Benefits and risks of large hydrostatic recesses in hydroelectric turbine thrust bearings

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ABSTRACT

Operators of hydroelectric power stations sometimes call upon engineers to modify existing hydroelectric turbines, usually several decades old, for improved maintainability and reliability. One common modification is the hybridisation of plain thrust pads to allow hydrostatic operation to reduce the risk of bearing wipe at low speed (virtually all new installations benefit from this feature). A modification such as this is not a difficult undertaking; however, there are numerous factors that need to be considered in order to maximize bearing performance. One factor that stands out above the others is whether the thrust bearing should be designed to lift the turbine immediately from the standing condition, which presents an interesting challenge: the recess has to have a sufficiently large area in order for the supply pressure to be able to overcome the dead weight of the turbine. If the combination of groove area and pressure is insufficient, then lifting is neither immediate nor guaranteed. This need not be a significant problem, as the bearings have exhibited adequate performance even in the absence of a hydrostatic lubricant supply. A case study is presented whereby relatively large hydrostatic recesses are added to the pads of thrust bearing. It is demonstrated with the aid of simple numerical modelling that the impact of the recess relative to the original pad is small under normal operating conditions. Most surprising, however, is that significant reductions in average oil film temperature and power dissipation are predicted.

Keywords: hydrostatic, thrust, bearing, hydroelectric, turbine

1. INTRODUCTION & BACKGROUND

Hydroelectricity constitutes approximately 16% of the world's electricity consumption and 3.4 percent of total energy consumption as of 2010 [1]. It accounts for about 85% of electricity from renewable resources [2]. Hydroelectricity production increased at a rate of 3.5% annually from 2003 to 2010 [1] but slowed to 1.6% in 2011 [3]. In Australia, there are more than 100 hydroelectric power stations with the capacity to produce 8,200 megawatts of electricity [4]. Some nations are able to produce 100% of electricity needs while numerous are able to provide more than 90%; Iceland, Norway and New Zealand produce the most hydroelectricity per capita [5]. As the world moves towards carbon emission reduction, pumped-storage hydroelectricity (PSH) provides opportunity for energy storage and load balancing from abundant but inconsistent energy sources such as wind and solar [6]. It is claimed that PSH accounts for more than 99% of bulk storage capacity worldwide [6] with claims of up to 87% conversion efficiency [7]. It follows that many aspects of the conversion process contribute to overall efficiency, including bearing design.

In this work, the efficiency of hydroelectric thrust-bearing pads is considered. Despite the remarkable efficiency of bearings operating in the hydrodynamic lubrication regime when compared to other regimes, the great magnitude of thrust borne by the bearing surfaces means that considerable energy loss is inevitable – enough for each bearing to power many electric kettles, for example! The question arises: Can bearing design be modified in any way to further increase efficiency?

