

Robust Design of Spiral Groove Journal Bearing

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ABSTRACT

Journal bearings are machine elements designed to produce smooth (low friction) motion between solid surfaces in relative motion and to generate a load support for mechanical components. In a Journal bearing, the entire load is carried by a thin film of fluid present between the rotating and the non-rotating elements. The thickness of the film is very sensitive to ambient temperature, radial clearance and misalignment. Though these parameters are difficult to control, it is important to consider these as noises while designing the Journal bearing. When noises in the design space are very strong, the conventional DOE and RSM methods suppress the effect of control factors and render them irrelevant in the design process. This leads to overdesigning the product, which in turn adds more cost.

The main objective of this work is to design a bearing, which is insensitive to noises at each stage of the product life cycle. Parameter design and tolerance design for journal bearing have been carried out using orthogonal array based Taguchi methodology. Optimal levels of the control factors have been found from the factor effect plots for the S/N ratio. It is observed that the Taguchi design shows a reduction of 68% in the Unit Manufacturing Cost (UMC) and about 6 times increase in the average life of the bearing when compared to the baseline design.

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INTRODUCTION

Bearings are members that allow constrained relative motion between two or more parts and promote free rotation around a fixed axis or free linear movement. Bearings may be classified broadly according to the type of motion, principle of operation, the directions of applied load, etc. Based on the nature of contact, they are classified as Sliding contact type and Rolling contact type bearings [1].

Sliding contact type bearing (Fluid film bearings) are machine elements designed to produce smooth (low friction) motion between solid surfaces in relative motion and to generate a load support for mechanical components. In fluid film bearing, the entire load of the shaft is carried by a thin film of fluid present between the rotating and non-rotating elements. The different types of journal bearings are - Plain journal bearing, Partial arc journal bearing, Axial groove journal bearing, Spiral groove journal bearing, Elliptical journal

bearing and Three or four lobe journal bearing. Based on the application and load carrying capacity required, one can select the appropriate bearing.

In the design of journal bearings, the Sommerfeld number, or the bearing characteristic number, is a dimensionless quantity used extensively in hydrodynamic lubrication analysis. The Sommerfeld number contains major geometrical and operating variables and indicates the operational state of a journal bearing [2,3].

The Sommerfeld number is typically defined by the following equation:

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P'} \quad (1)$$

where,

S is the Sommerfeld number,

r is the radius of the journal,

c is the radial clearance,

μ is the absolute viscosity of the lubricant,

N is the rotational speed of the shaft,

P' is the load per unit of projected bearing area.

In order to differentiate between the lubrication regimes, a non-dimensional quantity called the fluid film parameter (λ) (Equation 2) is used. It is defined as the ratio between the minimum film thickness to the composite surface roughness of the journal and the bearing [2].

$$\lambda = \frac{h_{min}}{\sqrt{\sigma_j^2 + \sigma_b^2}} \quad (2)$$

where,

λ is the fluid film parameter,

h_{min} is the minimum film thickness,

σ_j is the surface roughness of the journal,

σ_b is the surface roughness of the bearing.

The reliability of machine elements depends greatly upon the amount of fluid film present between the mating surfaces. If a proper amount of film thickness is not maintained during machine operation, it can cause changes in the friction coefficient, as well as several other failures like deformation, dynamic instability or failures due to contamination (also known as scuffing). Figure 1 shows summary of different lubrication regimes and possible failures [2, 4 and 5]. The three categories of lubrication regimes are:

1. Boundary lubrication ($\lambda < 1$)
2. Mixed film lubrication ($1 \leq \lambda \leq 5$)
3. Hydrodynamic lubrication ($\lambda > 5$)

As discussed above, for reliable operation of the bearing, it is necessary to maintain the minimum film thickness at the safe limit, shown in Figure 1. Apart from the journal bearing geometrical parameters, the minimum film thickness also depends largely on the type of operating fluids and their properties, variation in clearance and misalignment of journal bearing. Thus, for industrial applications, such as automotive and aerospace, where there is a large variation in the environmental conditions, it becomes a challenge to design the bearing for reliable operation.

