DEFLECTION AND PRESSURE ANALYSIS OF OIL FILM BEARINGS[1]

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Abstract

Recent tests on rolling mill oil film bearings have indicated that the oil film thickness is three to five times greater than predicted by computer models. It has been implied that the increase in oil film thickness is due to the deflection of the sleeve and bushing, which would spread out the pressure field and increases the oil film thickness. If the oil film thickness is three to five times greater than expected, the maximum operating load can be increased, taking advantage of the inherent safety factor in the bearing. To confirm the test results, DanOil engineers modeled the sleeve deflection produced by the hydrodynamic pressure field and then used this deflection in a sophisticated bearing computer program to calculate the new pressure field. The iteration of the pressure field and deflection was continued until the model converged. The paper presents the method of analysis and the results.

Key words: Bearing; Film thickness; Methodology.

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1 INTRODUCTION

The oil film bearing is used on hundreds of flat rolling mills throughout the world. The bearings are operating on plate mills, hot mills, cold mills, temper mills, etc., with long life and trouble-free operation. The bearings run on a thin film of oil with very low friction, which results in long bearing life. There is no metal-to-metal contact so there is no wear.
The oil film bearing for rolling mills consists of a sleeve (the journal) and a bushing (the bearing), as shown in Figure 1.

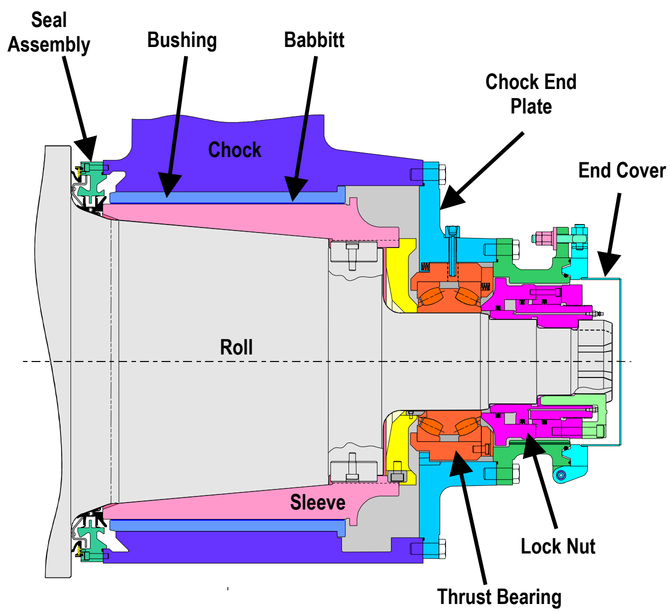

Figure 1. Backup roll bearing.

The journal and bearing surfaces are separated by an oil film which is supplied to the clearance space between the surfaces. The centers of the journal and the bearing are not coincident but are separated by a distance defined as the eccentricity *e*. This eccentricity and the relative motion of the sliding surfaces establish a converging-wedge, which permits a pressure field to develop by viscous effect within the oil film.
It is this pressure field that supports the bearing load, as shown in Figure 2. The Figure refers to perfect cylindrical sliding surfaces.

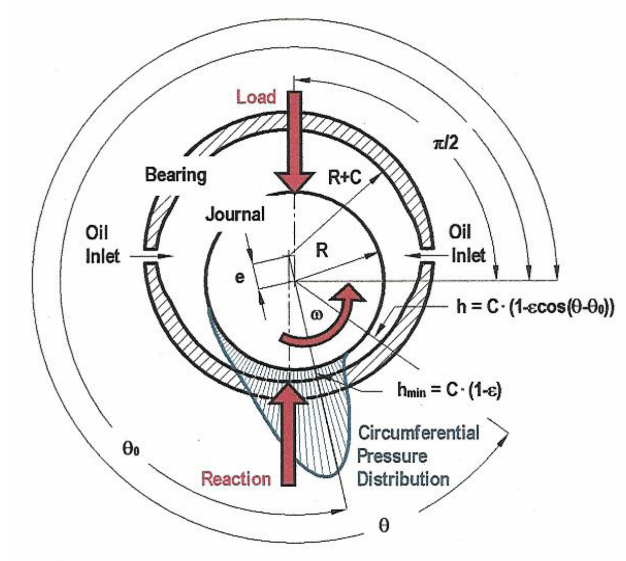

Figure 2. Journal bearing schematic and notation

In the schematic $x = R\theta$ and $U = R\omega$, where R is the journal radius, C is the radial clearance, $\varepsilon = e/C$ is the eccentricity ratio, θ is the angular co-ordinate of the oil film thickness, θ_0 is the angular co-ordinate of the minimum oil film thickness and ω is the angular speed of the journal.

