A brief review on Optimum Design of A Journal Bearings for I.C. Engine

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Abstract:

The ever-increasing demand of lower initial and running costs for bearings, to withstand competition, has prompted engineers to apply optimization methods in bearing design. The optimization scheme that minimizes power loss , minimization of temperature rise and side leakage in journal bearing applications in machinery like diesel engine which operating in laminar flow is reviewed and referred in this paper. The power loss equation is derived and shown to give good results when simulating data from the literature. A hybrid optimization technique that uses a genetic algorithm to converge to the global minimum and sequential quadratic programming for accurate location of the minimum point is used. The method gives consistently good predictions and has been verified by comparing predictions to available data in the literature.

This paper attempts to solve the difficulty of selecting the deciding factor in the condition describing more than one objective bearing optimization problems. The objective of bearing optimization is to minimize a combination of power loss and oil flow. Constraints are imposed on the maximum fluid pressure, minimum film thickness, maximum temperature rise and critical speed. Numbers of researchers have been working on various aspects of performance of the journal bearing, ranging from temperature rise, geometry of grooves, instability, clearance etc. Here a brief review is presented of the literature available in recent years. Since there have been lot of developments in computer hardware and software technologies, the parametric optimization has been performed by many researchers.

Keywords: objective optimization, journal bearing design, generic algorithm Hybrid technique, Side leakage, power loss, bearing oil temperature

1 INTRODUCTION

Sliding bearings are so much in use today, which means they are applied to most of machines that we need and required efficient, unique and cost effective system. This can be understandable because of some advantages that this sort of bearing has compared with rolling bearings. Generally, their production is not so complicated which makes the price lower for simple mounting they can be made in parts and in operating they produce less noise and vibration. In case of correct lubrication all sort of sliding bearings are very practical for maintenance and they have long operating life which is probably most important reasons for their common use. Especially, self lubricating sliding bearings are very useful in the new age and there are two different sorts of them.

Hydrodynamic principles, which are active as the shaft rotates, create an oil wedge that supports the shaft and relocates it within the bearing clearances; a shaft spinning within a journal bearing is actually separated from the journal bearing's metal facing by an extremely thin film of continuously supplied oil that prohibits metal to metal contact. Journal bearings shown in figure no.1(a) and (b) have many designs to compensate for varying load requirements, machine speeds, cost, or dynamic properties . A stable bearing design holds the rotor at a fixed attitude angle during transient periods such as machine start-ups/shutdowns or load changes.

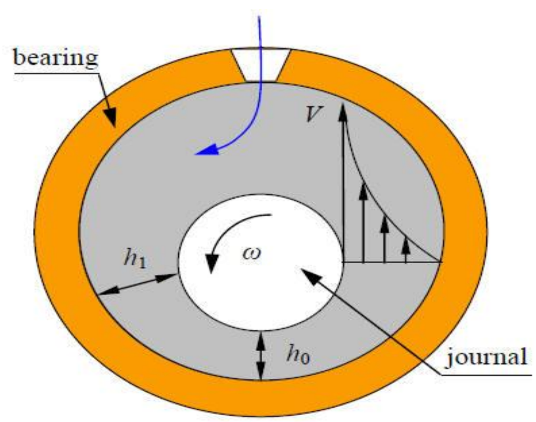

Figure : 1(a)

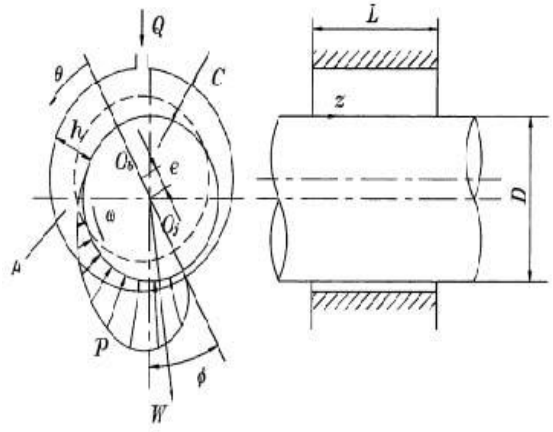

Figure : 1 (b)