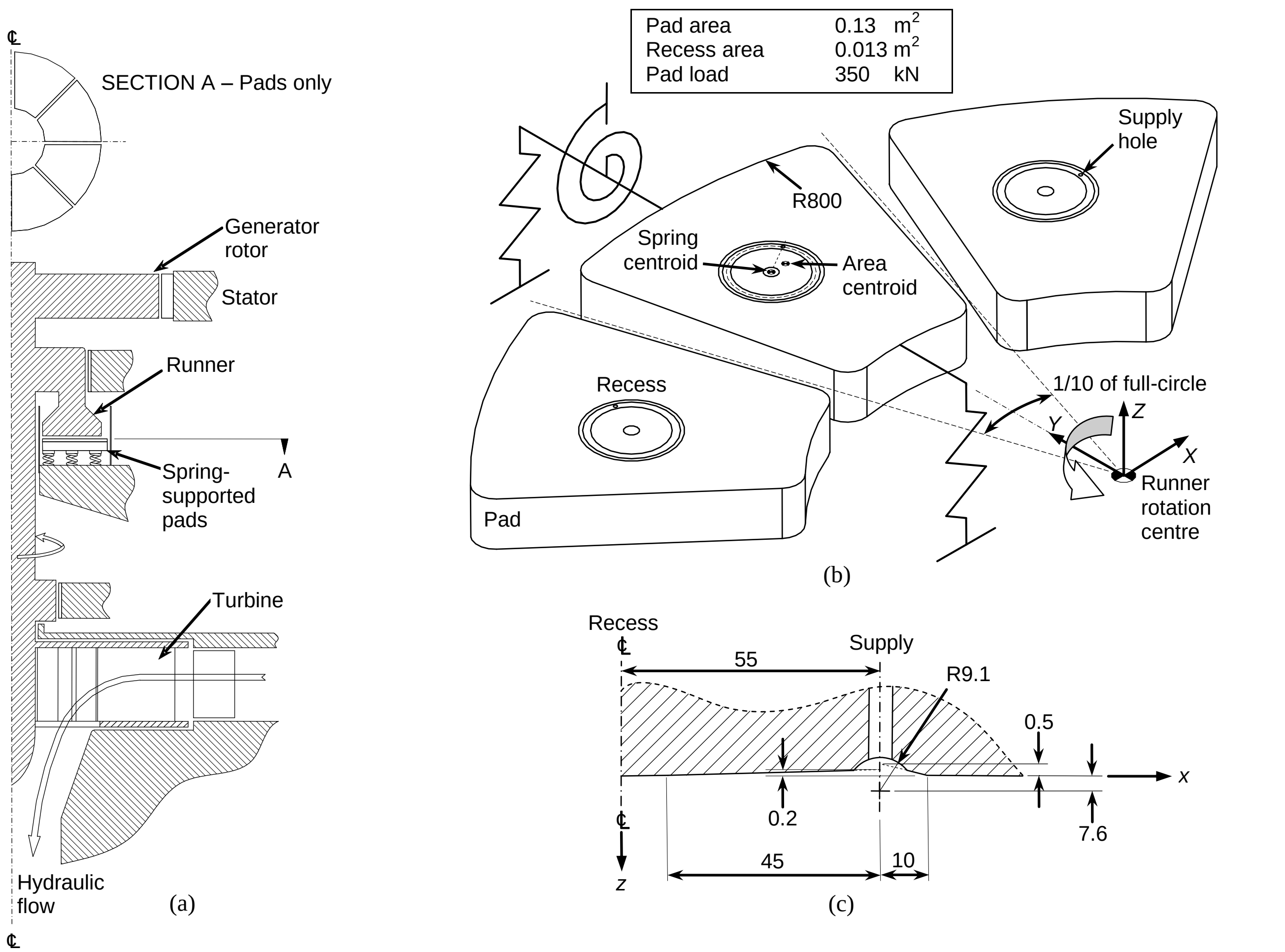


Figure 1. (a) Schematic arrangement of hydroelectric turbine. (b) Typical pad characteristics showing idealised suspension of the pad and the hydrostatic groove added (3 pads of 10 shown). (c) Recess section and dimensions (in mm).

The author's contact with the hydroelectricity industry arises from the evaluation of the cost-effectiveness of extending the life of existing plant compared to replacement with completely new installations, which are not only more costly, but also potentially disruptive to production. Older facilities often consist of bearings with plain pads (figure 1), while newer ones incorporate some form of hydrostatic assistance (external pressurisation) to prevent contact between the pad and the runner surfaces, especially at low speeds when the hydrodynamic film thicknesses may be too thin to separate the sliding surfaces completely. This is especially a problem during shutdown, when thermal crowning increases the likelihood that the central portion of the pad will be worn away preferentially, giving a concave profile when the pad temperature returns to uniformity. This presents potentially serious consequences as simulations show that even minor concavity can impede the spontaneous formation of a hydrodynamic film.

