Feasibility of Gas-Expanded Lubricants for Increased Energy Efficiency in Tilting-Pad Journal Bearings

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Lubricants are necessary in tilting-pad journal bearings to ensure separation between solid surfaces and to dissipate heat. They are also responsible for much of the undesirable power losses that can occur through a bearing. Here, a novel method to reduce power losses in tilting-pad journal bearings is proposed in which the conventional lubricant is substituted by a binary mixture of synthetic lubricant and dissolved CO_2. These gas-expanded lubricants (GELs) would be delivered to a reinforced bearing housing capable of withstanding modest pressures less than 10 MPa. For bearings subject to loads that are both variable and predictable, GELs could be used to adjust lubricant properties in real time. High-pressure lubricants, mostly gases, have already been explored in tilting-pad journal bearings as a means to accommodate higher shaft speeds while reducing power losses and eliminating the potential for thermal degradation of the lubricant. These gas-lubricated bearings have intrinsic limitations in terms of bearing size and load capacity. The proposed system would combine the loading capabilities of conventional lubricated bearings with the efficiency of gas-lubricated bearings. The liquid or supercritical CO_2 serves as a low-viscosity and completely miscible additive to the lubricant that can be easily removed by purging the gas after releasing the pressure. In this way, the lubricant can be fully recycled, as in conventional systems, while controlling the lubricant properties dynamically by adding liquid or supercritical CO_2. Lubricant properties of interest, such as viscosity, can be easily tuned by controlling the pressure inside the bearing housing. Experimental measurements of viscosity for mixtures of polyalkylene glycol (PAG) + CO_2 at various compositions demonstrate that significant reductions in mixture viscosity can be achieved with relatively small additions of CO_2. The measured parameters are used in a thermoelastohydrodynamic model of tilting-pad journal bearing performance to evaluate the bearing response to GELs. Model estimates of power loss, eccentricity ratio, and pad temperature suggest that bearings would respond quite favorably over a range of speed and preload conditions. Calculated power loss reductions of 20% are observed when compared with both a reference petroleum lubricant and PAG without CO_2. Pad temperature is also maintained without significant increases in eccentricity ratio. Both power loss and pad temperature are directly correlated with PAG-CO_2 composition, suggesting that these mixtures could be used as "smart" lubricants responsive to system operating conditions.
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Keywords: tilting-pad journal bearings, energy efficiency, lubricant mixtures, tunable fluids

1 Introduction

Lubricants are used in tilting-pad journal bearings to support the weight of the shaft under loaded conditions, to provide suitable stiffness and damping when the bearing is incorporated into a rotor system, and to protect the mechanical components from wear [1–3]. Both the mechanical (e.g., viscosity and density) and thermal (e.g., heat capacity and conductivity) properties of the lubricant are important to ensure proper system function. High viscosity and/or density fluids typically provide more stability and protection for the bearing. A lubricant viscosity that is too high can result in undesirable power losses and unacceptably high pad temperatures [4]. The thermal properties of the lubricant must be selected so that fluid shearing in the bearing does not lead to temperature build-up and viscosity changes caused by the degradation of the lubricant. Selecting an application-specific lubricant generally solves this trade-off between bearing stability and bearing efficiency. The result can be a bearing that operates optimally in a narrow window of speeds and loads but is less robust if system conditions change [5].

Volatile oil prices and an increased focus on energy efficiency have driven research in more highly efficient lubricants [6]. Gas-lubricated bearings have been developed recently to accommodate the high shaft speeds needed for clean energy processes [7]. Under high speed and temperature conditions, conventional liquid lubricants can degrade and power losses through the bearing can become significant. Under pressure, a gas stream can be delivered to the bearing pad to provide a thin film separating the shaft and the pads. Advances in bearing geometry have helped improve the technical feasibility of such systems. Nevertheless, the lower load capacity and rotor damping of gas-lubricated bearings when compared with oil-lubricated systems will limit their use [8].

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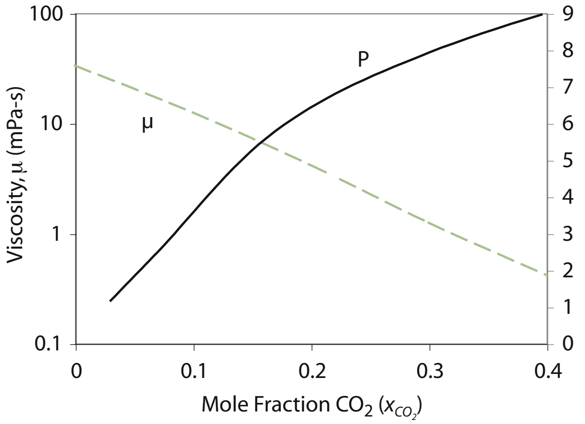

Fig. 1 The viscosity of a lubricant-gas mixture can be lowered by several orders of magnitude by controlling the composition using system pressure. These data, for PAG+CO_2 binary mixtures (at 35 °C), are from Refs. [14,16].

Another way to improve bearing efficiency is to deliver "tunable" lubricants, or those that can have their properties adjusted dynamically in response to changing loading or speed conditions [9]. This can be achieved by delivering binary mixtures of lubricants to the bearing and controlling the ratio of the mixture [10]. For example, one lubricant might have high viscosity and the other low viscosity and intermediate viscosities can be achieved by mixing the two. In such a system, the lubricant properties can be selected to produce conditions best suited for the application. Such systems would not be easily reversible since liquid-liquid separations are a challenge. Distillation, for example, is an energy intensive process that is not feasible on a small scale. For this reason, mixtures of liquid lubricants have not been widely adopted for dynamic control of lubricant properties.

