

The Chevron Foil Thrust Bearing: Improved Performance Through Passive Thermal Management and Effective Lubricant Mixing

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1. Introduction

An improved foil thrust bearing is described that eliminates or reduces the need for forced cooling of the bearing foils while at the same time improves the load capacity of the bearing, enhances damping, provides overload tolerance, and eliminates the high speed load capacity drop-off that plagues the current state of the art. The performance improvement demonstrated by the chevron foil thrust bearing stems from a novel trailing edge shape that splays the hot lubricant in the thin film radially, thus preventing hot lubricant carry-over into the ensuing bearing sector. Additionally, the chevron shaped trailing edge induces vortical mixing of the hot lubricant with the gas that is naturally resident within the inter-pad region of a foil thrust bearing. The elimination of hot gas carry-over in combination with the enhanced mixing has enabled a completely passive thermally managed foil bearing design. Laboratory testing at NASA has confirmed the original analysis and reduced this concept to practice.

2. Background

The foil bearing has been utilized in high speed rotating machinery for many decades [1]. These bearings have been attractive options for lightweight machines because they offer numerous system level benefits such as overall simplicity, reduction in weight, reduced friction, enhanced reliability, and zero oil contamination potential. The primary technical challenge in the application of foil bearing technology to high speed rotating machinery is the dual use of the system process fluid as the hydrodynamic lubricant. Typically process fluids are low kinematic viscosity fluids that are used to enhance the performance of the primary machine, however this property is contrary to the requirements of a good lubricant. Decades of development have produced advanced foil bearings that match stiffness of the compliant foundation with the hydrodynamic film pressure. The foil journal bearings have advanced to the point that considerable consideration has been given to their application to large gas turbine engines [2,3,4]. However, the primary limitation to more widespread application of foil bearing technology has been unacceptable performance of the foil thrust bearing and their inability to efficiently react axial machine loads. Specific limitation of the foil thrust bearing include low load capacity, low damping, high friction, load capacity drop-off at high speed, and unpredictable failures.

3. Foil Bearing Operating Regions

Traditional hydrodynamic lubrication theory provides two distinct regions of operation for the approximation of load capacity as a function of rotational speed. While operating at low speeds the load capacity follows a linear trend with speed. The specific approximation can be calculated with a simplified form of the incompressible Reynolds equation. For the case of a straight taper bearing with no side leakage the load capacity can be determined in closed form by the following algebraic equation.

$$w(k, h_2, \mu, U, L) = \frac{6 \mu U L^2}{(k-1)^2 h_2^2} \left\{ \text{Log}(k) - \frac{2(k-1)}{k+1} \right\} \quad (1)$$

The load capacity in the above circumstance is a function of absolute viscosity (μ), bearing length (L), contraction ratio (k), minimum film thickness (h_2), and shaft velocity (U).

At very high speeds the load capacity reaches an asymptotic limit where increases in speed no longer increase the capacity of the bearing. For this case the Reynolds equation can be simplified to the form that indicates the quantity of pressure ratio multiplied by the contraction ratio is a constant. The closed form solution to the load capacity then can be written as equation 2.

$$w_1(k, L) = \frac{L(1-k+k \text{Log}[k])}{-1+k} \quad (2)$$

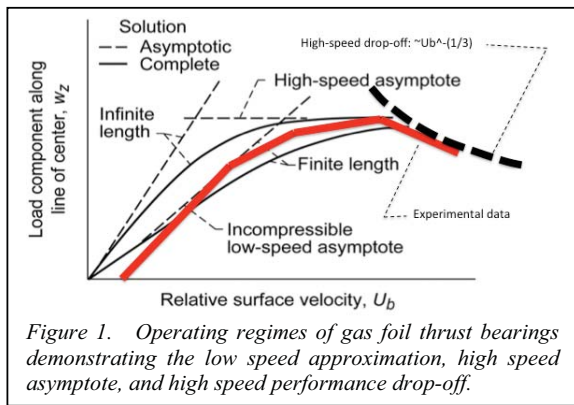
Here we see that the load capacity is only a function of bearing length (L) and contraction ratio (k). The shaft speed no longer affects the load capacity in this regime. Surprisingly, the minimum film thickness is also absent from the high speed load capacity prediction.

A third limit is imposed at ultra high speed or when the heat capacity of the lubricant and thermal management of the bearing are insufficient to maintain equilibrium with the viscous shear heating. In this third regime the load capacity can actually decrease with increasing speed and this has been found to be the case with the foil thrust bearing. This high speed load capacity drop-off is a purely function of high lubricant shear. The effect is enhanced by sporadic asperity contacts, but the load capacity drop-off is initiated by high shear in a full film condition. This high shear results in lubricant

temperature rise through the dissipation function and mass flowrate consideration. Combining these effects, the proportionality of load capacity to shaft speed is given by equation 3.

