Minimum Design Speed Considerations for Sleevoil Bearings for Industrial Fan Applications

Will Cannon, Application Engineer, Baldor Electric Company

Abstract

Applications that require slow speed operation and use hydrodynamic bearings can successfully operate in reduced film thickness conditions as long as the shaft hardness is adequate and the shaft surface finish is between 16–32 microinches. The reason is because a thinner oil film is sufficient as long as the shaft surface asperities are small and a minimal number of broken metallic fragments from metal to metal contact are allowed to pass through the oil film without causing damage. A small circulating oil unit will further increase the bearing reliability at slow speed operation by flooding the bearing with oil when the oil rings are delivering minimal oil from the slow shaft rotation and by providing continuous particle filtration.

**Disclaimer: *This paper and its contents are strictly concerning hydrodynamic bearings used in industrial fan applications; all suggested values are derived from years of experience.*

Introduction

Slow speed operation of fan systems within the air handling industry is generally performed due to two reasons: a coast down operation is required for hot (induced draft) fans to cool down before shutdown (often by using a turning gear), and operational efficiency improvements can be achieved during non-peak periods by slow speed operation using a VFD. In either case, when these fans are supported by hydrodynamic bearings, it is the oil film thickness developed from the bearing-shaft interaction that limits the minimum speed that can be maintained without causing premature wear and bearing failure. This paper will present a brief overview of lubrication theory and critical design parameters to achieve slow speed operation.

There is an extensive amount of technical literature available to describe lubrication and hydrodynamic theory; this paper is only intended to present a brief overview of public knowledge.

Lubrication Theory

There are four regimes of lubrication, and they are generally defined according to the film thickness parameter (Snyder, Ovaert, and Wedeven, 2014):

$$\lambda = \frac{h_{min}}{\sqrt{R^2_{journal} + R^2_{bearing}}} \tag{1}$$

where

l = film thickness parameter, h_{min} = minimum film thickness, $R_{journal}$ = surface finish of shaft (rms), and $R_{bearing}$ = surface finish of bearing (rms) (Singhal, 2008).

Boundary Lubrication (**l** < 1)

Boundary lubrication implies that the two mating surfaces are not separated by the lubricant and that extensive contact between the surface asperities exists. High coefficients of friction and significant wear from metal to metal contact are notable characteristics within this lubrication regime.

Mixed Lubrication (1 < **l** < 3)

This regime maintains contact between the two surfaces but has a lower coefficient of friction than boundary lubrication. The lubricant provides some amounts of load support, but it is insufficient to separate the surfaces. Significant wear occurs in this regime.

Elastohydrodynamic (EHL) Lubrication (3 < **l** < 10)

In EHL lubrication, the lubricant provides substantial amounts of load support to separate the surfaces. Although the coefficient of friction drops to a minimum, fatigue wear such as pitting can still occur in this regime due to the distribution of stresses which still cause surface deformations (Gohar, 2001).

Hydrodynamic Lubrication (10 < **l** < 100)

In hydrodynamic lubrication, the entire load is supported by the hydrodynamic pressure developed within the lubricant film. The coefficient of friction is slightly higher than the EHL regime, but there is minimal wear that will occur on either surface, indicating infinite life can theoretically be achieved in this regime.

Lubrication Regimes

Another relation that is beneficial in understanding the regimes of lubrication is the Hersey number:

$$H_s = \frac{\eta\omega}{P} \tag{2}$$

where

h is the absolute viscosity, **w** is the rotational speed, and P is the pressure. A larger Hersey number would imply a thicker oil film (larger **l**) whereas a lower oil film thickness would be present at a lower Hersey number (Stribeck, 1902). This dimensionless parameter can be compared to the coefficient of friction as it increases and changes lubrication regimes, as shown in a Stribeck curve (Figure 1).

where

μ = coefficient of friction. The position and shape of the actual Stribeck curve is specific to each application and is dependent upon the composite surface roughness (factor of **l**), as well the contacting materials, lubricant, and other factors (Snyder, Ovaert, and Wedeven, 2014).