In the chemicals sector, smart solvents are being designed using gas-expanded liquids (GXLs). A GXL is a binary mixture of a solvent, usually organic, and an industrial gas, most often CO_2. The medium is liquid but the composition can be controlled so that it has properties between those of a pure solvent and a gas. The mixture is maintained at intermediate pressures, around 1 MPa, when compared with higher pressure supercritical fluids, which are maintained around 10 MPa [11]. The presence of gas in the mixture enhances mass transfer of dissolved solutes relative to the pure solvents. GXLs also consume less solvent, which often has significant environmental or occupational health burdens, and replace it with dense liquid CO_2, a largely inert byproduct of numerous industrial processes. When the presence of gas is no longer desirable, it can be separated easily by dropping the pressure of the mixture and venting the gas. Though GXLs are being actively investigated, this research has been focused on chemical applications and little effort has been made to explore the analogous concept in lubricant systems.

The properties of lubricant/CO_2 mixtures under pressure have been studied in refrigeration applications [12]. CO_2 can be used as an alternative to chlorinated organic refrigerants that are being phased out for environmental reasons [13]. The phase behavior, tribology, and heat transfer characteristics of CO_2 and synthetic lubricants at high pressure and low temperature have been studied to determine the most appropriate operating parameters for this application [14]. At low gas concentrations (<40%), the viscosity of these mixtures can decrease by several orders of magnitude, as shown in Fig. 1. Since the pressure of CO_2 drives the composition of the mixture, it is possible to control viscosity directly by controlling pressure. The effect of this relationship on surface wear has been reported [15]. Nevertheless, the important implications of viscosity control using dissolved gases for power loss reductions in typical industrial bearings have not yet been investigated.

Based on the work in GXLs and CO_2-based refrigerants, it seems possible that tunable mixtures comprised of synthetic lubricants and pressurized carbon dioxide could be developed. These gas-expanded lubricants (GELs) could be used to control viscosity in real time. A controller measuring shaft speed or applied load could be coupled to the system allowing for adjustments to lubricant viscosity in response to rotor conditions. GELs have the potential to decouple viscosity and efficiency in tilting-pad journal bearings, without sacrificing performance capabilities in the bearing. A schematic of the proposed technology is presented in Fig. 2. Because bearings are ubiquitous in industrial settings, a method to improve their efficiency could represent significant energy savings on a large scale. In this work, the viscosity of binary polyalkylene glycol (PAG)/CO_2 mixtures was measured for a bearing-relevant set of conditions using a rheometer with a high-pressure cell. These results were input to a thermoelastohydrodynamic (TEHD) analysis code developed by this research group. The code solves the temperature and bearing dynamic coefficients simultaneously and has been previously shown to effectively predict the performance of lubricants in tilting-pad bearing systems [17].

2 Experimental Methods

The viscosity of the lubricants and lubricant-CO_2 mixtures was measured using an Anton-Paar MCR 301 rheometer equipped with a high-pressure measuring cell rated up to 15 MPa. The CO_2 was delivered using a Teledyne ISCO 500HP syringe pump with a constant temperature jacket. The temperature jacket was needed to ensure that liquid CO_2 was being delivered to the pressure cell such that an equation of state estimate of CO_2 temperature, pressure, and volume could be applied to calculate the mixture composition [18]. Temperature was controlled to within $\pm 0.1\,^{\circ}C$ using a Peltier style temperature controller integrated into the rheometer. The lubricant was added to the pressure cell volumetrically. Mea-

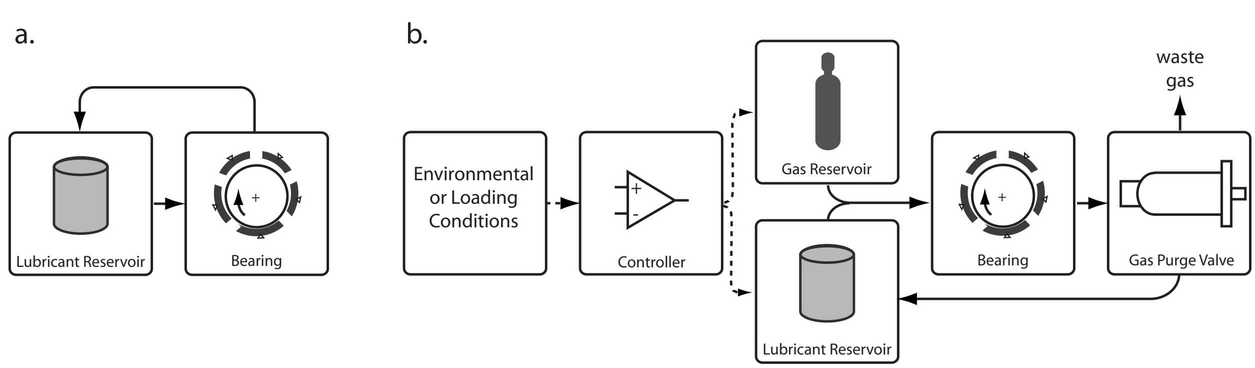

Fig. 2 Schematic of lubricated bearing systems under a conventional static setup (*a*) and in the presence of a GEL (*b*) where pressurized gas/lubricant mixtures can be delivered in response to variable operating conditions

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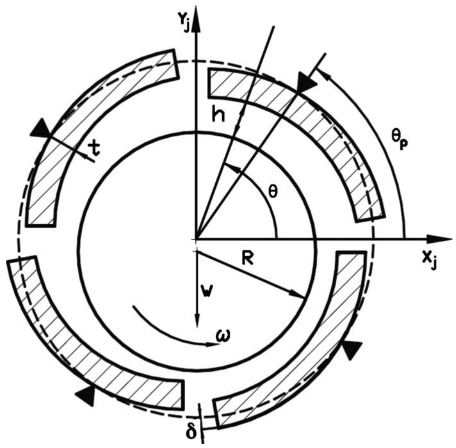

Fig. 3 Schematic of tilting-pad journal bearing with process parameters used in the modeling framework reported here

surements of viscosity were made over a range of shear rates to ensure that the mixtures exhibit Newtonian or near-Newtonian behavior. The range was selected to be $1-1000\ \mathrm{s}^{-1}$ based on early trials and on the range of shear rates that would be expected in a tilting-pad journal bearing environment.