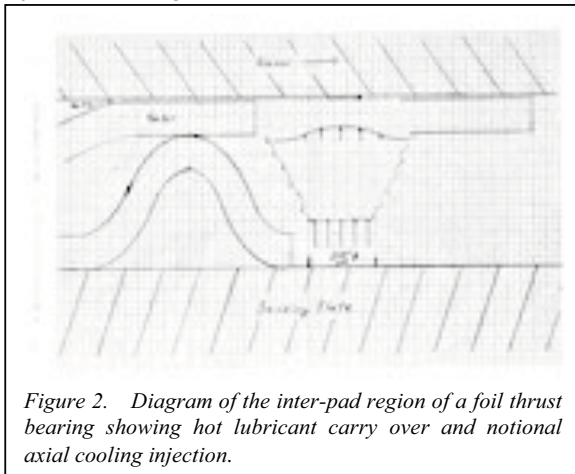
$$w_2(k, L) \sim \left(\frac{\rho C_p \delta T}{\mu U} \right)^{1/3} \quad (3)$$

Figure 1 demonstrated the general nature and relative location of each of these regimes on a load capacity versus speed chart. The red line in figure 1 represents a large dataset of experimental foil thrust bearing performance measurements made at the NASA Glenn Research Center at speeds up to 70000 rpm on 100 mm diameter thrust bearings.



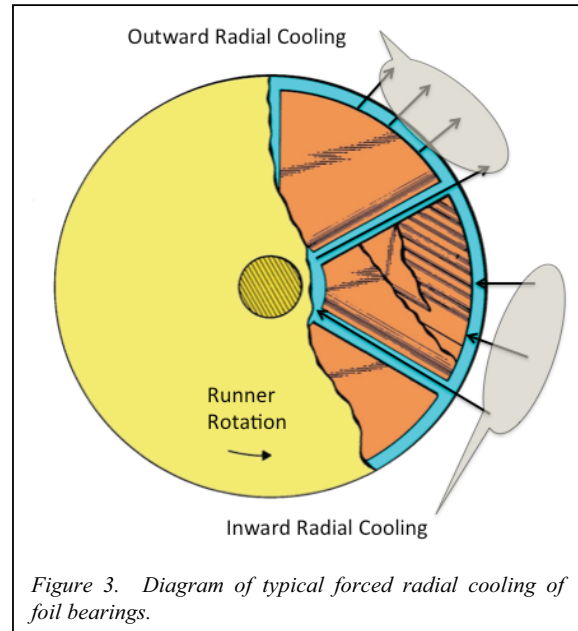
4. Current Thermal Management Methods

The current state of the art for thermal management of thrust foil bearings utilizes high pressure air injected normal or parallel to the rotating surface as a means to maximize forced convective cooling of the bearing sectors. These concepts are shown in figures 2 and 3. Figure 2 is included to highlight the physical scale of the hot gas in the fluid film compared to the geometry of a typical foil thrust bearing and axial force cooling injection. The figure demonstrates that the scale of the



lubricant film near the trailing edge of a bearing sector is on the order of 5 micrometers while the compliant foundation (bump foils) of a typical foil bearing are on the order of 500 micrometers. This scale clearly indicates two key points. The first point would be that it is nearly impossible to remove the thin layer of hot lubricant from the runner surface by using forced convection. The second key point is that the foil bearing cavity has an abundance of lubricant present at all times and it is the mission of the foil bearing designer to efficiently utilize this flooded lubricant condition.

The use of forced cooling in the current state of the art leads to several deleterious side effects. Primarily the performance of the overall system is compromised because the forced cooling necessarily robs the process fluid of the machine. Secondly the high flowrates required to cool a foil bearing often lead to substandard hydrodynamic conditions for the load supporting sectors. Finally, forced cooling often masks the true stress of an operating foil thrust bearing, which causes bearing failure to be chaotic and unpredictable.