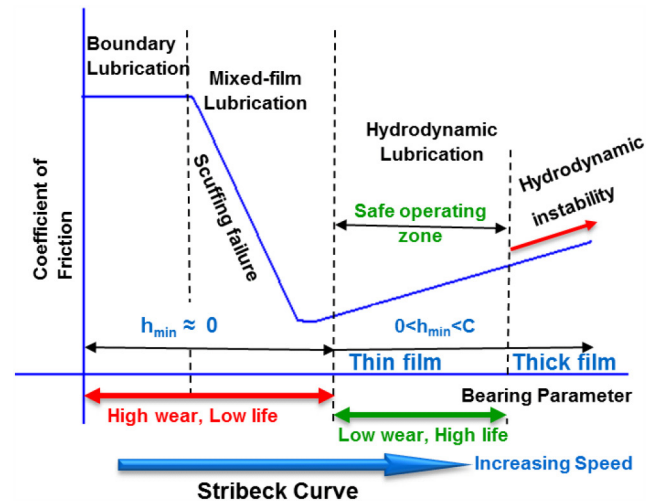


Figure 1. Stribeck curve and bearing safe operating zone with respect to h_{min}

Literature survey by Martin [6] summarizes how the bearing performance is impacted by oil feed features, oil film history, bearing profile, journal inertia, etc. An experimental study by Schneider and Blossfeld [7] shows the effect of journal surface finish on journal bearing load capacity. Radil et al. [8] and Chu et al. [9] have discussed how the variation of radial clearance affects the performance and life of the bearing. He et al. [10] observed that bearing performances are greatly affected by misalignment caused by the asymmetric structure of the journal. Jang and Khonsari [11] observed that misalignment affects major bearing performance parameters such as cavitation, thermal field, leakage flow-rate, and moments. Guha [12] observed that the load carrying capacity decreases with an increase in surface roughness parameter and it increases with degree of misalignment.

In order to increase the reliability of the bearing system, Hu et al. [13] was the first to do a robust design of experiments using manufacturing process variables, to reduce the variation in journal lobing. As the next step, they performed robust design of experiments using the lobe geometrical parameters as control factors and number of lobes as internal noise, to obtain the optimized film thickness [14]. In real life applications, designers are usually constrained by different operational requirements. Also, getting a true optimum solution is difficult and expensive, while applying One Variable At a Time (OVAT) approach [15]. Designers also have less or no control over environmental noises and system interaction noises. In author's knowledge, there is no research publication, where the robustness of the spiral grooved hydrodynamic journal bearing was studied for the impact of both external and internal noises.

The main objective of this paper is to describe the selection of the robust design parameters for a Spiral Groove Journal Bearing, which not only meet the load carrying capacity requirements, but also have the minimum sensitivity to both internal and external noises such as clearance, misalignment and temperature of the lubricating fluid. Taguchi's cross orthogonal array Design of Experiment (DOE) procedure [16, 17] is used to arrive at a robust solution.

SPIRAL GROOVE JOURNAL BEARING

A schematic of a Spiral groove journal bearing is shown in [Figure 2](#). The main geometrical parameters for a spiral groove journal bearing are- the diameter and length of the bearing, as well as the width, height and pitch of the groove.

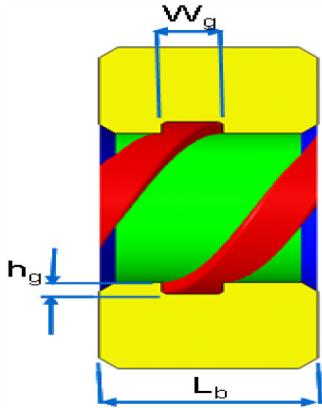


Figure 2. Spiral Groove Journal Bearing

For the robust design process, all the operational requirements and design variables were first obtained, and then classified as per their impact on the Journal bearing performance.