The high-pressure behavior of conventional lubricant mixtures has been investigated in the past and empirical relationships proposed to estimate mixture properties [10]. In particular, the viscosity of mixtures has been approximated using Eq. (1)

$$\ln \mu_m = x_1 \ln \mu_1 + x_2 \ln \mu_2 \tag{1}$$

where μ_m is the viscosity of the lubricant mixture and x and μ are, respectively, the mass fraction and viscosity of the components under the same temperature and pressure. This convenient relationship relies on easily measured properties that are usually readily obtained. For a diverse range of lubricant molecular structure, Eq. (1) has been shown to be an effective guide to mixture properties though the relationship has not yet been tested for mixtures of lubricants with compressed gas. A goal of the experimental portion of this work was to determine whether lubricant/CO_2 mixtures could be understood using a similar relationship.

3 Modeling Framework

A modeling approach developed by He [17] for tilting-pad journal bearings was used to evaluate the performance of GELs. The TEHD framework simultaneously predicts a number of key bearing performance measures including journal operating position, altitude angle, power loss, maximum temperature, and the bearing dynamic coefficients. Previous models did not effectively capture the lubricant or journal maximum temperature, and the resulting thermal expansions, and so were ineffective at predicting bearing operating conditions for real lubricants over a range of operating conditions [19]. The TEHD model solves the coupled pressure, temperature, and elasticity problem simultaneously using a series of iterations [20]. The hydrodynamic pressure is calculated from the generalized Reynolds equation, and a two-dimensional energy equation is derived to calculate the temperature distribution. The pad mechanical and thermal deformations are then calculated using a two-dimensional finite element numerical method. The model also accounts for lubricant turbulence effects and pivot flexibility. A schematic of the tilting-pad journal bearing along with important system variables is presented in Fig. 3.

3.1 Thermoelastohydrodynamic Modeling. The TEHD model is based on the generalized 2D form of Reynolds equation (Eq. (2)). The Reynolds equation forms the cornerstone of hydrodynamic analysis and was originally derived from the Navier–Stokes equation and the continuity equation assuming a thin film [21]. Several key assumptions are made to arrive at the Reynolds equation, namely that the pressure gradient across the film thickness is zero, that no slip conditions hold on the surface, and that the fluid exhibits constant density. The equation allows for the solution of the pressure profile in this thin film as a function of coordinates, film thickness, surface speed, and most importantly for this work, lubricant viscosity. The generalized form of the Reynolds equations (Eqs. (2) and (3)) allows for the variation of the viscosity across the film as well as for the presence of turbulence.

$$\frac{\partial}{\partial x}\left\{h^3\Gamma(x,z)\frac{\partial P}{\partial x}\right\} + \frac{\partial}{\partial z}\left\{h^3\Gamma(x,z)\frac{\partial P}{\partial z}\right\} = -UG(x,z)\frac{\partial h}{\partial x} \tag{2}$$

where

$$\begin{aligned}
\Gamma(x,z) &= \int_0^1 \left[\zeta_2(x,y,z) - \frac{\zeta_2(x,1,z)}{\zeta_1(x,1,z)}\zeta_1(x,y,z)\right]dy \\
G(x,z) &= \frac{1}{\zeta_1(x,1,z)}\int_0^1 [\zeta_1(x,y,z)]dy \\
\zeta_1(x,y,z) &= \int_0^y \frac{1}{\mu_e(x,y',z)}dy' \\
\zeta_2(x,y,z) &= \int_0^y \frac{y'}{\mu_e(x,y',z)}dy'
\end{aligned} \tag{3}$$

The lubricant effective viscosity (μ_e) appears in several terms of the Reynolds equation. Effective viscosity is used to account for the effects of increased stresses in the lubricant under turbulent conditions. These stresses in the lubricant are modeled using the near wall eddy viscosity model that has been applied widely in similar models of bearing dynamics. A local Reynolds number is used to estimate the flow regime around the bearing. Values below and above critical Reynolds numbers, 500 and 1000, respectively, are considered laminar and turbulent. For the transitional region between these values, a scaling factor (β) is used to modify the eddy viscosity. The effect of turbulence stresses on the lubricant, caused by speed, is combined with the eddy viscosity law (ε_m) to express the viscosity as an effective viscosity μ_e (Eq. (4)).

$$\mu_e(x,y,z) = \mu\left(1 + \beta\frac{\varepsilon_m}{\nu}\right) \tag{4}$$

Power loss in tilting-pad journal bearings results from shearing in the lubricant film. This shear stress is a function of the effective lubricant viscosity and the velocity gradient. As defined in Eq. (4), the effective viscosity of the lubricant incorporates the dynamic viscosity and the turbulent viscosity. The velocity gradient is obtained from the pressure solution of the generalized Reynolds equation. A viscous lubricant exerts a resisting torque on the torque that drives the bearing. The power loss describes the energy required to overcome the resisting torque. Petroff's equation for concentric cylinders is used here for a good approximation of this effect [3,22]. The TEHD model takes the journal eccentricity into account by dropping the concentric assumption of Petroff's model. Eccentricity is included in the estimate for power loss as a function of the journal speed and shear stresses (Eq. (5)).