The high demand for ideal self-acting journal bearings, particularly in Diesel engine, centrifugal compressors, pumps, motors, etc., is due to their excellent properties, such as exceptional damping, long-term performance, negligible friction and almost zero wear. The design of such an ideal bearing relies on thin-film hydrodynamic lubrication theory, which is concerned with separation of two surfaces in relative motion. The performance of a hydrodynamic journal bearing is governed by a number of bearing parameters, such' as radial clearance. length, diameter, viscosity, groove geometry, location, etc. There are several manual procedures available for designing a bearing. Generally, the selection of design variables in the bearing design is done by a trial-and-error method using many design charts, but there is no way. of ensuring that the resulting design has optimum characteristics.

The ever-increasing demand of lower initial and running costs for bearings, to withstand competition, has prompted engineers to apply optimization methods in bearing design. To reduce computational effort in optimization, a number of researchers have optimized bearing dimensions using only one-dimensional cases, which are far from reality. McAllister and Rohde optimized the load-carrying capacity of one-dimensional journal bearings for a given minimum film thickness. They used a long bearing approximation, which is inaccurate in most practical design ranges.

2 LITERTURE REVIEW

Myung-Rae Cho et al [1] presented the effects of circumferential groove on the minimum oil film thickness in engine bearings. The fluid film pressures are calculated by using the infinitely short bearing theory for the convenience of analysis. Journal locus analysis is performed by using the mobility method. A comparison of minimum oil film thickness of grooved and ungrooved bearing is presented. It is found that circumferential 360° groove only reduces the absolute magnitude of the oil film thickness, but 180° half groove affects the shape of film thickness curve and position of minimum oil film thickness.

Koichi Matsuda et al [2] suggested clearance configuration of fluid-film journal bearings and optimized in a sense of enhancing the stability of the full circular bearing at high rotational speeds. A performance index is chosen as the sum of the squared whirl-frequency ratios over a wide range of eccentricity ratios, and a Fourier series is used to represent an arbitrary clearance configuration of fluid-film bearings. An optimization problem is then formulated to find the Fourier coefficients to minimize the index. The designed bearing has a clearance configuration similar to that of an offset two-lobe bearing for smaller length-to-diameter ratios. The load capacity of the designed bearings is nearly in the same magnitude as that of the full circular bearing for smaller length-to-diameter ratios. The designed bearings successfully enhance the stability of a full circular bearing and are free from the whirl instability.

M. El-Sherbiny et al [3] formulated optimization problem to maximize the load-carrying capacity of hydrostatic journal bearings. Optimization process is based on the well known Rosenbrock method and from results concludes that precision bearings with small clearances and low pressure ratios are recommended for applications involving low supply pressures, while bearings with large clearances and pressure ratios close to 0.5 are recommended for applications involving high supply pressures.

S.M.Ghoneam, S.Strzelecki [4] derived operation of plain circular journal bearing at high speed is restricted by the excessive temperature that is generated in the oil film and by the loss of stability. However, low costs of machining and high transmitted loads are the advantages of these types of bearings. The operation of bearing and its thermal state are effected by modifications of bearing design. These modifications should result in the higher speeds of operation and better thermal state of bearing. There is small effect of axial profile variation coefficient on the Somerfield numbers and the static equilibrium position angles. Axial profile variation coefficient causes the small decrease in the oil film temperatures.

H. Hirani et al [5] have presented the optimum design methodology for improving operating characteristics of fluid-film steadily loaded journal bearings. A finite difference mass conserving algorithm is used to provide relatively accurate power loss and flow rate. Pareto optimal concept to avoid subjective decision on priority of objective functions, a genetic algorithm to deal with multimodal nature of hydrodynamic-bearing and develop a Pareto optimal front, fitness sharing to maintain genetic diversity of the population used in genetic algorithm, and axiomatic design to provide inside of objective functions and design variables. Algorithm for the optimum design of short journal bearings. The optimized results were compared with those of genetic algorithm and successive quadratic programming. All the results have the same tendency. The artificial life algorithm only uses the function value and doesn't need derivatives calculated analytically or numerically and so it has a strong possibility for being used for other optimization problems.