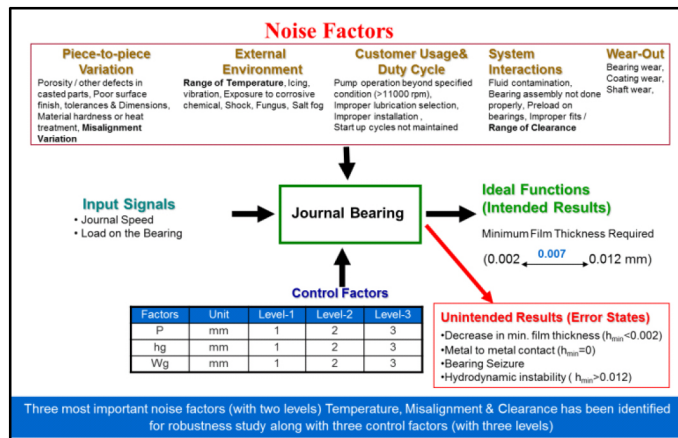


Figure 3. Parameter Diagram for Journal Bearing

[Figure 3](#) shows the Parameter Diagram (P-Diagram) for a journal bearing. Three important noise factors - clearance, misalignment and temperature, each with two levels, were selected for the Robustness study. The control factors selected were- Pitch (P), Height (h_g) and Width of the groove (W_g) with three levels each. Noise factors are identified based on the factors that cannot be controlled or are expensive to control. Temperature levels are based on customer requirements for the bearing operating conditions, while the clearance is the difference between the shaft and bearing radius. The levels for clearance are determined from the tolerances upon the shaft and bearing radii and also the effect of temperature is considered. Misalignment is calculated from the length of the shaft and maximum load coming on the shaft. [Equation 4](#) in the paper describes the formulation to determine the misalignment.

For a given bearing geometry, both the load carrying capacity and life requirements are controlled by minimum film thickness (h_{min}). Thus, h_{min} with a set of specification limits was chosen as an Ideal function. Upper limit for h_{min} is considered based on the hydrodynamic stability and lower limit is considered based on the surface roughness and the maximum allowable fluid pressure developed in the bearing, which will not cause any deformation in the bearing material. The target value is assigned so as to maximize the heat transfer. In this case, the upper limit for h_{min} is 0.012 mm, the lower limit is 0.002 mm and the target value is 0.007 mm. The input signal for this bearing is the journal speed and the load on the bearing.

Conventional Design Approach

Conventional design approach considers following steps:

- Determination of radial load and misalignment for the bearing
- Selection of bearing type based on cost, operational requirements and environmental factors
- Determination of basic bearing parameters based on analytical calculations and data history
- Optimization of bearing geometrical parameters through iterative process using Thermo hydrodynamic analysis

Thermo hydrodynamic analysis methodology used in this paper for Journal bearing analysis is validated with the test data reported in Ferron et.al. [18]. [Figure 4](#) shows the comparison of results from thermo hydrodynamic analysis used in this paper with test results reported in Ferron et.al. [18].

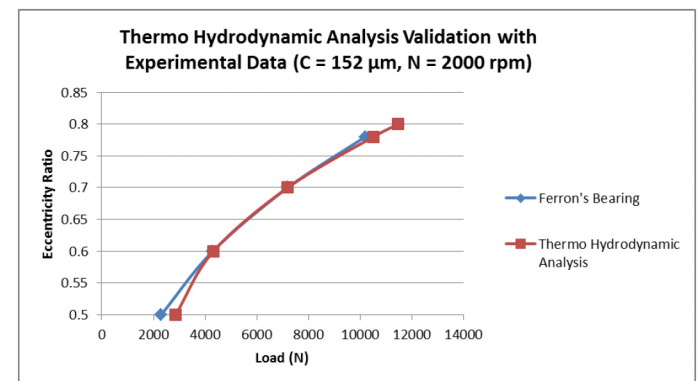


Figure 4. Thermo hydrodynamic analysis methodology validation with test results

[Figure 5](#) shows the free body diagram of a two stage centrifugal pump which shows different loads coming onto the journal bearing. Each stage of the pump is located at either end of the shaft with the motor in the middle. The total load transferred to the bearing or the load required to be carried by the bearing can be given as:

