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Optimal profiles for one dimensional slider bearings under technological constraints

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The optimal slider bearing profile for maximum bearing load is studied by using direct constrained optimal control techniques. The constraints include the Reynolds and the energy equations. The energy equation takes into account the shear strain rate in the lubricant. The dependence of lubricant viscosity on temperature is considered. Technological constraints such as the maximum lubricant pressure and temperature and the minimum lubricant film thickness are included into the model. The realistic problem considered here yields optimal bearing profiles which are much more complex than the classical Rayleigh step bearing profile. The optimal bearing profile consists of an alternation of regions of constant height and more or less abrupt height variations. The number of constant height regions depends on the type of the constraint and in many cases is larger than three. The minimum value of the bearing height is one of the most important constraints. Four levels of model approximations have been tested. The most important model improvement is to take into account the temperature dependence of the lubricant viscosity. Several bearing design and operation parameters, such as bearing length, inlet height, sliding velocity and lubricant inlet pressure and temperature, have been considered. They all have complex influence on the optimal bearing profile.

1. Introduction

Bearings are used to reduce the frictional losses between two rotating or sliding mechanical parts. The load carrying surfaces are completely separated by a lubricant film, eliminating the risk of surface wear as long as a film of sufficient thickness is maintained [1,2]. In practice, slider bearings are designed for supporting transverse load, the lubricating pressure being generated by the lateral motion of two surfaces which are not quite parallel [3,4]. Pad bearings are widely used to support axial thrust loading in various rotating machinery. The majority of such bearings in industrial applications employ centrally pivoted plane-pads for reasons of easy manufacturing and reversal operations [5]. Thrust bearings are used to carry loads in applications where roller bearings are unsuitable due to dimensional limitations, demands for operational lifespan, high loading requirements or difficult access for mounting [2,6]. Slider bearings are widely used in the transmission systems of many engineering applications like mechanical seals, machine tool ways, piston rings, plain collar thrust bearings and computer hard disks for their load-carrying capacity, excellent stability, and durability [3,7–9].

A great deal of emphasis was placed on the effectiveness of the bearing geometry to generate pressure (self-acting bearings) and thus increase load capacity [10]. The geometry of the contacting elements determines the shape of the lubricant film. Various researchers have considered different configurations of the lubricating film in the clearance zone in their analysis [11]. Eight typical geometries can be seen in Fig 27.6 of [10]. They include partial arc bearing and the wedge-shaped configuration used in the classical Kingsbury and Michell thrust bearings. Other geometries are the parabolic and the exponential profiles [4,11,12]. The inclined shaped slider has been analyzed in most studies [1,2,9,12–14]. A few studies have concentrated on waviness profiles (see [15] and references therein).

Two types of slider bearing optimization problems have been studied during the time. The first problem involves finding optimum design parameters for given bearing profile. The second, more difficult problem, is to finding the optimal bearing profile.

The first problem mainly focused on bearing profiles used in practice. The optimum slope of an inclined bearing that provides a maximum load-carrying capacity has been studied in many papers (see e.g. [5,10]). The optimum performance has been studied for

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other bearing shapes (see [4,16] and references therein). Advanced optimization tools such as sensitivity analysis and genetic algorithms are used in the last years in connection with more complex bearing shapes [15,17,18]. Various profiles and texture types are adopted and the optimum geometrical parameters have been obtained considering different performance criteria [18].

The second problem started with the seminal work of Rayleigh [19]. He showed that there exists a profile with a simple jump discontinuity which supports more weight than do profiles of the form $1+mx^n$ or $e^{\beta x}$ with m, n and β positive [20]. This geometry eventually became known as the Rayleigh step bearing; it consists of two parallel surfaces — one having a rectangular cross-sectional dam. Rayleigh tried to find the profile which supports the greatest weight but he only found a profile which caused the first variation of the weight functional to vanish. The problem has been fully solved in [20]. Using the calculus of variations, one can verify that the stepped bearing is the optimum unconstrained solution for a one-dimensional slider [10]. An algorithm to optimize the shape of a 3D square slider bearing has been derived in [21]. The optimum shape was found to be close to trapezoidal pocket geometry [8,21]. A variational technique has been used in [22] to obtain the bearing profile which maximizes the load carrying capacity of an infinite length journal bearing. The solution is a concentric step bearing.

The step bearing has been first applied in practice in 1950 to sector-shaped thrust bearings and later to journal bearings. However, the marketplace has been captured by the Kingsbury and Michell tilting-pad bearing, despite they are more difficult to fabricate and they have 20% lower load capacity than the infinitely wide step bearing [10]. Continuous demand from industry requires lighter components with longer lifetimes, less power loss and lower lubricant consumption. An alternative that can improve bearing performance without changing the operating conditions or lubricant properties is to introduce modifications in the geometry of the contact. This can be done via new surface shapes (such as a step or a texture [8] or innovative design solutions [15]). Rapid developments in computer capabilities have enabled numerical research. Designs tend to be more and more aggressive with lower margin for error [8].