The hybridisation of a thrust bearing with hydrostatic assistance therefore becomes an attractive proposition. When speeds are small during shutdown and startup, a hydraulic pump injects oil into a cavity machined into the surface of the hydrodynamic pad. The pressure of the oil, acting over the projected area of the recess acts to overcome the dead weight of the turbine and rotor assembly. The separation of the surfaces results in low-friction startup, which also practically eliminates wear. As the turbine speed increases due to the hydraulic flow, the oil film no longer requires hydrostatic pressurisation to maintain the lubricating oil film. Upon switching off the hydrostatic pump, a check-valve prevents leakage of the hydrodynamically-pressurised oil back towards the hydrostatic supply manifold.

How big the hydrostatic recess should be made is an interesting problem and there are different approaches to its solution. The first and probably most common involves making the recess as small as possible so that its presence influences minimally the hydrodynamic operation of the pad when compared to an un-recessed pad; this represents the more computationally convenient approach. The alternative is to use a large recess which has greater impact on the performance of the pad, but can be designed as a deliberate and important feature in the operation of the bearing. The first option appears the more attractive but it ignores the fact that recess size determines whether hydrostatic pressure alone can overcome the dead-weight of the turbine assembly and thus spontaneously establish a full hydrostatic film.

Delays in hydrostatic lifting have been noted in the literature (e.g. [8]). At startup, initial pressurisation is important as the bearing lands do not contribute to the generation of hydrostatic pressure. Only if the turbine is lifted externally, or if imperfections allow percolation of oil between the contacting surfaces, can sufficient area be acted upon by the pumped pressure. It must be remembered that the supply pressure needs to be limited because of the relatively soft low-friction material lining the pad. Oil percolation cannot be guaranteed, as even careful machining of the recess may introduce a raised lip at the edge that can make an effective seal. The only way to circumvent this uncertainty is to have a sufficiently large groove, given a sensibly limited supply pressure of, say, less than 16 MPa, which can generate enough force to lift the turbine and establish a fully-developed hydrostatic flow. The necessity for such measure may be debatable; however, with the increasing requirements for rapid and frequent startup for effective grid balancing, the rationale for spontaneous lifting becomes stronger.

Shutdown, although critical due to the possibility of thermal crowning and pad damage, is not significantly affected by the size of the recess as the prior separation of the surfaces means that hydrostatic pressure is generated over the entire pad area. At startup, any small sliding movement from energising the turbine will assist the hydrostatic oil to invade the interstices of the lands, leading to the establishment of hydrostatic flow. Any wear due to initial sliding should be extremely small. An alternative would be to use hydraulic jacks to lift the turbine just enough to prime the oil film under cold-start conditions. Notwithstanding the above arguments, some interesting observations have been made as a result of the current investigation on large recesses and their impact on bearing performance under hydrodynamic conditions. On one hand, bearing efficiency seems to be improved under typical operating conditions; on the other hand, stability at low temperatures may be compromised.

2. THEORY AND DEVELOPMENT

2.1. Principles of hydroelectric turbine design

The principle of operation of a hydroelectric turbine is illustrated in figure 1(a). The thrust bearing consists of spring-supported “sector-shaped” pads which bear the weight of the turbine assembly and the hydraulic thrust generated by the water as it flows past the water wheel. The number of pads is arbitrary; however, the convention is for pads not to be excessively elongated in any one direction, as shown for example in figure 1(b). Two journal bearings typically constrain the turbine radially. The springs and additional constraints (not shown in the figure) allow movement of the pad in three degrees of freedom: two in rotation (governed by torsional stiffnesses κ_x and κ_y) and one in translation (governed by axial stiffness k_z). The tilt of a pad is governed by equilibrium between the spring torque and the torque induced by the hydrodynamic pressure acting over the pad’s surface.