$$F_R = (W_t + F_t) + \frac{W_m + W_s}{2} \quad (3)$$

where,

W_i is the weight of the impeller,

W_m is the weight of the motor,

W_s is the weight of the shaft,

F_i is the radial load.

Radial load is a function of the pump geometry, discharge flow and fluid properties, and can be determined empirically from [18]:

Misalignment in the bearing can be defined as being made up of two components as follows:

$$\gamma = \gamma_{load} + \gamma_{mfc} \quad (4)$$

where,

γ is the total misalignment of one bearing,

γ_{mfc} is the misalignment due to the manufacturing process and assembly of the pump,

γ_{load} is the misalignment caused by load obtained by solving the following equation:

$$EI \frac{d^2 y}{dx^2} = -(W_i + F_i)x + F_R(x - L_1) - (W_m + W_s) \left(x - L_1 - \frac{L_2}{2} \right) + F_R(x - L_1 - L_2) \quad (5)$$

where,

L_1 and L_2 are the distances in mm.

Once the load and misalignment are known, three dimensional Thermo hydrodynamic performance analyses are performed to calculate the pressure and temperature distribution. On the fluid film, wherever the pressure falls below the vapor pressure, it is equated to zero (Figure 6). The load carrying capacity is then calculated by surface integral of the pressure distribution using Equations 6, 7, 8:

$$F_R = - \iint P \cos \phi dx dz \quad (6)$$

$$F_T = \iint P \sin \phi dx dz \quad (7)$$

$$W = \sqrt{F_R^2 + F_T^2} \quad (8)$$

where,

F_R and F_T are the radial and tangential loads,

W is the resultant reaction of the fluid film in the bearing and

Φ is the bearing angle.

This procedure is repeated for different eccentricity ratios. The equilibrium eccentricity ratio is then calculated, when the reaction load developed by the fluid film (Equation 8) equals the load requirement for the bearing [18]. The minimum film thickness is then computed using Equation 9:

$$h_{min} = (c * (1 - \epsilon)) - \frac{L_2}{2} * \tan(\gamma) \quad (9)$$

where,

ϵ is the equilibrium eccentricity ratio $= e/c$,

e is the equilibrium distance between journal center and bearing center,

L_2 is the length of the shaft between the two bearings.

Before performing the design optimization study, the developed methodology was validated with the experimental data [19] and analytical results [20, 21].

In the conventional approach, bearing geometrical parameters (control factors) are iteratively selected based on experience on a similar pump and evaluating the bearing performance at the worst case condition; such as temperature, clearance and misalignment using Thermo hydrodynamic analysis or Iso-viscous analysis with temperature correction. The problems observed with the above methodology are:

- Film thickness varies largely with the environmental conditions,
- Low field reliability of the bearing,
- Expensive and time consuming development cycle,
- Requirement of very tight tolerances on the control factors to obtain an acceptable robustness level.

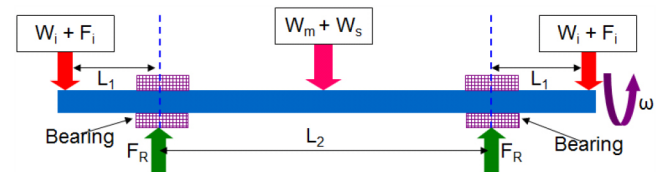


Figure 5. Loads on Journal Bearing