2 THEORETICAL BACKGROUND

The governing equation for the pressure field is the Reynolds equation, which is a second order linear partial differential equation. In steady-state conditions one form of the Reynolds equation in three dimensions is:

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(h^3 \frac{\partial p}{\partial z}\right) - 6\mu U \frac{\partial h}{\partial x} = 0$$

where p is the pressure, h is the oil film thickness and μ is the absolute constant viscosity of the oil. The spatial co-ordinates are denoted by x and z. The bearing is fixed, and the journal is moving in the circumferential x-direction with constant peripheral speed U. The journal and the bearing are of finite width in the z-direction perpendicular to the x-axis. Both the effects of flow in x-direction and z-direction are included.

The solution of the Reynolds equation in three dimensions is obtained by a finite difference technique. Once the pressure distribution is known the bearing operating characteristics, such as minimum oil film thickness, peak pressure, frictional losses, lubricant flow requirement and temperature rise, can be determined.

Although it is well known from experience that the viscosity varies with both the temperature and the pressure, these effects are not included in the Boyd & Raimondi[1] solution because of the numerical complications involved. The error made by assuming that the viscosity does not change with temperature, may be minimized by using the viscosity at the mean temperature of oil inside the bearing. The bearing deformations due to the pressure field and the variation of viscosity with pressure are assumed to be negligible. Oil film rupture due to negative pressure is considered.

3 CONUNDRUM

At the 41st Rolling Seminar Processes, Rolling & Coated Products, Morgan Construction Co. presented test results[1] that indicated the test oil film thickness, in load direction, was three to five times greater than Reynolds equation solution calculated by Boyd and Raimondi[2] at high loads and low speeds.

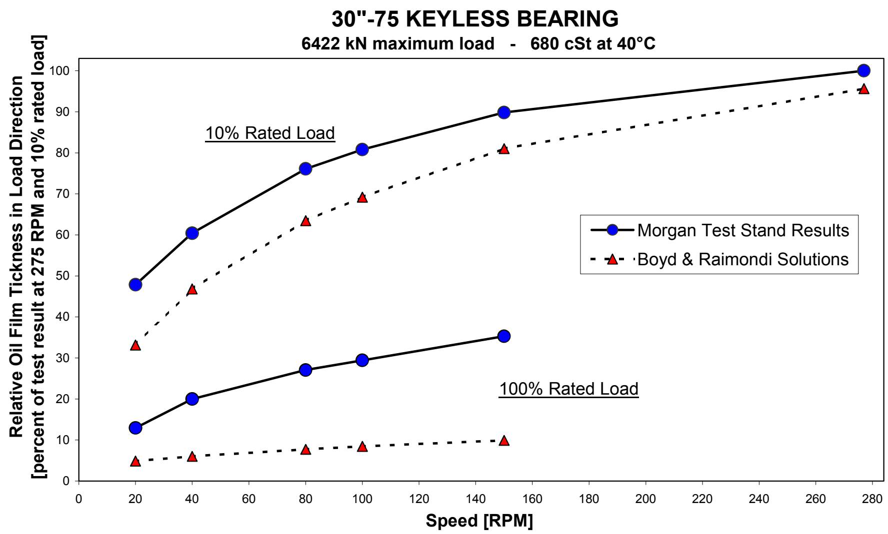

Figure 3. Minimum oil film thickness in load direction.

It has been suggested that the sleeve and bushing deformations could cause the oil film-pressure field to behave in a manner not predicted by the classical theory. According to this thought, at high loads and low speeds, the sleeve deformation flattens the journal surface, which causes a spreading out of the pressure field. This, in turn, would increase the oil film thickness. A thicker oil film would allow the maximum bearing capacity to be increased. This hypothesis intrigued DanOil engineers.
At low loads and high speeds the deformation is less and the oil film thickness is closer to the computer solution (see top curves). This would be logical, because the deformation would be less.