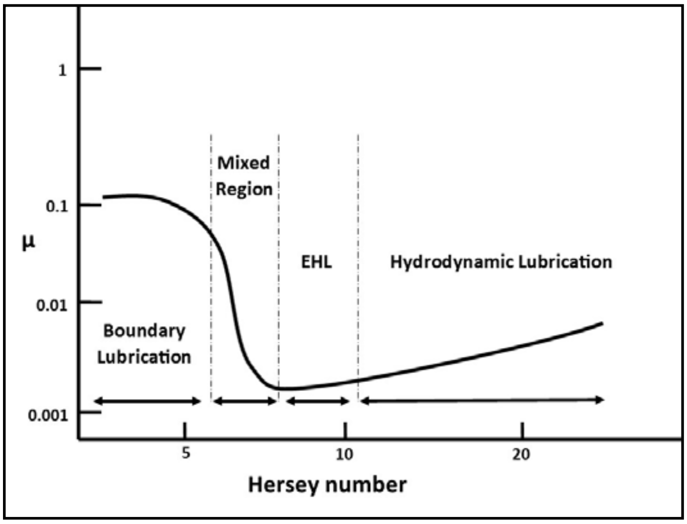

Figure 1 Stribeck curve comparing coefficient of friction with dimensionless Hersey number (Hamrock, Schmid, and Jacobson, 2004).

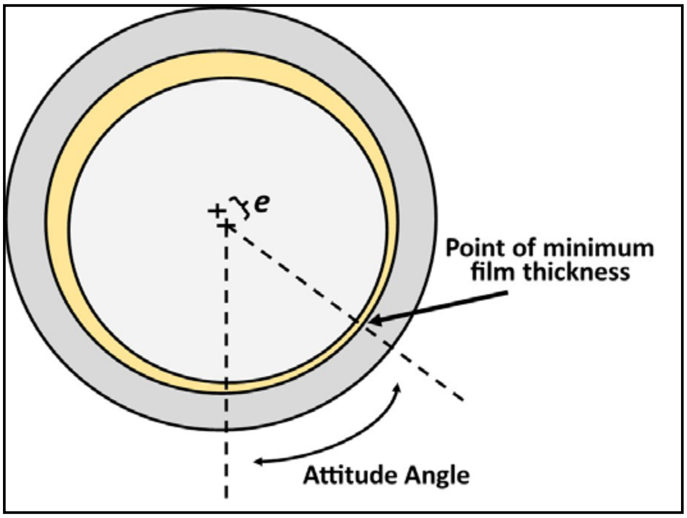

Figure 2 Illustration showing the attitude angle and point of minimum film thickness.

Hydrodynamic Theory

In addition to lubrication regimes, hydrodynamic theory must also consider application details in order to analyze the performance of a fluid film bearing at low speeds. The three requirements for the development of hydrodynamic pressure are: relative surface motion, converging geometry, and a viscous lubricant (He, 2005). One of the main governing parameters of hydrodynamic operation is the Sommerfeld number (S), also known as the bearing characteristic number, which is a dimensionless parameter that provides a relationship for dynamic stiffness and damping coefficients.

$$S = \frac{\upsilon NLD}{W}\left(\frac{R}{c}\right)^2 \quad (3)$$

where
ν = viscosity, N = shaft speed, L = bearing length, D = bearing diameter, W = rotor weight, R = radius of shaft, and c = radial clearance. The Sommerfeld number relates the geometric parameters of the bearing with the application specifics of weight, oil viscosity, and speed to determine the eccentricity ratio (ε):

$$\varepsilon = \frac{e}{c} \quad (4)$$

where
e = shaft eccentricity (distance between shaft and bearing centers). The eccentricity ratio is directly proportional to the attitude angle and both will vary with changes in the Sommerfeld number. The attitude angle will be the location where the shaft will ride on the oil film within the bearing at steady-state operation.

Understanding the fundamentals of hydrodynamic theory is critical to designing and producing a bearing that will continuously operate satisfactorily under various speed and load conditions and with extended bearing life.