The pads are usually lined with Babbitt metal for low friction and reduced damage potential to the runner surface. The pad lining surface must be very precisely finished to a high degree of flatness, often by the process of scraping. The pads are usually constructed from steel or cast iron materials with excellent dimensional stability to prevent warping. The thickness of the pads is relatively large to prevent flexing under load. The thickness is about one-sixth of the pad dimension which makes any flexing almost negligible. The coefficient of thermal expansion of the pad material is very important as the temperature gradient perpendicular to the pad surface introduces a certain amount of thermal deformation. Thermal crowning is unavoidable and deliberately accounted for as it actually improves the performance and stability compared to a perfectly planar pad. The runner is flat to high precision and lapped to prevent abrasion of the soft pad lining. A recess for hydrostatic assistance can be machined into the Babbitt lining as shown, with details for this case study, in figure 1(c). The shape and location of the recess have an important influence on the operation of the bearing.

2.2. Reynolds' equation

The basis of the numerical simulation is Reynolds' equation,

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(h^3 \frac{\partial p}{\partial y}\right)=6\eta\left(U \frac{\partial h}{\partial x}+V \frac{\partial h}{\partial y}\right)$$

where p is the film pressure, h is the film thickness, η is the iso-viscous dynamic viscosity and $U=U_1+U_2$ and $V=V_1+V_2$ represent the sum of local surface velocities of the pad and runner in the x- and y-directions respectively. The pad surface is fixed and therefore contributes zero to the entraining velocities U and V. Converting to a finite difference relation of the first order, one may obtain the following iterative relationship for the solution of pressure over the domain of the bearing,

$$p_{i,j}=\frac{\dfrac{p_{i+1,j}+p_{i-1,j}}{\delta x^2}+\dfrac{p_{i,j+1}+p_{i,j-1}}{\delta y^2}+\dfrac{3}{h}\left[\dfrac{\partial h}{\partial x}\left(\dfrac{p_{i+1,j}-p_{i-1,j}}{2\delta x}\right)+\dfrac{\partial h}{\partial y}\left(\dfrac{p_{i,j+1}-p_{i,j-1}}{2\delta y}\right)\right]-\dfrac{6\eta}{h^3}\left(U\dfrac{\partial h}{\partial x}+V\dfrac{\partial h}{\partial y}\right)}{\dfrac{2}{\delta x^2}+\dfrac{2}{\delta y^2}}.$$

2.3. The effect of speed and viscosity

For a simple square linear pad and iso-viscous conditions, it may be shown that the adiabatic average oil film temperature rise is independent of surface velocity and viscosity,

$$\Delta T_{\mathrm{av}}=\frac{W}{\rho\sigma B^2}K^2\left(\frac{2(K+2)\ln(K+1)}{K(K+1)}-\frac{3}{K+1}\right)\div\left(\ln(K+1)-\frac{2K}{K+2}\right)$$

yet it remains strongly dependent on the load W relative to square pad area B^2, properties of the oil (specific heat σ and density ρ) and the convergence number ($K=h_{\mathrm{in}}/h_{\mathrm{out}}-1$). This behaviour is somewhat consistent with that observed in the simulation results insofar as, at relatively low viscosities, a baseline temperature is established which is independent of speed and viscosity. It must be remembered that in the real bearing, the convergence is not static and can change in two planes depending on the combined effects of speed and viscosity.

2.4. Thermal considerations

In the simulation, evaluation of thermal effects involved calculating the iso-viscous adiabatic average temperature rise of the oil film. This represents a worst-case scenario and is only plausible when there is negligible time for the heat generated by the viscous shearing of the oil to conduct into the surfaces of the runner and pads. The phenomenon depends on film area, surface speed, film thickness, mass and thermal properties of the oil. Simple modelling of the system reveals that conduction of heat through the oil film to the metallic surfaces is significant, which is beneficial as it means that the viscosity drop of the oil film will not be as severe as predicted by the adiabatic condition. If the ratio of heat generated by convection to that transferred by conduction to the bearing surfaces can be approximated by the formula,

$$\beta=\frac{H_{\mathrm{conv}}}{H_{\mathrm{cond}}}\approx\frac{2\chi B}{Uh^2},$$

where χ is the thermal diffusivity of the oil, B is the linear dimension of the pad, h is the average film thickness and U the average sliding velocity, then for the bearing in question typically about 50% of the heat generated is dissipated by conduction. The ratio beta is sometimes referred to as the Peclet number. This can help estimate the temperature rise of the oil film compared to the adiabatic temperatures obtained from the computer simulation.