In this paper we report new results concerning the second optimization problem, i.e. the profile optimization of one-dimensional sliding bearings. Our approach brings several novelties in respect to previous "full" shape optimization approaches. First, limitations of technological and operational nature are taken into account. Such limitations do always exist in practical applications [18]. From a mathematical point of view they are associated with constraints. Constrained optimization solutions are more complex and difficult to find than the unconstrained solutions (such as the Rayleigh step slider bearing) obtained in previous studies. Second, powerful theoretical tools such as direct optimum control methods are used. These methods allow treating constrained optimization problems. They were not available to early researchers and are more adaptive to practice than the variational techniques used later. Third, thermal effects are taken into consideration. Optimization of fluid film bearings with thermal effects is complicated since the film temperature changes along the bearing [16]. The energy equation is based on the model proposed in [5]. That model is further developed here and the shear strain rate in the lubricant is included into the analysis. This effect is not very often considered and is even less often used in optimization approaches. Fourth, the dependence of the lubricant viscosity on temperature is explicitly taken into consideration. This is a complication in optimization studies. Sometimes it is avoided by using effective lubricant viscosity based on the overall temperature raise of the lubricant [16,22]. Fifth, industrial bearings are usually based on "pure" sliding, due to relative rotation and/or normal motion [10]. External pressurization is sometimes needed to keep the bearing surfaces separated during start and stop conditions to avoid surface wear and reduce frictional energy loss. The two pressure-generating schemes are occasionally combined in "hybrid" bearings, which are considered here.

2. Model

One dimensional (1D) bearings are considered here. Of course, 2D modeling is to be preferred instead of 1D modeling, when possible. This is not (yet) the case with optimal control modeling. 1D optimal control has a solid background provided by Bellman–Jacoby theory and the powerful Pontryagin Maximum Principle. On the other hand, 2D optimization is much more involved. Several artificial intelligence-inspired techniques (such as genetic algorithms [17]) and parametric studies [23] have been used but they optimize in fact pre-defined classes of bearing profiles. 2D optimal control belongs to the category of optimal control with distributed parameters. This branch made significant progress in the recent years but a mature theory is still missing.

Several hypotheses are usually adopted during bearing optimization. For instance, infinitely wide bearings (i.e., no side leakage) are considered in many papers (e.g. [10,16]) and this is our assumption. A Newtonian, incompressible, isoviscous, lubricant under steady state operation with fluid inertia effects neglected is studied in [16]. The same hypotheses are adopted here but the lubricant viscosity depends on temperature. Several simplifications have been adopted in [5] to study centrally pivoted plane-pad bearings so that simple analytical solutions can be obtained for the particular case of an *inclined* sliding bearing: (i) conduction heat transfer is assumed to be secondary to the convection heat transfer for the lubricant film; (ii) convection heat transfer occurs at the cross-film average velocity of the lubricant; and (iii) the shear strain rate in the lubricant is uniform, approximated by the bearing-average velocity-induced strain rate. Some of these assumptions are relaxed here.

The geometry of plane slider bearings can be described by three parameters: the bearing length (l) and the bearing inlet and exit heights (h_{in}, h_{out}) (Fig. 1). The bearing profile, which defines the way of height variation throughout the flow strip, may be described by few parameters [7] when simple geometries are considered (such as the inclined and the Rayleigh-step bearings) but in the general case involves many or even an infinity of parameters.

In hybrid bearings the lubricant film action comes both from the external pumping pressure and from the relative movement of two lubricated surfaces. In most cases the thin film hypothesis holds. Then, the Reynolds equation, which neglects inertial forces but concentrates on pressure forces and viscous shear, provides a convenient approach for steady-state flows of Newtonian incompressible lubricants [7,8]. As long as the film thickness is small (below 120 μm), comparison between solutions based on Reynolds

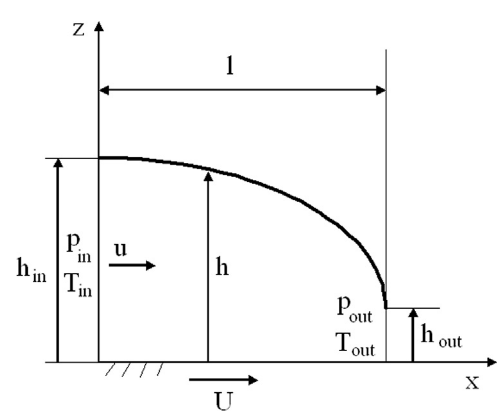

Fig. 1. The geometrical configuration considered here.