Design Parameters

Shaft Surface Finish

The most critical design parameters in determining the minimum allowable speed while maintaining hydrodynamic lubrication are the oil viscosity and load (properties of the Hersey number [Eq. 2]), geometric tolerances of both the bearing and shaft (properties of the Sommerfeld number [Eq. 3]), and the composite surface roughness between the shaft and bearing (property of the film thickness parameter [Eq. 1]). Assuming that the pressure applied to the bearing from the rotor and shaft weight is constant, and also assuming that both the bearing and shaft geometries are adequately designed and manufactured (minimal run out and defects), then there are only two remaining factors that can be manipulated to decrease the minimum allowable speed while maintaining hydrodynamic lubrication: increasing the lubricant viscosity and decreasing the composite surface roughness. Selecting an oil with a higher viscosity grade would improve the minimum allowable operating speed, but at normal operating speeds, the higher grade viscosity would generate much more heat from oil shearing than a lower viscosity grade, as well cause the shaft to ride on the oil film at an attitude angle beyond the stability region which can cause instability and excessive vibration. Therefore, it can reasonably be concluded that the composite surface roughness from the bearing and shaft is the only variable that can be manipulated to improve the minimum allowable design speed. Furthermore, since the surface finish of the bearing load zone as provided by the manufacturer is constant, it is the shaft surface finish which can be improved that will lower the minimum allowable operating speed. However, experience has shown that there exists an optimal shaft surface finish range from 16–32 micro-inches; any less than this range has produced diminishing results in developing a hydrodynamic film.

Minimum Speed

Under normal operating conditions, the minimum recommended oil film thickness is 1 mil (thousandths of an inch), based on a shaft surface finish of 32 microinches. However, if a shaft is within this optimal surface finish range of 16-32 microinches, then the minimum allowable film thickness can be reduced to accommodate slow speed operating conditions (Table 1).

For example: if the minimum operating speed of an application was 100 rpm for a shaft with a 32 μ-in surface, then with a 16 μ-in surface, the minimum operating speed could

Table 1 Reduced acceptable film thickness values based on optimal shaft surface finish ranges. **These values are based on experience.		
Shaft Surface Finish [µ in]	**Acceptable Minimum Film Thickness [mils]**	**Approximate Percentage of Minimum Operating Speed**
16	5/10	60%
24	7/10	80%
32	1	100%

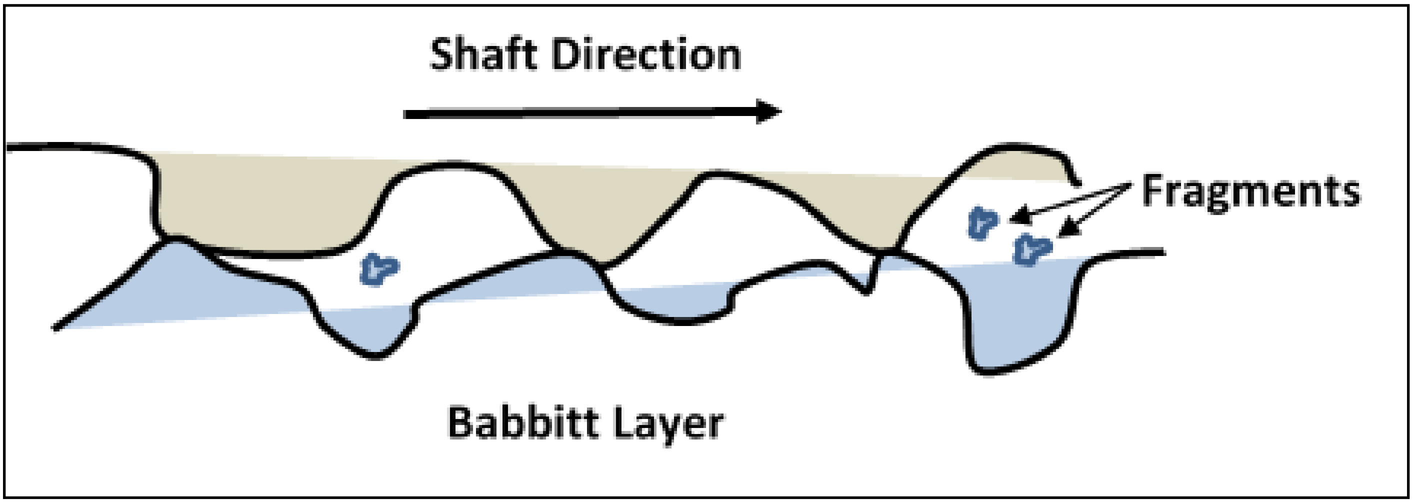

Figure 3 The shaft surface asperities will break the liner surface asperities and leave broken fragments that can advance bearing wear and failure.

be approximately 60%, or 60 rpm. The justification for this is that there will still be a complete separation of surfaces (full hydrodynamic lubrication) since the surface asperities are considerably small. Shaft run-out must be kept to a minimum because any high spots on the shaft will lessen the minimum film thickness and possibly provide an area of contact between the surfaces, inducing friction heat and even causing pre-mature failure.