K. Matsumoto et al [6] have described the optimum design methodology, hybrid optimization technique combining the direct search method and the successive quadratic programming and is applied effectively for the optimum solutions. It determines to minimize the objective function defined by the weighted sum of maximum averaged oil film temperature rise, leakage flow rate, and the inversion of whirl onset speed of the journal under many constraints.

S.M.Ghoneam, S.Strzelecki [7] derived operation of plain circular journal bearing at high speed is restricted by the excessive temperature that is generated in the oil film and by the loss of stability. However, low costs of machining and high transmitted loads are the advantages of these types of bearings. The operation of bearing and its thermal state are effected by modifications of bearing design. These modifications should result in the higher speeds of operation and better thermal state of bearing. There is small effect of axial profile variation coefficient on the Sommerfeld numbers and die static equilibrium position angles. Axial profile variation coefficient causes the small decrease in the oil film temperatures. There is effect of axial variation profile coefficient on the profiles of axial oil film temperature distributions.

Hiromu Hashimoto [8] has optimized the design procedure of high-speed, short journal bearings under laminar and turbulent flow conditions developed based on Successive Quadratic Programming, Genetic Algorithm and Direct Search method. Short bearing assumption to the modified turbulent Reynolds equation, simplified closed form design formulae are obtained for the eccentricity ratio, friction force, film temperature rise, supply lubricant quantity and whirl onset velocity as a function of design variables such as radial clearance, slenderness ratio and averaged viscosity of lubricants. Design variables, which optimize the objective function given by a linear summation of temperature rise and supply lubricant quantity with respective weighting factor, are determined for a wide range of journal rotational speeds under various constraints.

D. Dowson et al [9] gave solutions of the Reynolds equation which have been computed for a bearing configuration which received little attention theoretically. The optimum design objective is stated explicitly in terms of the operating characteristics and is minimized within both design and operative constraints and, also form the basis for the development of a computerized optimum design technique.

A. Seirig et al [10] have presented an automated system for the selection of the length, clearance and lubricant viscosity which optimize the performance of a hydrodynamic journal bearing under specified values or ranges of load and speed . The feasibility of applying optimal programming techniques for the development of bearing design is highly dependent on the merit criterion; careful attention must be given to the selection of the criterion which best suits the particular circumstances of the design. This step of the design process relies mainly on the judgment and experience of the designer.

Boedo and Eshkabilov [11] implemented a genetic algorithm scheme to optimize the shape of fluid film general bearings under steady journal rotation. They considered only one objective function, namely the maximization of load capacity. However, the optimum design of a journal bearing is a multi objective problem which must be solved with multiple objectives taking into consideration design constraints. Each objective function (i.e. minimization of temperature rise, minimization of oil feed flow, minimization of power loss) has a different individual optimal solution. Multiple optimal solutions exist because no one solution can be optimal for multiple conflicting objectives.

To reduce computational effort in optimization, a number of researchers have optimized bearing dimensions using only one-dimensional cases, which are far from reality. McAllister and Rohde [16] optimized the load-carrying capacity of one-dimensional journal bearings for a given minimum film thickness. They used a long bearing approximation, which is inaccurate in most practical design ranges. A recent paper from Hashimoto [8] provides a good review of work related to journal bearing optimization. In that paper Hashimoto presented an optimum study for high speed short journal bearings using successive quadratic programming. For Eccentricity > 0.8 and L/D > 0.3, the short bearing approximation predicts highly unreliable results. Therefore, using the short bearing approximation for any journal bearing optimization can never be justified. For that reason the present paper uses a hybrid solution scheme [12], which is a harmonic summation of short and long bearing approximations and provides reliable pressure solutions [12].