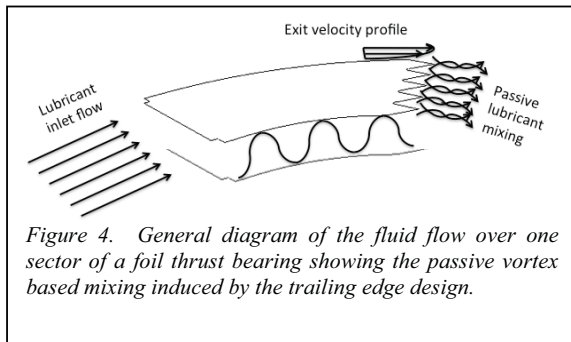


5. Passive Thermal Management through Sector Trailing Edge Shaping

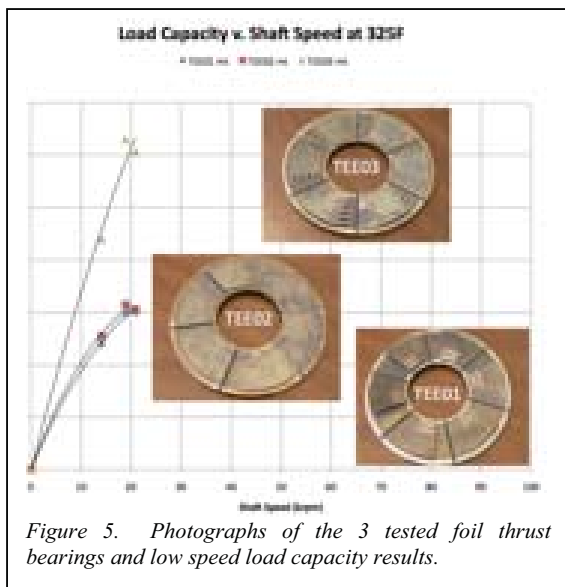
To overcome the deleterious effects of high speed load capacity drop-off and of forced convective cooling for foil thrust bearings an innovative concept was identified that utilized the vast amounts of gas, which is normally present in the bearing cavity, to refresh the lubricant prior to entry into the ensuing sector. This concept modifies the trailing edge of each top foil to enhance mixing in the inter-pad region. The first trailing edge shape that was tested resembled the chevron nozzles, which are now common in modern aircraft gas turbine engines. The shape of these nozzles has been designed

to enhance fluid mixing between the engine flowpaths and the ambient air. A similar function is required by the foil bearing, namely, the trailing edge treatment is needed to mix the hot lubricant flow with the stagnate interpad gas. Since this concept in the aircraft engine domain has been named the chevron nozzle a similar name was coined for the bearing concept, the chevron foil thrust bearing.

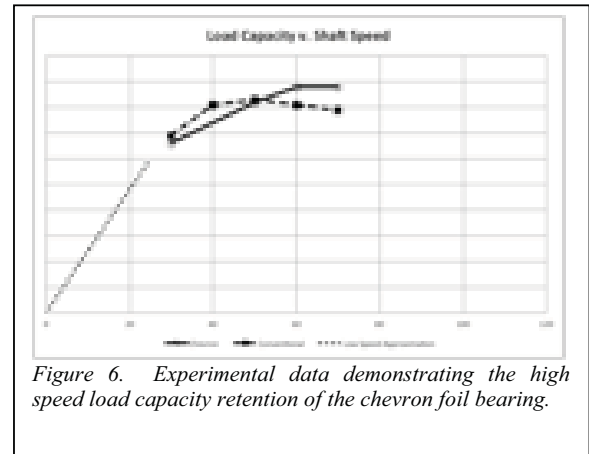
By utilizing the top foil to initiate mixing, the hot lubricant is impacted directly and, in fact, initiates the mixing process thus preventing the hot lubricant carry over from one sector to the next. The key features of this concept are shown in figure 4.



In order to advance this innovation from concept into practice, a family of foil thrust bearings has been designed, manufactured, and tested to verify the improved performance of the chevron bearing. This family of bearings is shown in figure 5. Differences among these three bearings are restricted to the top foil trailing edge. All bearings are manufactured according to the methods described by Dykas, et al. [5]. Bearing TEE01 is a baseline design having a conservative trailing edge to leading edge gap of 15 degree. Bearing TEE02 utilizes a common approach to increase load capacity by increasing sector area; hence the trailing edge to leading edge gap is zero. TEE03 is the



baseline chevron foil thrust bearing having 5 trailing edge chevrons. Low speed performance of the three bearings is shown in figure 1 while figure 2 contains high-speed performance. Load capacity of the chevron foil thrust bearing at low speeds is improved due to its tolerance to mixed lubrication situation that dominates load capacity testing, while high speed performance is maintained due to the ability of the chevrons to break apart the hot lubricant film and prevent hot gas carry-over and thermal run-away failures.



6. Conclusion

The chevron foil thrust bearing offers a significant improvement to the current state of the art in foil bearing technology and directly addresses the weak link in the application of this technology to more challenging machines. Optimization of the trailing edge design in concert with the compliant foundation and specific machine needs will enable a more complex systems to reap the benefits of oil-free turbomachinery technology.

7. Nomenclature

C_p	specific heat capacity of lubricant (kJ/(kg K))
h_1	sector inlet film thickness (m)
h_2	sector minimum film thickness (m)
k	contraction ratio
L	length of bearing sector (m)
U	velocity of runner (m/s)
μ	dynamic viscosity (Pa-sec)
w_x	load capacity per unit width (N/m)
dT	temperature change across sector (K)
ρ	density of lubricant (kg/m ³)

8. References

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