$$P_{\mathrm{loss}} = \omega R\int_0^L\int_0^{2\pi}\tau(\theta,z)d\theta dz = U\int_0^L\int_0^{2\pi}\tau(\theta,z)d\theta dz \tag{5}$$

The temperature profile in the bearing can be solved using the 3D form of the generalized energy equation. In an effort to reduce the computational intensity of the 3D version of this model, previous work has demonstrated that the temperature profile in the axial direction is generally a constant polynomial function based on

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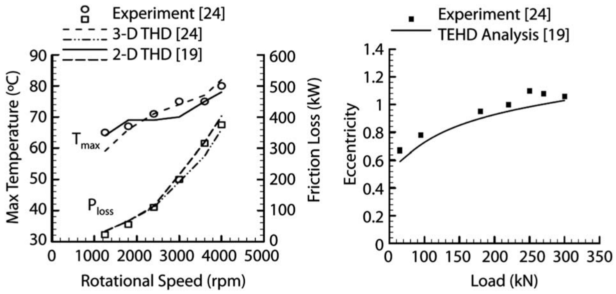

Fig. 4 Published modeling and experimental results demonstrate the effectiveness of the model in predicting journal temperature (left) and eccentricity (right) data published in Ref. [24]

known boundary conditions [23]. To simplify the energy equation, the temperature in the axial direction is integrated and the resulting 2D equation is solved for the radial and circumferential directions (Eq. (6)).

$$\rho C_P\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k_e\frac{\partial T}{\partial y}\right)+\frac{\partial T}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\mu_e\left(\left(\frac{\partial u}{\partial y}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2\right) \quad (6)$$

The response of the bearing to dynamic loading, speed, or lubricant conditions is modeled as a function of a set of linearized stiffness and damping coefficients at the location of the bearing on the shaft. The forces acting on the shaft for small amplitude motion are

$$\begin{bmatrix} F_x \\ F_y \end{bmatrix}=\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}\begin{bmatrix} x_j \\ y_j \end{bmatrix}+\begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}\begin{bmatrix} \dot{x}_j \\ \dot{y}_j \end{bmatrix} \quad (7)$$

The reduced set of coefficients is composed of the principal direction (xx and yy) and the cross coupling direction (xy and yx). These speed dependent dynamic coefficients are calculated by expressing the film thickness and pressure as a linear combination of a steady state component and the perturbed component. The Reynolds equation is solved for the perturbed pressure, and the coefficients are calculated through the integration of the perturbed pressure.

3.2 Model Validation. The TEHD model has been validated and shown to correlate effectively with experimental results for a number of journal bearing configurations. The effectiveness of the TEHD model was first shown in Ref. [17]. In Fig. 4, the model results are compared with experimental results from Ref. [24]. The TEHD model approximations for journal temperature correlate closely with the experimental and modeling results of Taniguchi et al. The eccentricity estimates of the model also effectively capture the behavior of the bearing.

Table 1 Bearing parameters used in this work

Parameter	Value	Unit
Diameter	0.1	m
Pad thickness	0.02	m
Length	0.07	m
Clearance	7.9×10^{-5}	m
Preload	0.47, 0.15	
Offset	0.5	
Configuration	LBP	
No. of pads	4	
Oil supply temperature	40	°C
Oil supply rate	0.63	l/s
Radial load	10,000	N
Pad thermal conductivity	50	W/m K
Convection coefficient	736	W/m^2 °C

4 Modeling Parameters

The bearing size and geometry employed in this paper were selected to be representative of a large number of tilting-pad bearings based on the work of Fillon et al. [25]. The bearing is assumed to be fully flooded and hot oil carry over from pad to pad is 100%. The conditions are listed in Table 1.

5 Lubricants Modeled

Polyalkylene glycol (PAG) was selected for this work because it is a widely used class of synthetic lubricants and because the phase behavior and viscosity of high-pressure CO_2/PAG mixtures are well understood [14]. Petroleum-based lubricants are not suitable for gas-expanded lubricant applications because the heterogeneous chemical composition of petroleum-based lubricants and variations between blends makes their high-pressure phase behavior in CO_2 highly variable [26]. In addition, undesirable side effects are common such as the extraction of low molecular weight components. Synthetic lubricants, such as PAG, are well suited for mixing in CO_2 since relevant properties (such as molecular weight, polarity, branching, etc.) can be specified a priori and are available commercially [27]. For this work, a PAG blend was selected with viscosity close to that for a reference fluid, ISO 32 (Table 2).

6 Experimental Results

Measurements of GEL viscosity showed that CO_2 can be effectively used to reduce the viscosity of the PAG lubricant tested here. The results, presented in Fig. 5, reflect viscosity reductions of nearly 65% over the range of compositions tested here at 40°C. At 100°C the effect is equally pronounced. The viscosity values used for this modeling effort are also presented in Fig. 5. These values were selected by incorporating published values [13] with the experimental values collected here. PAG is a commercial lubricant and variations between suppliers and batches are common.

Table 2 Lubricant properties modeled in this work. Values were obtained experimentally and from Refs. [14,28].