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Daniel W. Boeringer et al [14] suggested Particle Swarm Optimization (PSO) which requires less computational bookkeeping and generally only a few lines of code. A particle swarm optimizer is implemented and compared to a genetic algorithm for phased array synthesis of a far-field side lobe notch, using amplitude-only, phase-only, and complex tapering. The results show that some optimization scenarios are better suited to one method versus the other (i.e. particle swarm optimization performs better in some cases while genetic algorithms perform better in others), which implies that the two methods traverse the problem hyperspace differently. The particle swarm optimizer shares the ability of the genetic algorithm to handle arbitrary nonlinear cost functions, but with a much simpler implementation it clearly demonstrates good possibilities for widespread use in electromagnetic optimization

Bo-Suk Yang et al [15] have used enhanced artificial life algorithm for the optimum design of short journal bearings. The optimized results were compared with those of genetic algorithm and successive quadratic programming. All the results have the same tendency. The artificial life algorithm only uses the function value and doesn't need derivatives calculated analytically or numerically and so it has a strong possibility for being used for other optimization problems.

Most of the available studies [2, 3, 11,16-21] treat optimization of journal bearings as a continuous variable optimization problem; however, many of the bearing variables are discrete in actual applications. For example, available oil viscosity does not come in the continuous range, as illustrated in Table 1. Designing a bearing radial clearance with a resolution of less than 1 /im does not make any sense. Similarly, resolution in a slenderness ratio better than 0.01 increases the manufacturing cost. Therefore this paper deals with discrete data, which are more common in practice. .

Table 1 ISO viscosity grades

ISO (International Organization for Standardization) viscosity grade numbers	Viscosity grade ranges (cSt at 40 °C) Minimum	Maximum
2	1.98	2.42
3	2.88	3.52
5	4.14	5.06
7	6.12	7.48
10	9.0	11.0
15	13.5	16.5
22	19.8	24.2
32	28.8	35.2
46	41.4	50.6
68	61.2	74.8
100	90	110

$1 \text{ cSt} = 10^{-6} \text{ m}^2/\text{s}$.

3 PROBLEM FORMULATION

From a designer's perspective, a journal bearing must support the applied load by occupying minimum space with minimum energy loss and minimum wear. Prediction of the load carrying capacity requires knowledge of film geometry and the solution of the pressure developed inside the journal bearing. Energy loss can be estimated if friction force and velocity are known. Wear can be minimized by putting an upper limit on the maximum fluid film pressure and maximum temperature rise, and a lower limit on the minimum fluid-film thickness. To facilitate the bearing design, Hirani et al. [12] presented various formulae in tabular form. This table assumes eccentricity as an input parameter and calculates all other state variables in terms of the eccentricity ratio. In reality the bearing design is an inverse problem, where load is given as an input parameter and the eccentricity ratio is to be determined. To deal with such inverse problems a curve-fit equation suggested by Hashimoto [8] may be used:

$$\varepsilon(x) = \exp\left(-2.236\Lambda\sqrt{\frac{n_s \mu D^4 \Lambda}{48 C^2 W}}\right)$$

In the present work, above equation is used for an initial approximation of eccentricity ration, which is updated to the correct eccentricity ration. The journal bearing model considered here for the three important parameters for the optimization point of view. They are Side leakage, power loss and temperature rise in the journal bearing design considerations. These three parameters are important and play vital role in the performance of journal bearing.

4 CONSTRAINTS

Constraints on the main design parameters are the essence of the optimum design. Every design variable has lower and upper bounds. In this paper the lower limit of oil viscosity is kept as 10-3 Pas and the upper limit as 0.1 Pas. Similarly, the lower bound of clearance is 40 µm and the upper bound of clearance is 295 µm. The slenderness ratio ranges from 0.2 to 0.83. Restrictions can be imposed on the minimum film thickness depending on misalignment conditions and surface roughness of mating surfaces. Generally it is recommended to keep the minimum film thickness more than 5-10 µm. Constraint on the maximum film pressure is imposed due to limited material strength, say 1OMPa for Babbitt material and 20MPa for copper lead.