4 NOVEL APPROACH

Until today there has been no computer software available combining pressure and deflection analysis, which simulates the actual oil film bearing operating conditions. The classical methods, like Boyd and Raimondi[2], assumes that there is no deformation or misalignment, that the journal and the bearing are perfect cylindrical surfaces.
In fact, the sleeve deformation and the pressure field are dependent variables. The sleeve deformation is a function of the pressure field and the pressure field is a function of the sleeve deformation.

DanOil engineers have conceived a novel approach. It is a methodology, which couples the solid and fluid fields of the oil film bearing in an elasto-hydrodynamic analysis. A finite difference method (**FDM**) software, which calculates the pressure field together with the distortion of the sliding surfaces, is combined with a finite element method (**FEM**) software, which calculates the deformations of the sleeve and bushing due to the pressure field. The two programs are linked through MATLAB, a high-performance technical computing language.

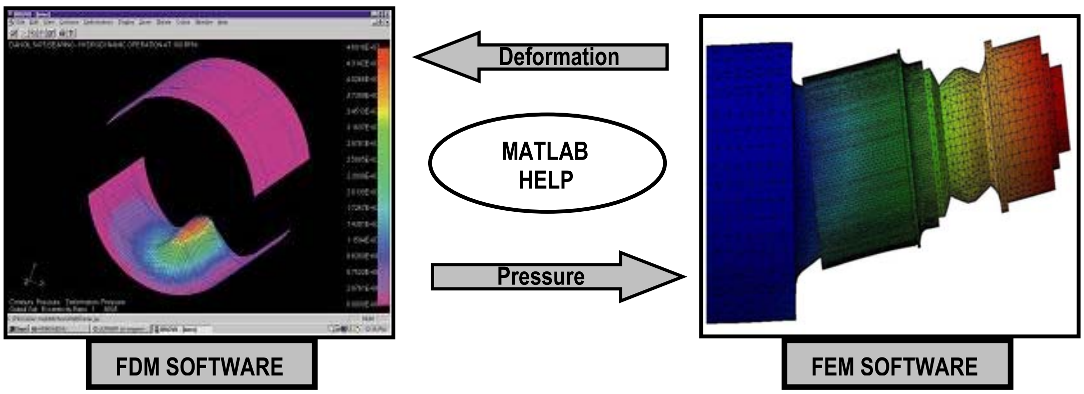

Figure 4. DanOil methodology.

The DanOil methodology is illustrated in Figure 4. It is an iterative algorithm, which consists of the following fundamental steps. First, an initial value for the oil film thickness and pressure distribution is calculated. MATLAB then transfers the pressure distribution as a load to the FEM software. Next, the deformation of the journal and bearing surfaces is calculated and transferred back to the FDM software. The new deformation is used in the FDM calculation to provide an improved oil film thickness and pressure distribution for the next iteration. The algorithm continues to loop through the solid and fluid analysis until the desired convergence is reached.

Initial results

First, the FDM software was run without deformation, producing the same results as those presented by Morgan with the Boyd and Raimondi solution. Then the DanOil methodology, with the sleeve and bushing deformation considered, was rerun. There was, however, only a slight increase in the oil film thickness but nowhere near the 300% to 500% increase indicated by the test results. So, there had to be another factor which increases the oil film thickness.

Viscosity

Lubricants, especially petroleum-base lubricants, undergo considerable increase in viscosity when they are subjected to high pressures. Fuller[3] gives some examples of the viscosity change with pressure and states that "the increase in viscosity due to this pressure will raise the friction losses in the bearing; but, what might be more important**, the increased viscosity due to pressure will increase the load-carrying capacity of the bearing and the oil film thickness over what might be**

expected if the pressure-viscosity effect were ignored." DanOil engineers believe this is what happens in a bearing under operating conditions.

The DanOil methodology was modified to consider the increase in viscosity as a function of the mean temperature as well as the mean pressure. As suggested by Fuller, DanOil engineers decided to estimate the viscosity, introducing an empirical relationship:

$$\mu = \mu_0 e^{(aP+b)}$$

where μ_0 is the absolute viscosity at a reference condition, *P* is the mean pressure and *a* and *b* are empirical test constants that depend on the mean temperature.