Additional Performance Improvements

The benefits of having a hard shaft within the recommended surface finish range is the constructive wear-in it that will occur on the liner surface. As the shaft rotates directly on the babbitt layer during start-up and shut-down when hydrodynamic lubrication is not possible and there is metal to metal contact, the contact between the shaft surface asperities will break the surface asperities of the liner, since the shaft material is much harder than the babbitt material (Figure 3).

The broken fragments can accelerate bearing wear and the onset of bearing failure unless they are sufficiently small and minimal in quantity, in which case the shaft will burnish the babbitt layer and improve the liner surface. A sufficiently small fragment (–25% of the size of the film thickness) implies that it is sufficiently smaller than the minimum oil film thickness once full hydrodynamic lubrication develops so that it can easily pass through the film without causing abrasive damage to either surface. The fragment size that will be broken off can be minimized by either decreasing the load or increasing the hardness of the shaft (Smart, 2009). Additionally, the condition of the oil should be continuously monitored to prevent a build-up of metallic fragments in the oil.

Since slow speed applications generally require a minimum oil film thickness less than 1 mil, it is of critical importance to the life of the bearing that the shaft is both hard and within the optimal surface finish range, which ensures the fragments broken off will not be abrasive during hydrodynamic lubrication and a proper oil maintenance schedule is in place.

Additional Reliability at Slow Speeds

However, since shaft surface defects are inevitable, additional reliability can be incorporated into slow speed applications by using a small circulating oil unit. Self-lubricating hydrodynamic bearings are entirely dependent on oil rings to rotate on the shaft to deliver oil to the bearing (Figure 4).

This means that during start-up, shut-down, and slow speed operation, the oil rings are rotating at slower speeds and the quantity of oil being delivered is reduced which can cause the bearing to run in starved conditions. Since the reli-

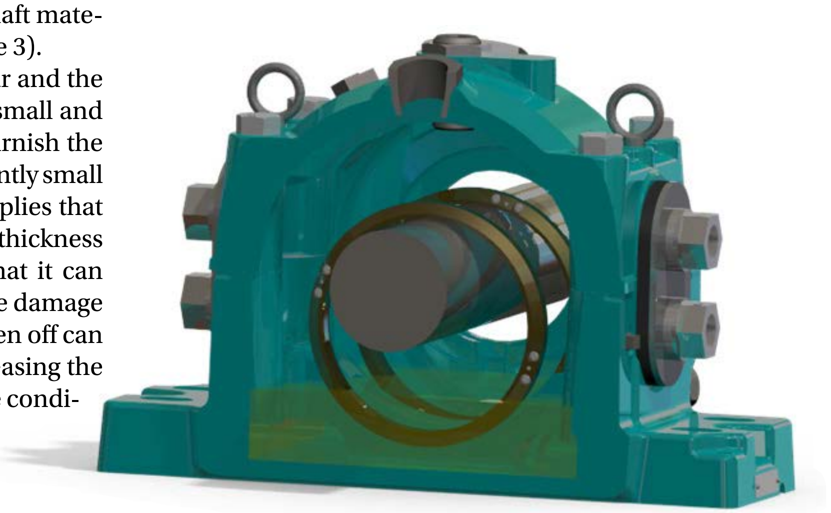

Figure 4 Oil rings rely on shaft rotation to deliver oil to the bearing.

able operation of the bearing is entirely dependent upon the shaft and bearing surfaces being separated by a film of oil, it is best practice to use a small circulating oil unit to ensure adequate oil supply when the oil rings are not rotating sufficiently often.

Conclusion

Hydrodynamic lubrication is dependent upon many factors, and with careful design and manufacturing, hydrodynamic bearings can adequately support applications that require slow speed operation. In order to design an application to continuously operate in a film thickness region below the recommended 1 mil, the shaft surface finish and hardness are critical parameters that determine whether the bearing will survive or fail in these conditions. Circulating oil units enhance the reliability of slow speed operation, as they deliver oil to the bearing and prevent starved conditions when the oil rings are turning at a reduced speed and are limited from providing an adequate oil supply for full hydrodynamic lubrication. **PTE**

William Cannon is an application engineer with Baldor Electric Company, a member of the ABB Group, with responsibility for Baldor·Dodge sleevoil bearings. Since joining the company in 2014, he has been involved in product development projects with hydrodynamic bearings, and he received a company-specific engineering achievement award for the Baldor·Dodge RTL-Spherical Bearing. William is a graduate of Clemson University.

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