(I) SIDE LEAKAGE:

Side leakage is determined analytically using the methods proposed by Martin [21] and used by Khonsari and Booser [22]. In their approach the flow is composed of two components, QI due to the pumping action of an eccentric shaft rotating within the sleeve and a pressure-induced component, QP due to the supply pressure. A leakage flow determination scheme that takes the supply pressure ps into account is proposed. QL is determined first

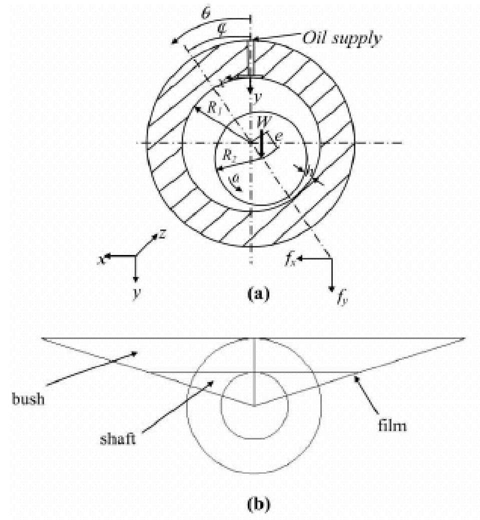

Figure – 2

(a) Bearing Geometry & Nomenclature

(b) Unwrapping of bearing. The cross-sectional areas of the two regions are equal.

QL = πNsCDLQL ….(1)

which is similar in form to Martin's equation, QL = fLπ LDNS C, where fL = f(ε) is a correction factor for the film start position (included in the term QL in equation.

QL = exp[0.20 - 0.500 γ1.5 + 1.05 ln(ε)]
0.5 ≤ γ≤ 1.5, 0.1≤γ≤0..9 ….(2)

where the coefficients were determined empirically by Gadala and Zengeya [24] who simulated pressure-fed bearings in the literature and did curve-fitting to the data given that QL = f(ε, γ) Results from equation (2) consistently gave better predictions compared to Jang's equation for the bearings simulated.

Leakage due to supply pressure QP is determined as proposed by Martin [22]. A brief description of the method is included here for continuity. For the case where the oil groove is opposite the load line (Fig.2l(b)) Qp =fg(l + ε cosØ)3(psC3/µ), where fg is a groove function which depends on the groove location and dimensions. Total leakage is determined by finding a datum flow, Qm, which calculates the flow rate by ignoring oil film continuity Qm = QL + QP – 0.3(QLQP)0.5, then corrected for film reformation. Total leakage is determined as QLtot. = QmSQp1-S where S = 0.7(Lg/L)0.7 + 0.4 for an axial groove of length Lg.

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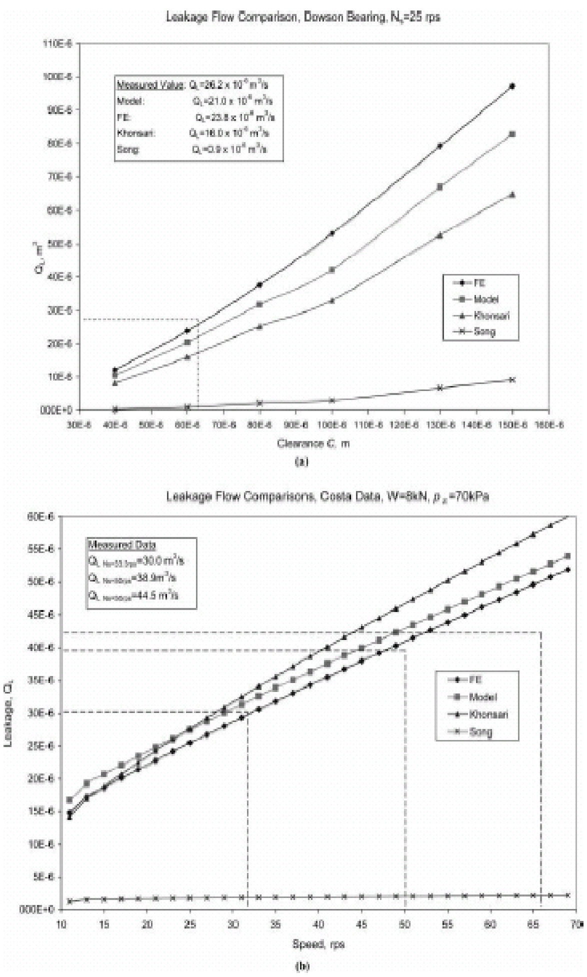

Figure 3
(a) Leakage flow QL as function of radial clearance C, Dowson's data (C=0.63.75 μm)
(b) Leakage flow simulation of Costas's data [26], W=kN, ps = 70 kPa