Lubricant property	PAG[a]	95% PAG+5% CO_2	85% PAG+15% CO_2	Reference fluid[b]	Units
Density	970	891	732	861	kg/m^3
Dynamic viscosity at 40°C	17	15.3	11.9	28	mPa s
Dynamic viscosity at 100°C	4	3.6	2.8	4.8	mPa s
Specific heat capacity	2075	2089	2117	1950	J/kg K
Thermal conductivity	0.159	0.152	0.138	0.149	W/m K

[a]PAG used for these trials is UCON 50-HB-100 water soluble, 50% by weight oxyethylene and oxypropylene, manufactured by Dow Chemical (Midland, MI).
[b]Reference fluid is ISO 32, a generic mineral-based lubricant used as a benchmark in tilting-pad bearing systems, obtained from McMaster-Carr (Robbinsville, NJ).

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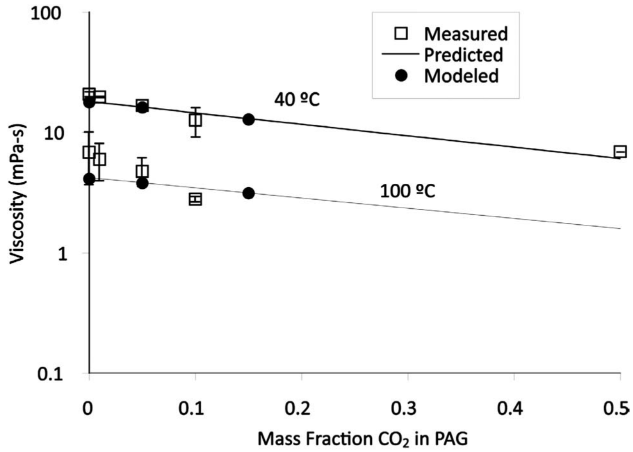

Fig. 5 Viscosity of PAG-CO_2 mixtures as a function of mixture composition at 40°C and 100°C. The measured results were obtained experimentally while the predicted value is obtained using Eq. (1). The modeled values were input to the TEHD model of bearing performance.

To evaluate the effect of variable shear rates on these binary mixtures, lubricant viscosity was measured over a range from 1 s^{-1} to 1000 s^{-1}. The results, not shown here because the lubricant behavior was Newtonian, suggest that GELs will not experience significant phase separation over the range of shear values typically encountered in a tilting-pad journal bearing. As a final benchmark of the viscosity measurements, the viscosity of ISO 32 was also measured. The measured viscosity of 24.9 mPa s ±1.4 mPa s is consistent with manufacturer supplied values of 28 mPa s at 40°C.

In Fig. 5, viscosity measurements are reported up to a maximum CO_2 concentration of 15%. Viscosity measurements at higher mass fractions of CO_2 are not possible at higher temperatures because the mixture separates into liquid-liquid-vapor or liquid-vapor combinations. It is unlikely that GELs would be delivered under these conditions since the heterogeneous mixtures could negatively impact bearing performance. The pressure required to achieve a single-phase mixture over a range of CO_2 mole fractions and temperatures is presented in Fig. 6. The experimental data, adapted from published work, demonstrate how higher pressures would be required at higher temperatures to maintain a single-phase lubricant mixture. In practice, this should

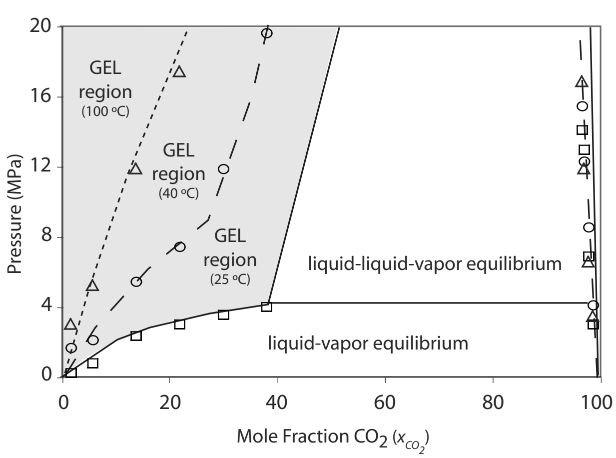

Fig. 6 Phase behavior of PAG-CO_2 mixtures as a function of mixture composition with GEL regions highlighted for several relevant temperatures. Experimental results are adapted from Ref. [14].

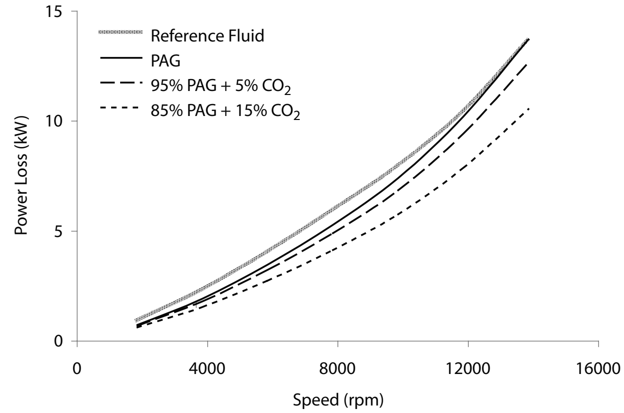

Fig. 7 Power loss as a function of speed. Efficiency improvements of nearly 20% are observed over a range of operating speeds when using GELs compared with a reference lubricant.

not present any limitations since at high temperature the viscosity of pure lubricants is already low. These data do suggest that bearing operating conditions should be well characterized so that operating pressures can be selected that achieve the desired properties in the GEL.