2.5. Thermal distortion of the pad

The pads considered here are sufficiently thick that elastic distortion can be considered to be small compared to thermal distortion. The thermal distortion can be estimated by assuming spherical crowning arising from a uniform and linear temperature variation from one face of the pad to the other. Both the coefficient of thermal expansion (11×10^{-6} °C^{-1} for steel) and thickness of the pad are needed for the calculations. Thermal crowning can be a useful and stabilising phenomenon and should be taken into consideration in the optimisation of bearing design. The change of pad shape causes the pressure distribution to move forward relative to the spring centre, thus promoting a convergent film profile – one of the requirements for hydrodynamic film formation. Concavity of the pad of even a few micrometres, arising from wear or manufacturing defects, could result in film collapse because the pressure force would act behind the spring centre and tilt the pad the wrong way. This results in large friction forces, high surface temperatures and, potentially, damage to the bearing surface. Fortunately, in most cases, this is averted through the compensating phenomenon of thermal crowning. Minor cavitation was observed during simulation of the most crowned pads; this was taken into consideration in the calculation of local forces and shear stresses.

3. EXPERIMENTAL METHOD AND RESULTS

3.1. Computational details

A square grid covering the surface of the bearing was used. During Gauss-Seidel iteration, the forces generated by the oil film pressure distribution were used to adjust the film geometry and also the hydrostatic pressure when appropriate [9]. The film geometry depends on pad height as well as rotation in two planes. Eventually, the geometry stabilises when the pressure forces are in equilibrium with the spring forces. The process takes less than five minutes on a standard desktop computer for a 300×300 grid and demonstrates good grid convergence (within three percent difference in film height compared to a 2000×2000 grid). The viscosity variation of the oil with temperature is shown in table 1, which represents an ISO68 mineral-base hydraulic oil.

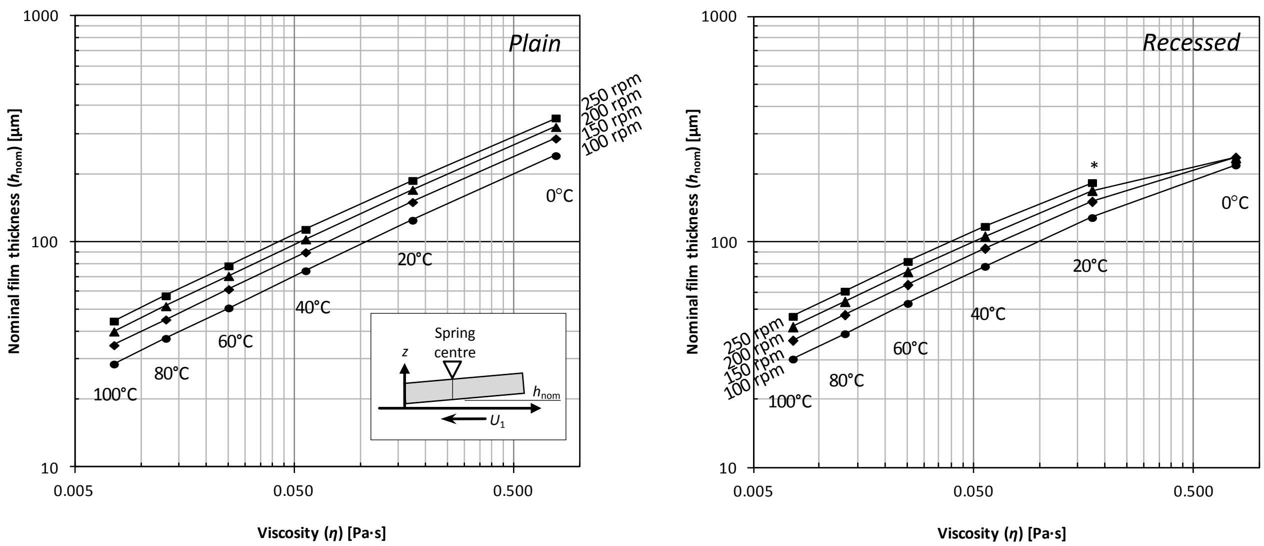
Table 1: Viscosity extrapolation of ISO68 oil as per ASTM D341.