5 POWERLOSS

Power loss computations present one of the main focuses here. Hirani and Sun [5] use a mass conserving finite difference (FD) scheme to solve for the power loss from the equation PL = $f(\mu U/h + (h/2)(\partial P/R\partial\theta))URd\theta dz$. This is good and reliable except that it involves more computational effort than the proposed method. Although software for industrial applications normally include FD integration techniques, integration with optimization programs involves more effort than a straight application of the equations presented here. Power loss is determined by finding the frictional force Ffr from the relation Ffr = f w, where f is the coefficient of friction and W is the load on the bearing. Power loss becomes PL = Ffr(2πRNs). The coefficient of friction is evaluated from (R/C)f which is a function of γ and ε

R/C)f = exp(1.9994+(2.73622/(γ)0.5)-4.97838(/(γ)0.5) …(3)

where the coefficient values in equation (3) are determined empirically by simulating data in the literature and using curve-fitting techniques.

The equations presented in this section are implemented in Matlab [27] using the hybrid optimization technique. The procedure involves using the genetic algorithm (GA) for rapid convergence to a global minimum and SQP for an accurate minimum point prediction. The genetic algorithm toolbox with the Hybrid option was used. The fitness function is the objective function defined in section sequential quadratic programming and non-linear constraints are the same as in that section.

6 OPTIMIZATIONPROCEDURE

The procedure presented here will help designers to implement the optimization model proposed. This can be easily done in Matlab. By using Sequential quadratic programming (SQP). Techniques and Matlab software it can be easily programme. This SQP method requires three main components, namely, the main program, objective function, and non-linear constraint m-files. The call to the optimization subroutine 'fmincon' is done in the main program, while computations in part B are done in the objective function m-file and part C computes the non-linear constraints. Define global variables that will be passed between the three m-files (ΔT,ε, Ns). The three m-files are attached at the end of the appendix. While care was taken in preparing these files, no guarantee is given that the codes are error-free. A brief description of the main parts follows. Create a folder and copy the three files. Run the main program, which will call the other m-files.

Part A: Main Program

-Define upper and lower limits for design variable XT = (C, γ, NS) and provide these as lb and ub in the call to 'fmincon'

-Provide program with initial guess for design variable x0

-Set options and call 'fmincon' (this calls both Part B and C)

Part B: Objective Function

Set all the problem parameters here (since the hybrid technique will use same objective function)

* For Ns from Ns1 to Nsmax
Step 1: Compute journal surface velocity U
2: Use μ to compute Somerfield number S
3: Find eccentricity ratio ε
4: Compute the temperatures Tmax and Tshaft and temperature increase Δ T
5: Update the viscosity μ
6: Go to step 2 and re-compute S until convergence.
7: Compute the side leakage QL
8: Compute the objective function F(X) given the constraints gi(X)
9: Record F(X)
10. Loop and update Ns
Part C: Non-linear Constraint Computation
-Define minimum allowable hmin, ΔTa, wcr, and pa
-Compute pmax,w, and wcr
-Define non-linear constraints g1, g2, g3, and g4

RESULT:

A computer program has been developed based for bearing optimization on the logic described in the previous sections. A journal bearing of diameter 0.1 m lubricated with a lubricant of density 860kg/m3 and specific heat 4190J/kgK is being analyzed. Three different cases of objective function, minimizing temperature rise, minimizing oil flow and minimizing the weighted sum of temperature and flow, are analyzed. The results of these cases are presented in Table 3. In Table 3, the optimized design vector consists of radial clearance (in µm), slenderness ratio and oil viscosity (in 10-3 Pas). Results for temperature (in degrees Centigrade), power loss (in W), maximum pressure (in MPa), minimum film thickness (in µm), oil flow (incm3/s) and critical speed (in r/s) are evaluated for different cases of load and speed. It is interesting to note that increasing the speed from 40 to 330 r/s reduces the required oil viscosity to 25 per cent. It reduces the critical speed by 20 per cent, increases flow by 7.5 times and increases power loss by 11.5 times.