The results are shown in Figure 5 and 6.

Figure 5 shows that the prediction of oil film thickness is now very accurate. Within normal operating conditions the deformation of the sliding surfaces is negligible and only a good estimate of the viscosity relationship is really essential.

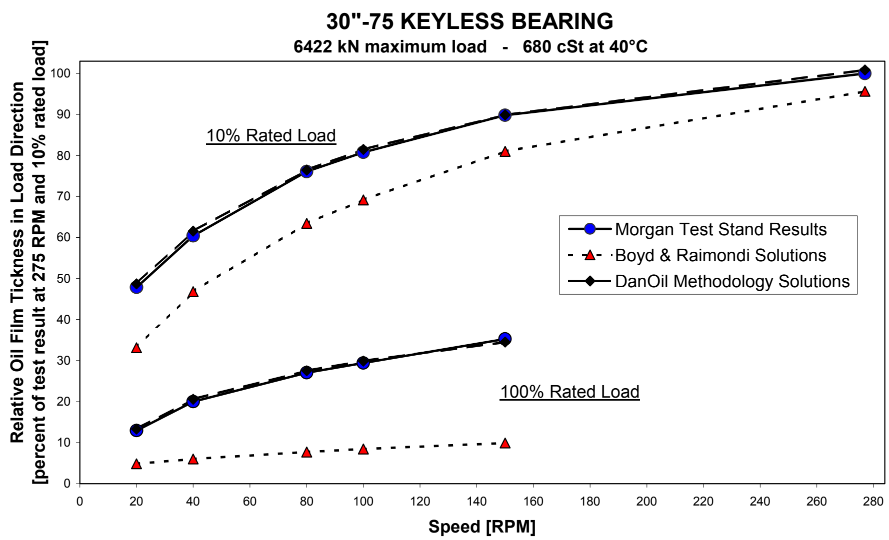

Figure 5. Minimum oil film thickness in load direction.

Figure 6 shows the pressure distribution at the centerline of the bearing under maximum load at 20 rpm. The peak pressure calculated with the DanOil methodology is much lower than given by the classical solution under high loads. When the speed is increased, the two solutions converge. At low loads no significant differences of pressure exist. Because the peak pressure is lower than expected, it is logical that the sleeve deformation is lower than expected.

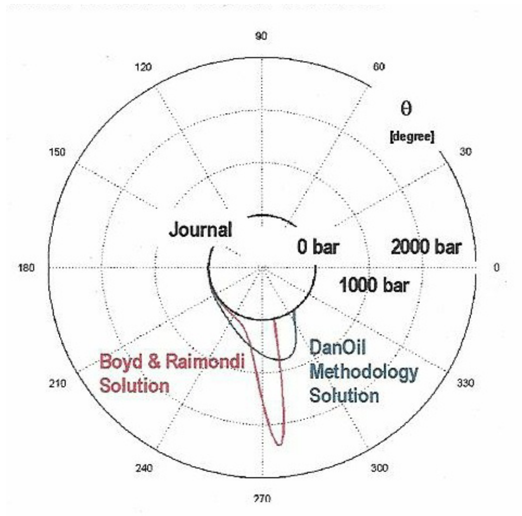

Figure 6. Pressure distribution at the centerline of the bearing (100 % rated load, 20 rpm and 680 cSt at 40°C).

5 COMPARISON OF PERFORMANCES BETWEEN KEYLESS AND THIN-WALL BEARINGS

It has been suggested that the thin-wall sleeve will deflect more than the thick-walled keyless bearing, which will spread out the pressure field and increase the oil film thickness. DanOil engineers ran their model comparing the deformation of the thin-walled sleeve and the thicker-walled keyless sleeve. The result is shown in Figure 7.