Temperature (°C)	Viscosity (cS)	Viscosity (Pa·s)
0	900	0.783
5	600	0.522
12	360	0.313
25	150	0.130
50	43	0.037
75	17	0.015
100	9	0.008

3.2. Variables studied

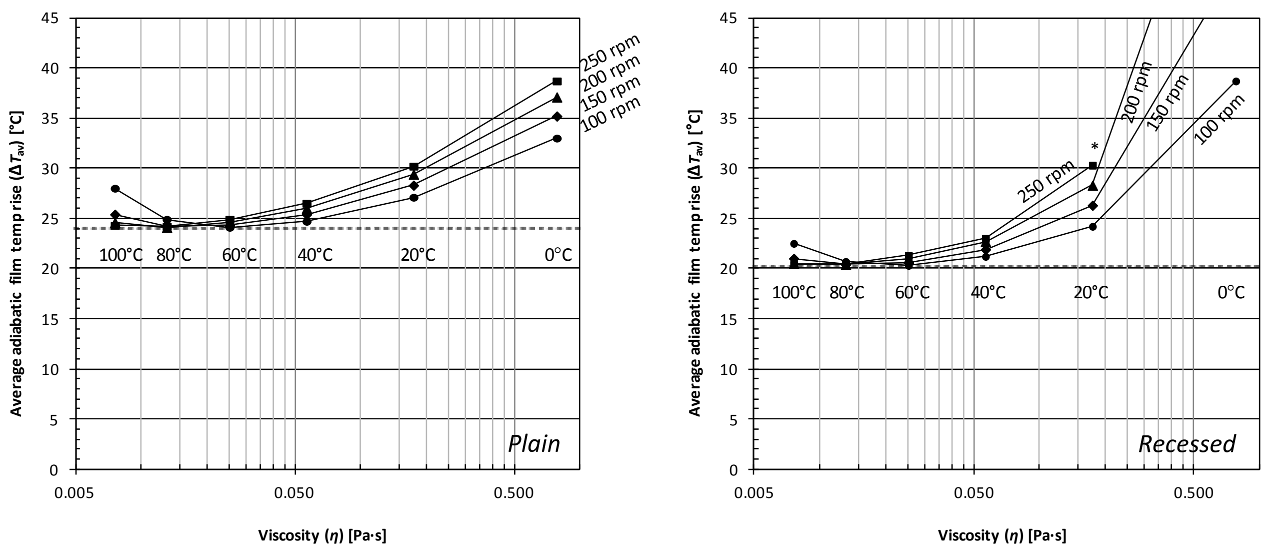
The variables studied include the nominal film thickness h_{nom} (measured at the spring centroid), as a function of the iso-viscous film viscosity, at different speeds as shown in figure 2(a) for the plain linear pad and 2(b) for the recessed pad, acting under purely hydrodynamic conditions (no hydrostatic pressurisation). It can be seen in the first case that there is a strong relationship between film thickness and oil viscosity of the sort $h_{nom} \propto \eta^{1/2}$. For the recessed condition, the relationship holds for high temperature or low viscosity, but it seems to break down at low temperature or high viscosity. It should be noted that the nominal film thickness is greater for the recessed case. This is because the recessed pressure distribution causes the pad to favourably increase its convergence towards the optimum. At higher viscosity the opposite is true for the recessed pad.