Increasing the load from 10 to 20 kN hardly affects the optimized design vector, but increases maximum pressure, reduces film thickness, increases power loss and increases oil flow. Bearing materials generally have a limited strength, which fixes the allowable maximum pressure to be carried by the bearing. Similarly, for reliable hydrodynamic operation, a lower limit is imposed on minimum film thickness. Therefore it is customary to impose constraints on heavily loaded bearing optimization problems. The results of Table 2 show that by constraining the upper limit of maximum pressure as 10MPa, optimized clearance decreases and viscosity achieves a higher value. This change in design variables is helpful for maximum pressure, minimum film thickness and oil flow, but has disadvantages for power loss and critical speed.

Treating minimization of oil flow as an objective function provides very different results. It decreases clearance significantly and increases oil viscosity. The effect on temperature rise and power loss is significant. Minimizing temperature rise and oil flow simultaneously makes the bearing problem a multiobjective function optimization problem.

In such problems, the scaling factor and weight ratio play very important roles. On examination of earlier results 104 can be recommended as the scale factor for oil flow. Three different case were studied. A substantial reduction in friction loss and oil leakage is reported in all three cases. These results are very encouraging and give a motivation for the use of the proposed optimization methodology for every journal bearing design.

MULTIOBJECTIVE OPTIMIZATION OF A JOURNAL BEARING

Table 2 Comparison of optimized results for various objective functions

		Optimized vector			Results					
Objective function	Parameters/ constraints	C_r	Λ	μ	ΔT	Power loss	P_{max}	h_{min}	Flow	Critical speed
Minimum temperature rise	$N = 40.0$ r/s Load = 10 kN	295	0.83	4	0.2	211	9.07	6	2.85	9065
	$N = 330$ r/s Load = 10 kN	295	0.81	1	0.3	2617	7.5	10.7	22	7189
	$N = 330$ r/s Load = 20 kN	295	0.83	1	0.4	3538	17.9	6	24	8893
	$N = 330$ r/s Load = 20 kN $P_{max} < 10$ MPa	255	0.83	5	1 3	9463	9.987	26	18	3345
Minimum oil flow	$N = 330$ r/s Load = 20 kN $P_{max} < 10$ MPa	44	0.55	46	22 5.7	1.8778×10^5	6.57	40	0.18	1337
Minimum temperature and flow	$N = 330$ r/s Load = 20 kN $P_{max} < 10$ MPa	121	0.72	3	2.7	8028	9.658	23	6.69	2089
	$N = 40$ r/s Load = 20 kN $P_{max} < 10$ MPa	156	0.82	17	1.4	741	9.902	17	1.31	3952

7 CONCLUSIONS

An evolution optimization methodology for full journal bearing has been applied. The effect of minimizing temperature rise, minimizing oil flow, minimizing the weighted summation of oil flow and temperature rise, and linear combinations of minimizing oil flow and power loss under different constraints are demonstrated. The results lead to the recommendation of using simultaneous minimization of oil flow and power loss. Pareto-optimal concepts are utilized to help the win-win situation between power loss and oil flow. Optimization studies with three design variables are performed. An orthogonal array is used in selection and setting of the values of design variables and tolerances. The results of Tables 2 indicate that the oil viscosity is of the highest importance. Therefore, during bearing design special attention should be given to lubricant viscosity.

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APPENDIX I FOR NOTATION:

C	Radial clearance (m)
Cp	Specific heat of lubricant (J/kgoC)
D	journal diameter (m)
L	Length of journal (m)
E	Eccentricity (m)
ΔT	fluid film temperature rise (oC)
Ec	Eckert Number
W	Load applied to bearing (N)
S	Somerfield Number
μ	Average viscosity of the lubricant (Pa s)
hmin	minimum film thickness(m)
w	Angular velocity of journal
lb	Lower bound
F	fluid film force (N)
ub	Upper bound
g	Gravitational acceleration (m/s2)
pmax	Maximum allowable pressure (Pa)
objf	Objective function
NS	Journal speed, revolution/second
R	Radius of the journal (m)
QL	Leakage flow rate (m3/s)
ε	Eccentricity ratio
U	Shaft surface velocity (m/s)
X	Vector of design variables
γ	Length/ Diameter ratio
gi	Non linear constraint function