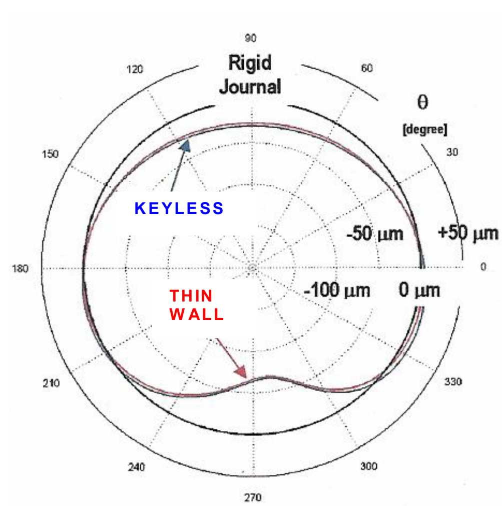

Figure 7. Sleeve deformation at the centerline of the bearing (100 % rated load, 20 rpm and 680 cSt at 40°C).

The change in oil film thickness as a function of speed is shown in Figure 8. The differences are insignificant when comparing the two sleeves.

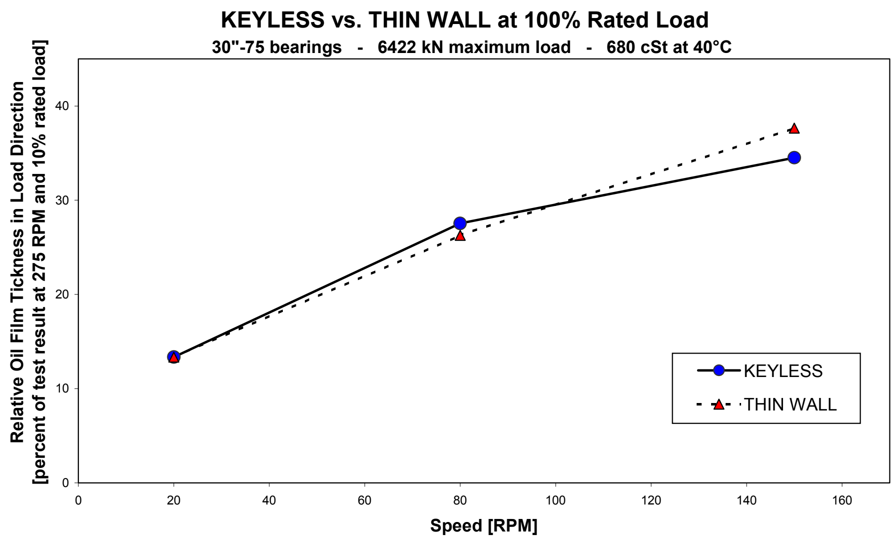

Figure 8. Minimum oil film thickness in load direction.

Figure 9 and 10 show respectively the first and third principal stress at the centreline of the thin wall sleeve and the keyless sleeve. Again, there is an insignificant difference in stress.

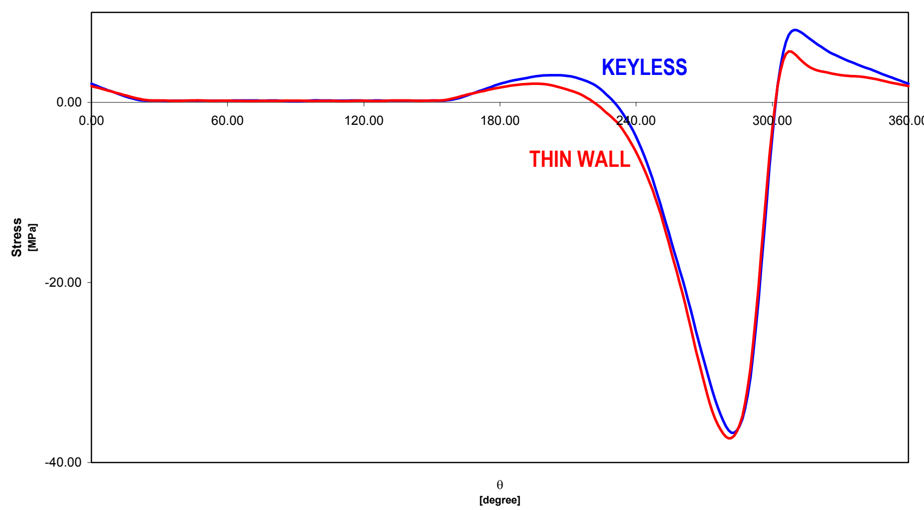

Figure 9. First principal stress at the centerline of the bearing (100 % rated load, 20 rpm and 680 cSt at 40°C).