7 Modeling Results

The modeling results suggest that the use of GELs could provide a power loss reduction greater than 20% over a wide range of speed up to 14,000 rpm (Fig. 7). This result can be best understood in the context of the power loss equation (Eq. (5)). Power loss is a function of shaft rotational velocity (i.e., speed) and shear forces in the lubricant. Shaft rotational velocity is usually specified by the application and cannot generally be modified to reduce power losses. Shear force in the lubricant, on the other hand, can be controlled independently of the device in which the bearing is being used since it is proportional to lubricant viscosity. Shear force is the product of effective viscosity and the velocity gradient in the lubricant film. Lower viscosities will result in lower power loss. In effect, by dialing down viscosity in a GEL, it is possible to directly lower the power loss in the bearing, without substantially changing the function of the mechanical system in which the bearing is used.

The three lubricant/CO_2 conditions modeled here, PAG with 0%, 5%, and 15% CO_2, were selected to cover the range of easily achieved mixtures. 100% PAG (0% CO_2) was included to show that the performance advantages come from the mixture of CO_2 and lubricant and not just from the use of PAG relative to the ISO 32 reference fluid. The power loss values for the reference fluid and the 100% PAG were very comparable. CO_2 can be added to PAG gradually up to a composition of ~40% after which the pressure required to stabilize the mixture is unreasonably high for most industrial situations. In addition, the viscosity and other properties of such mixtures would be comparable to conventionally available gas-lubricated bearings. In practice, GEL delivery to the bearing would not be limited to the discrete mixture compositions reported here (e.g., x_{CO_2}=5% or 15%). Instead, the CO_2 delivered to the bearing over a continuous range of compositions would depend on the specific lubricant and the pressure rating of the seals in the bearing. Also, in practice, a GEL could be used to produce an independent power loss curve where power loss increases more gradually as a function of speed than the results reported in Fig. 7. In such a scenario, CO_2 would be added to the lubricant gradually as speed is increased to reduce the associated power loss.

A preliminary energy balance on a bearing using GELs suggests that the energy needed to compress the CO_2 up to the required pressure is much lower than the energy savings provided by using

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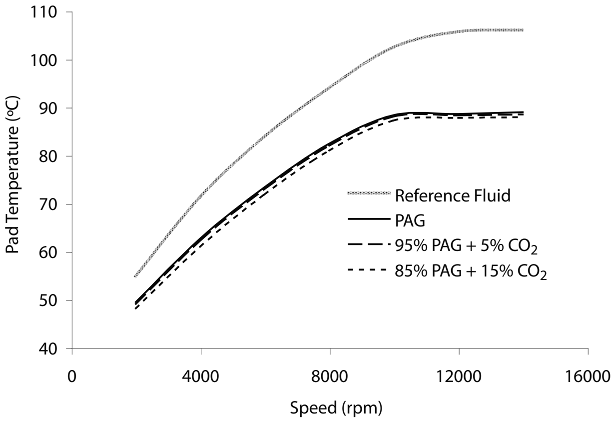

Fig. 8 Pad temperature as a function of speed. A desirable reduction in pad temperature can be achieved using GELs due largely not only to the superior heat removal characteristics of the PAG lubricant but also to the reduction in power loss provided by GELs.

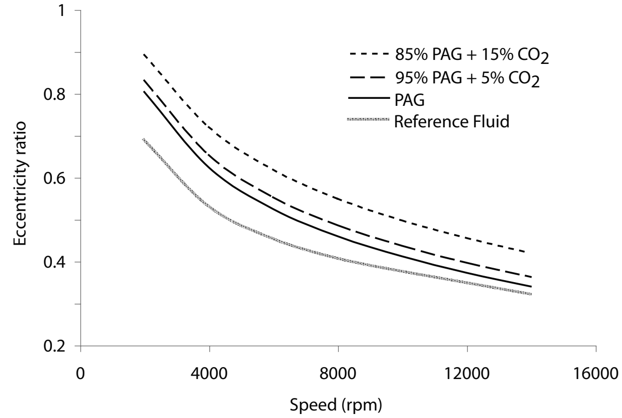

Fig. 9 Journal eccentricity ratio as a function of speed. The presence of CO_2 in the GELs does increase the eccentricity ratio as expected but not outside the normal operating range for this type of bearing.

the GEL. The energy requirement for CO_2 compression is on the order of 1 W assuming that the CO_2 is compressed isentropically from 4 MPa, roughly the pressure of a commercially available canister of CO_2, up to 9 MPa at 25 °C, the highest pressure we would expect to need if mixing PAG and CO_2 (Fig. 1). This power consumption estimate is several orders of magnitude lower than the 1–3 kW of power savings that are possible by using GELs. For this estimate the flow rate of lubricant/CO_2 mixture into the bearing is assumed to be 0.003 m^3/min and 15% CO_2. These flow rates are much lower than those required by gas-lubricated bearings where the pads must be continuously supplied with a steady stream of compressed gas.

Consistent with the predicted power loss reductions, pad temperatures were estimated to be approximately 15% lower when using a PAG or GEL mixture compared with the reference fluid (Fig. 8). The temperature profile in the lubricant is an important characteristic in practice since excessively high temperatures can lead to thermal degradation of the lubricant or damage to the pad material. The temperature reduction observed here seems to be driven by the fact that PAG and the PAG+CO_2 mixtures have a higher heat capacity and thermal conductivity than the reference fluid, ISO 32 (Table 2). This is not particularly surprising since PAGs are designed for use in heat transfer as well as lubrication applications. But the lower temperatures may also be caused in part by the lower viscosity of the PAG+CO_2 mixtures. The lower shear that results from the addition of CO_2, responsible for reductions in power loss, is also likely to contribute to lower residual heat in the bearing. The fact that the pad temperature is seemingly independent of CO_2 concentration is encouraging since CO_2 has a lower molecular density and thermal conductivity than most lubricants. It would be reasonable to expect that the heat capacity of a lubricant would be negatively affected when introducing dissolved gas. The model results suggest that the unfavorable thermal properties of CO_2 are not significant for the bearing geometry under the conditions tested here.

