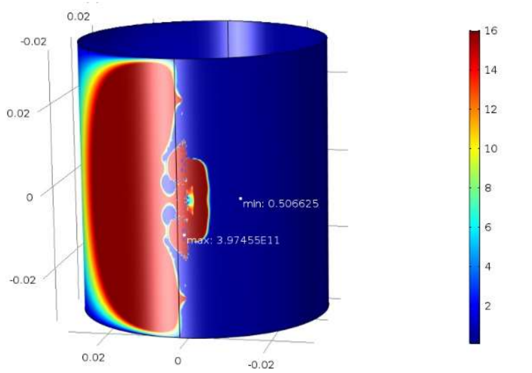

Figure 4 Pressure profile of bush at high bulk modulus using the continuous solving method

The next approach was to use the Elrod switch function, (*g*), as a method of setting a low value for bulk modulus in the cavitated area whilst keeping a high value of bulk modulus in the non-cavitated area. This was believe to be a more realistic condition when compared to a working journal bearing [5].

Two models are explored in this study, called the slope function and the step function. Both allow a higher bulk modulus to be use in the non-cavitated are. Once the Elrod switch function drops below 1, a lower bulk modulus is used. Figure 5 gives a generalized view of the shape of the functions used in the study, along with the currently accepted low value of bulk modulus used in most computational models (called 'constant bulk modulus').

The Step function works by setting the bulk modulus at a realistic high value throughout the whole fluid film, but whenever the value of *g* drops below 1, the bulk modulus is forced to the predetermined lower value.

The Slope function works in a similar manner to the step function, but once the value of *g* drops below 1, the bulk modulus slowly decrease in a linear manner until the lower limit of bulk modulus is reached.

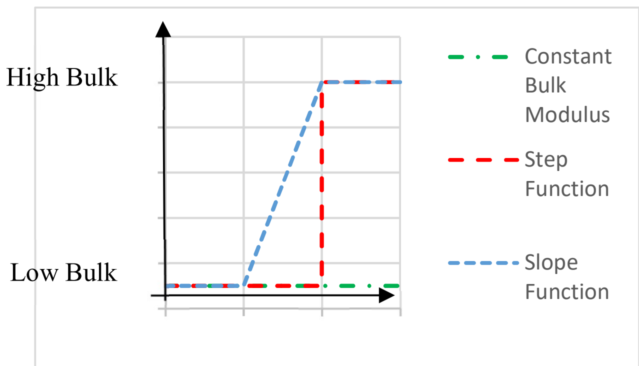

Figure 5 Showing the different bulk modulus models

In the model presented in this body of work is a ridged isothermal model, so does not include elastic or thermal effects. These effects have bene disabled within this model so the effect of varying bulk modulus on the pressure profile can be observed without the film thickness changing (which would affect the load seen by the bearing).

2.1. Objects of study

The proposed functions implemented in the hydrodynamic lubrication model (HL) are expected to reach higher loads and rotational speeds, whilst still producing more realistic pressure profiles.

It is believed that these functions will provide a more realistic pressure solution the traditional methods. An accurate bulk modulus is being used within the non-cavitated area of high pressure, whilst a lower value is used in the cavitated areas, which better matches the expected lower value [5]. It is also believed that the slope function will be able to reach higher loads then the stepped solution, as it provides a gradual transition between the cavitated and non-cavitated domains. This is mathematically more robust and is believed to more closely represent the physical effects present in an experimental rig.

3. References

1. Dowson, D. and G.R. Higginson, *Elasto-Hydrodynamic Lubrication*. Vol. 23. 1977, UK: Pergamon Press.

2. Kelm, R.D., *Journal Bearing Analysis*. 1999, Kelm Engineering, LLC: http://www.kelmengineering.com/.

3. Kuznetsov, E., S. Glavatskih, and M. Fillon, *THD analysis of compliant journal bearings considering liner deformation.* Tribology International, 2011. **44**: p. 1629-1641.

4. Kuznetsov, E. and S. Glavatskih, *Dynamic characteristics of compliant journal bearings considering thermal effects.* Tribology International, 2016. **94**: p. 2880305.

5. Dowson, D. and C.M. Taylor, *Cavitation in Bearings.* Avn Rev. Fluid MEchanics, 1979. **11**: p. 35-66.

6. Elrod, H.G., *A Cavitation Algorithm.* Journal of Lubrication Technology, 1981. **103**(3): p. 350-354.

7. Elrod, H.G. and M.L. Adams, *A computer program for cavitation and starvation problems.* Proceedings of the 1st Leeds-Lyon Symposium on Tribology, 1974: p. 37-41.

8. Boehman, A.L., D. Morris, and J. Szybist, *The impact of the bulk modulus of diesel fuels on fuel injection timing.* Energy & Fuels, 2004. **18**: p. 1877-1882.

9. Notes 6. Liquid cavitation in fluid film bearings. 2016 [cited 2017.

4. Acknowledgements

The financial support of Rolls-Royce Control and Data Services, via the Innovate UK project is gratefully acknowledged. Additionally, we would like to acknowledge AOrbit, which has received funding from the Clean Sky 2 Joint Undertaking under the European Union’s Horizon 2020 research and innovation program under grant agreement No 738176.