The effect of viscosity on pad film thickness and efficiency is important and must be studied. In order to do this, an energy balance was conducted using the local node shear stress and velocity, and the fluid flow entering the pad. The average adiabatic film temperature rise ΔT_{av} was then calculated and plotted for the plain and recessed pads, for different values of viscosity and rotational speed, as shown in figure 3. The interesting point to note is that the presence of the recess reduces the minimum temperature possible, as represented by the horizontal dashed line. At high viscosity the groove is detrimental to performance and potentially causes the oil film to "collapse".



(a) (b)

Figure 2. Nominal film thickness for (a) plain pad, (b) recessed pad.
*No solution possible at 250 rpm, 0°C for recessed pad due to solution divergence.

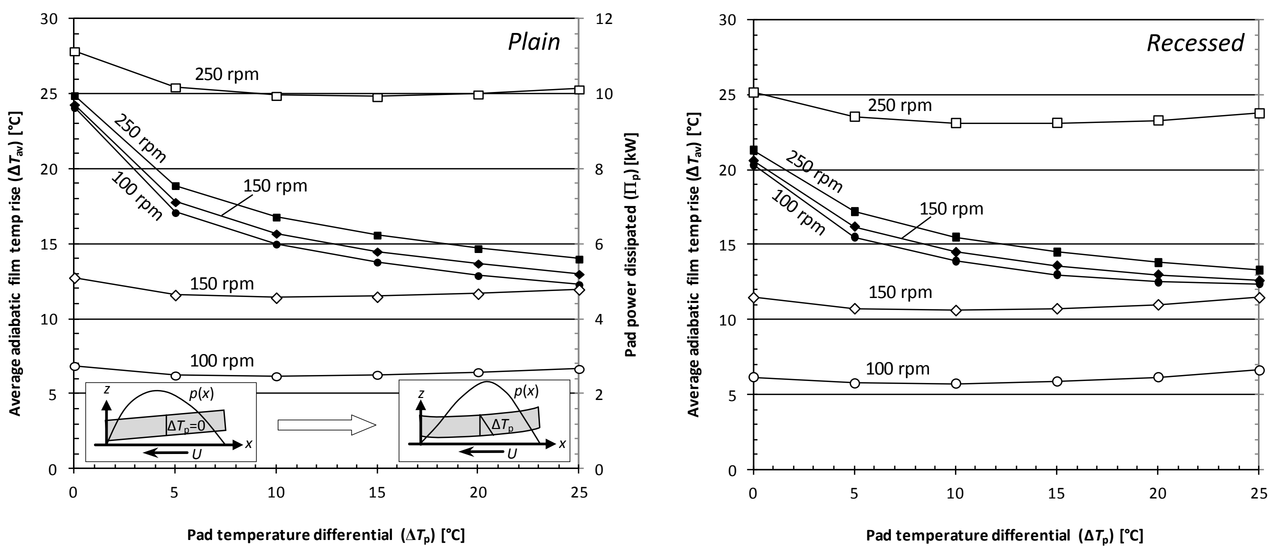


(a) (b)

Figure 3. Average adiabatic film temperature rise for (a) plain pad, (b) recessed pad.
*No solution possible at 250 rpm, 0°C for recessed pad due to solution divergence.

So far, the effects of thermal crowning have been ignored, but in reality such simplification is unacceptable. Consequently, the effect of crowning resulting from a higher thermal gradient normal to the pad surface was investigated and summarized in figure 4 at a nominal viscosity corresponding to a film temperature of 60°C. The effect on both ΔT_{av} and the frictional power dissipated Π_p are shown in figures 4(a) and (b) for plain and recessed pads respectively. It can be seen that thermal crowning reduces the temperature rise for both plain and recessed pads. The temperature difference is only marginally lower for the recessed pad at higher temperature differentials. The power dissipation is relatively stable, with better performance at moderate pad temperature differentials. The presence of the recess is beneficial for power reduction under most of the conditions shown in figure 4. This is due to the fact that viscous shear rates are smaller in the region of the recess when compared to the plain pad [10].