The eccentricity ratio results show that GEL mixtures containing up to 15% CO_2 effectively support the load of the bearing without a major increase in shaft eccentricity (Fig. 9). These results were encouraging in light of the computational method used to estimate the load capacity in the TEHD model. The numerical method used to solve the model does not compute load directly but rather solves for the pressure and iteratively determines lubricant load capacity. When the model does not converge, it indicates either that the lubricant is incapable of supporting the applied load or that the modeled conditions cannot be adequately captured using the numerical methods employed in the model. The model successfully converged under each of the GEL conditions modeled here suggesting that the mixtures are capable of handling the applied load. In addition, when the eccentricity ratio is high, the thin lubricant film can produce heat build-up that is unfavorable. This excess heat accumulation was not observed under any of the conditions tested. Relative to the ISO 32 reference fluid the eccentricity of the PAG mixtures is somewhat higher but still within the normal operating range for tilting-pad journal bearings. Particularly at higher speeds, the eccentricity difference between the fluids was not significant. These eccentricity ratio data are encouraging and indicate that GELs should not interfere with the normal bearing function.

A sensitivity analysis was conducted on the model to determine the effect of bearing preload on lubricant function. The results of this analysis, shown in Fig. 10 for power loss, indicate that the effect of GELs is relatively independent of preload. Preloads of 0.47 (original test conditions) and 0.15 were modeled. In both cases the model converged successfully and the percent power loss improvement remained nearly constant. In fact, the results demonstrate that the use of GELs could reduce power loss more significantly than a drop in preload. The benefits of GELs on pad temperature were similarly independent of preloading conditions. Similar reductions in power loss could be achieved, for these preload conditions, using a shorter bearing or less viscous lubricant, but that is not the principle benefit of using GELs. The advantage

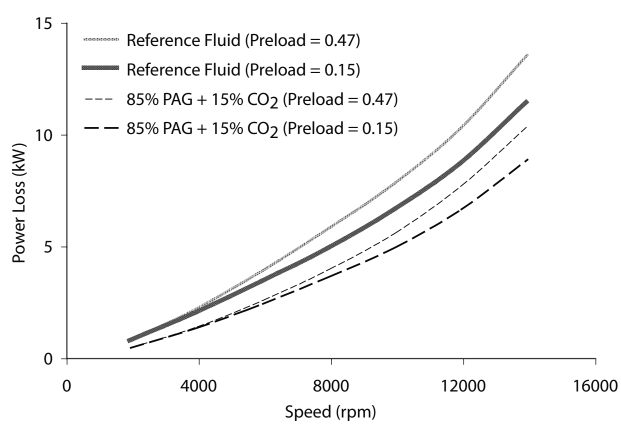

Fig. 10 Bearing power loss under two different preloading conditions suggests that the effect of GELs is independent of bearing geometry/preload

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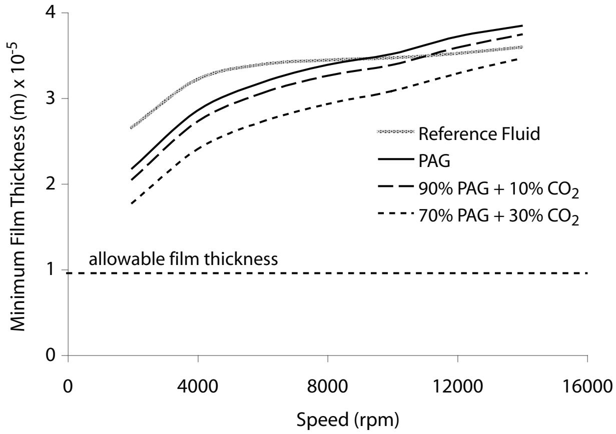

Fig. 11 Minimum film thickness as a function of speed for GELs and a reference fluid as compared with the allowable film thickness as estimated by Martin and Garner [29]

of GELs is that they provide a dynamic means by which to change the performance of a bearing during operation without changes in the equipment.

Since contact between a bearing and a shaft can result in damage to both, a minimum film thickness is specified as a bearing design parameter that can predict this undesirable contact. Here the minimum film thickness was estimated using the bearing clearance and compared with the recommended film thickness specified by Martin and Garner [29]. The results, shown in Fig. 11, demonstrate that under low speed conditions, the minimum film thickness is a factor of 2 higher than the allowable film thickness. At higher speeds the margin is even higher suggesting that the use of GELs will not impede the function of the bearing. The values presented in Fig. 11 are for the pad with the minimum film thickness, though the other three pads had a similar profile. Lubricant peak pressure, which is highly correlated with minimum film thickness, followed a similar trend [30].