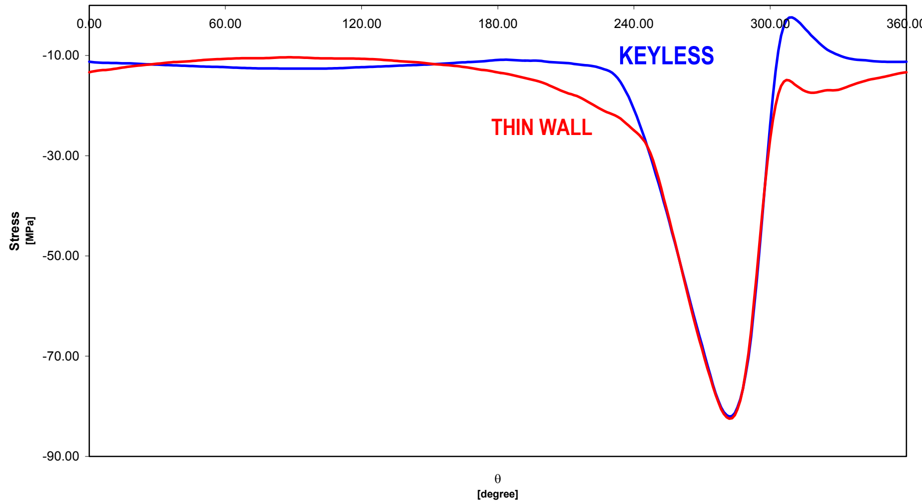

Figure 10. Third principal stress at the centerline of the bearing (100 % rated load, 20 rpm and 680 cSt at 40°C).

6 BEARING MAXIMUM OPERATING LOAD

The bearing maximum operating capacity is based upon the oil film thickness and bearing temperature. It is also based upon the mechanical strength of the various components in the bearing assembly, i.e., sleeve stress, bushing babbitt strength and chock deformation. To consider just one factor, such as sleeve stress, is short sighted. In fact, the sleeve stress is the least important factor because the stress is very low compared to its operating limit. Sleeve failures almost never happen. Usually the bushing babbitt fails, which destroys the sleeve if operation continues. The babbitt failure is due to fatigue or wiping of the babbitt, which is a form of creep. The weakest component in the bearing is the babbitt in the bushing.
DanOil uses a special manufacturing technique, which increases the compressive stress of the babbitt by 54%. If the super strong babbitt is used, the compressive strength can be increased by 102% and the creep reduced by 90%. DanOil engineers have increased the capacity of the keyless oil film bearing by 24%. This can be used to reduce the size of the bearing, chock and mill stand.

7 CONCLUSION

DanOil engineers have combined two numerical programs to simultaneously model the hydrodynamic pressure field and the elastic deformation of the oil film bearing, resulting in the actual representation of the bearing operating characteristics. The model accurately predicts the bearing characteristics in an operating environment when combined with the change in lubricant viscosity with pressure.
With this model, DanOil engineers have shown that the thin-walled sleeve has no advantage over the tried and tested keyless sleeve. The keyless bearing has been used on thousands of mill stands throughout the world with excellent results.
The model shows that the oil film thickness increases at low speed and high load due to the increase in viscosity with pressure. The increase in bearing capacity due to the increase in oil film thickness is possible, but care should be taken regarding the

bushing. The babbitt in the bushing must be strong enough to withstand the 24% increase in operating load. DanOil engineers have shown that the maximum operating load can be increased for the thin wall sleeve or the keyless sleeve, but only with the application of a new, stronger babbitt material. DanOil manufacturing techniques produces a stronger babbitt, which can be used successfully at the higher mill loads.

REFERENCES

1 T. C. Wojtkowski, G.F. Royo, P. N. Osgood, REDUCING MILL CAPITAL COST BY ADOPTING THE MORGOIL KLX BEARING, 41° Seminário de Laminação - Processos e Produtos Laminados e Revestidos.
2 Boyd and Raimondi, A Solution for the Finite Journal Bearing and its Application to Analysis and Design: III. ASLE Transactions, Vol 1, No. 1, April 1958.
3 Dudley D. Fuller, Theory and Practice of Lubrication for Engineers, Second Edition, John Wiley & Sons.