If the assumption is made that the surface temperature of the pad is half of the adiabatic temperature rise, then it can be estimated that the pad temperature differential ΔT_p is approximately equal to 8.5°C at 150 rpm for the plain pad which corresponds to about 6 μm height difference 100 mm away from the spring centre. Importantly, it means that the pad is close to optimally designed as the power dissipation is minimised. The same applies to the recessed pad, which, however, dissipates between 5 and 10% less power. It must be remembered that this analysis applies when the average oil film temperature is close to 60°C. Nonetheless, it is expected that the recessed pad will be more efficient than the plain pad when the average film temperature is above 40°C. This is generally well below normal operating film temperatures.



(a) (b)

Figure 4. Crowned pad performance for nominal oil viscosity (η) at 25.2 mPa·s (60°C). (a) Plain pad, (b) recessed pad. Left ordinate (ΔT_{av}) for solid markers; right ordinate (Π_p) for hollow markers. The recessed pad exhibits lower power dissipation and adiabatic temperature rise, especially at low crowning where the temperature gradient is small.

4. DISCUSSION

The addition of a hydrostatic recess appears to be beneficial in this case study. The groove satisfies the requirements for provision of spontaneous hydrostatic lifting of the turbine assembly. The author's experience with other installations shows this to also have been the case. However, any modifications should be made with caution as the operation of such thrust bearings is sensitive to the planarity of the pads. The simulations show that spherical concavity of the pad surface by as little as a few micron over a 100 mm radius, fails to generate the necessary convergence to establish a hydrodynamic film.

Thermal crowning may be considered both a blessing and a curse. Repeated rapid shutdown of a crowned pad (see, for example [11]) can cause enough wear to eventually induce a concave profile under startup conditions.

This leads to more heating, which leads potentially to more wear. If the defective pad's temperature is not monitored then catastrophic failure can occur with little warning. This is one plausible mechanism for failure of a bearing design which is otherwise extremely robust. The correct use of hydrostatic lubrication during startup and shutdown to cool the pad surfaces should prolong the lifetime and reliability of the bearings substantially.

The influence of the recess on performance may be summarized as follows. The hydrodynamic pressure is leveled in the recess at low viscosity. This shifts the pressure centroid forward and increases the convergence of the pad. In the absence of thermal crowning this has a positive effect on both film thickness and efficiency as far as this case study is concerned. At high viscosity, the pressure distribution under the tapered regions of the recess are no longer flat and unfavourably tilt the pad to a smaller convergence, leading to reduced film thicknesses and higher power dissipation. It has been found that tapers on the outer part of the groove promote this negative behaviour and should be avoided [9]. The saving grace is that the higher film temperature increases crowning and also reduces oil viscosity, which both produce a stabilising effect on bearing operation. Nonetheless, it must be said that stability of the plain pad is overall superior to that of the recessed pad.

5. CONCLUSIONS

The following conclusions were drawn from this work:

- A simple and fast iso-viscous computational model of a plain-pad thrust bearing has been developed to investigate the effects on performance of adding a relatively large hydrostatic recess.
- One advantage of this method is that almost any geometry can be investigated relatively easily and quickly.
- In this case study, the presence of the recess improves the tilt of the pads to give a thicker oil film at film temperatures above 40°C.
- Viscous power dissipation is reduced by about 10% due to the presence of the recess under normal operating conditions. This is due to a combination of improved film geometry and reduced viscous shear.
- The benefit of the groove is less evident at low film temperatures and when the effect of thermal crowning of the pads is included in the simulation.
- There is the potential for unstable operation of the recessed pad at very low temperatures or high oil viscosity.
- The most significant advantage of large recesses is the immediate effect of hydrostatic lubrication at startup.

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