As a final measure of bearing performance in the presence of GELs, the stiffness and damping coefficients of the bearing were calculated. The results shown in Fig. 12 suggest that GELs could impart beneficial stiffness behavior to a bearing when incorporated into a rotor system. Since the bearing was modeled with four pads, the stiffness coefficients are symmetric such that K_{xx} and K_{yy} are identical. The effects of temperature and thermal deformation were included in the model but were found to have a negligible effect on dynamic coefficients under the conditions modeled here. The values shown in Fig. 12 are for K_{yy} under high and low preload levels. At high speed, the results show a reduction in the stiffness coefficient when CO_2 is introduced into the PAG mixture. The reduction suggests a decreasing bearing stiffness at the high speed. The soft bearing will allow a shaft motion inside the bearing and hence an increase in the effective damping of the system. The increase in the effective damping of the system is an advantage for systems that operate at high speeds particularly in light of the stiffness coefficient for the reference fluid, which increases with speed. The bearing damping coefficients (C_{xx} and C_{xy}) showed an almost negligible change between the reference fluid, pure PAG, and the mixture of PAG/CO_2. This combination of a flexible stiffness coefficient and an unchanged damping coefficient suggests that the presence of CO_2 in GELs is unlikely to compromise the stability of a rotor system and that it will likely be beneficial for bearings, particularly for those that operate at higher speeds.

8 Conclusions

This work proposes the use of lubricant/gas mixtures that can be tuned in response to changing speed or loading conditions to improve the energy efficiency of tilting-pad journal bearings. These GELs could provide the foundation for a new type of smart lubricant with viscosity that is adjustable in real time. Using the chemical parameters of lubricant/gas mixtures measured by the authors and obtained from literature, a thermoelastohydrodynamic model of a tilting-pad journal bearing was used to estimate the bearing characteristics when using these GELs. The results show a compelling 20% drop in power loss in the presence of these mixtures compared with conventional lubricants. This efficiency improvement was observed over a wide range of operating speeds up to 14,000 rpm. Estimates of the pad temperature showed an important reduction when using GELs, which could help prevent the degradation of the lubricant or damage to the pads. This temperature reduction was attributed to the superior heat transfer properties of the GEL mixture and to the decrease in frictional losses. Journal eccentricity increased slightly when using GELs but it was still within the normal range for this type of bearing and under high speeds it did not constitute a problematic increase from conventional lubricants. The bearing stiffness and damping coefficients indicate that the presence of CO_2 in the lubricant mixture can actually soften the stiffness of the bearing at higher speeds, making the bearing less susceptible to perturbations when incorporated into a rotor system. Taken together, these results suggest a promising application of lubricant-gas mixtures. Such smart fluids could greatly reduce the power losses typically associated with bearing systems.

Additional research is necessary to explore the application of this technology in the laboratory or in other bearing systems. Al-

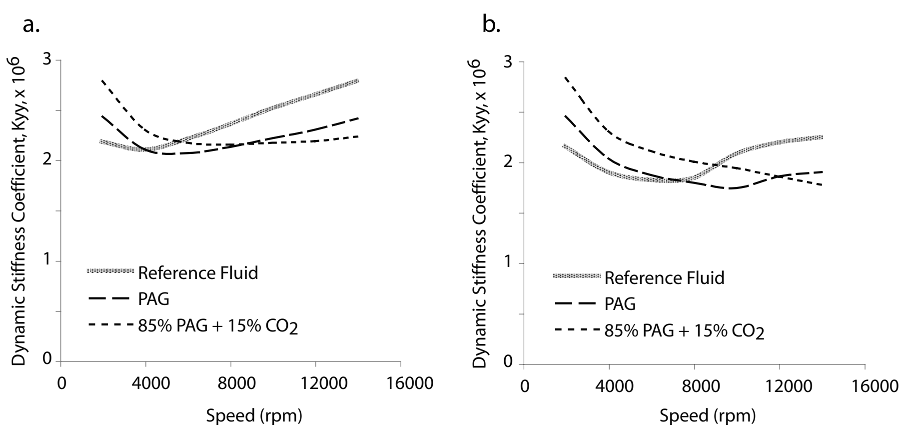

Fig. 12 Bearing stiffness coefficients for the reference fluid, PAG, and PAG+CO_2 under preloads of 0.47 (left) and 0.15 (right)

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though the modeling results presented here are known to correlate well with experimental observations, certain factors might not be captured by the existing model. For example, it is possible that during depressurization, the change in phase behavior of GELs could result in phase separation and heterogeneous forcing on the bearing. The existing model also does not suggest how well GELs will function under the elastohydrodynamic conditions observed in other devices such as gears. One application where this technology could have a positive impact would be wind turbines, which are subject to variable loading and high power losses through bearings. To explore these possibilities future research will investigate the power loss and lubrication performance of GELs in experimental bearings to fully evaluate their impact on process performance.

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Nomenclature

$C_{xx}, C_{yx}, C_{xy}, C_{yy}$ = damping coefficients
C_p = lubricant specific heat
D = bearing diameter
h = film thickness
k = lubricant heat conductivity
k_e = effective heat conductivity that includes turbulence
$K_{xx}, K_{yx}, K_{xy}, K_{yy}$ = stiffness coefficients
P = pressure
P_{loss} = friction power loss
R = journal radius
t = pad thickness
T = temperature
T_j = journal temperature
U = journal surface velocity
u = lubricant velocity in the circumferential direction
v = lubricant velocity in the radial direction
w = lubricant velocity in the axial direction
W = load
x = circumferential coordinate ($=R\theta$)
x_j = horizontal journal coordinate
y = radial coordinate
y_j = vertical journal coordinate
z = axial coordinate
δ = pad tilt angle
δ_{elast} = elastic deformation
δ_{therm} = thermal deformation
ε_m = eddy viscosity
μ = lubricant dynamic viscosity
μ_e = effective viscosity that includes turbulence
ν = lubricant kinematical viscosity
θ = circumferential angular coordinate
θ_p = pivot angular location
ρ = lubricant density
τ = shear stress
ω = shaft rotational angular velocity

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