American Society for Precision Engineering

ENABLING BEARING TECHNOLOGIES FOR HIGH-POWER OIL-FREE TURBOMACHINERY

Bugra Ertas
Mechanical Systems
GE Research Center
Niskayuna, NY 12309 USA

ABSTRACT

The following topic will focus on the recent advancements in gas film bearing technology aimed at enabling oil-free operation of high-power turbomachinery. Virtually all turbomachines such as compressors, turbines, and electric machines in the megawatt power range today utilize oil-lubricated bearing systems to support the rotating shaft assembly. To date these bearing systems have provided reliable operation of rotating equipment. However ever-increasing demands in improved power density and efficiency are challenging engineers to consider advanced bearing systems utilizing the working fluid of the machine to lubricate bearings. Using process-fluid lubricated gas bearings versus oil-bearings not only yields improved performance, but also is an enabler for emerging applications on the fringe of turbomachine technology development.

The present work provides an overview of existing gas bearing technologies followed by the challenges associated with implementing these devices into large turbomachinery systems. With these hurdles in mind, the paper reviews recent efforts by researchers to improve these engineering metrics. Finally, the work herein outlines the evolutionary progression of compliant-hybrid gas bearings over the last decade; ultimately requiring the use of additive manufacturing. Concluding remarks focus on the remaining challenges and technology hurdles for realizing oil-free operation in megawatt-power-class turbomachinery.

INTRODUCTION

Virtually all high-power turbomachinery in the megawatt class range utilize oil-lubricated bearings to lubricate the rotating assembly. Oil is a preferred lubricant due to its abundance, hydrodynamic pressure generating capability, and the ability to provide high levels of damping due to the relatively high viscosity when compared to gases. However, implementing an oil-system requires the engineering of complex lubrication circuits, bearing sumps, and ancillary components. The oil needs to be protected from the process fluid of the machine and visa-versa. The concept outlined in this paper replaces oil with the process fluid of the machine to lubricate the bearing. With this approach entire lube system and can be eliminated while greatly simplifying the mechanical design of bearing sumps. Furthermore, process gas lubricated bearing systems can expose new design spaces allowing for novel machine architectures. If done successfully, process fluid lubrication yields benefit in cost, reliability, and performance. A good example comes from implementation of foil bearings in air cycle machines and microturbine generators [1,2]. These types machines have demonstrated remarkable reliability with gas foil bearings but are limited to low power ratings typically <500KW. The two main challenges existing with gas bearings, like foil bearings, for penetration into megawatt class turbomachinery are load capacity and damping.

TECHNOLOGY HURDLES

Existing state-of-the-art gas bearings used in turbomachinery are mainly foil bearings.

Foil bearings are a type of hydrodynamic bearing with a flexible and compliant bearing surface (Figure 1). Pressure generation is created with relative speed between the journal and top foil. This pressure generation is responsible for the load carrying capacity and becomes larger with increasing rotor speed. Figure 2 shows a typical representation of how a foil bearing operates in a mission cycle.

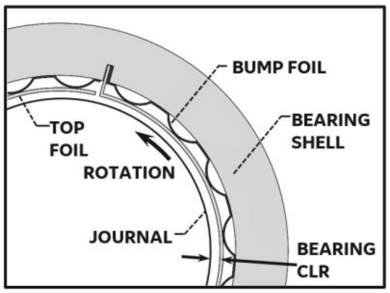

FIGURE 1. Typical Foil Bearing

Initially at 0rpm, since there is no journal speed, the journal is resting on the bearing foil surface. As the journal starts to rotate there is sliding contact and friction between solid surfaces. This regime creates a start-up torque spike. As journal speed increases the bearing transitions through a boundary layer (B.L.) lubrication regime followed by complete hydrodynamic lift-off with a fully developed lubricating film. In result, the foil bearing needs to be able to sustain several start-stop cycles enduring sliding friction between parts. To achieve this, coating needs to be employed. With this approach the foil bearing is limited to lighter rotor weights and therefore lower power applications as the coating technology is the limiting factor.

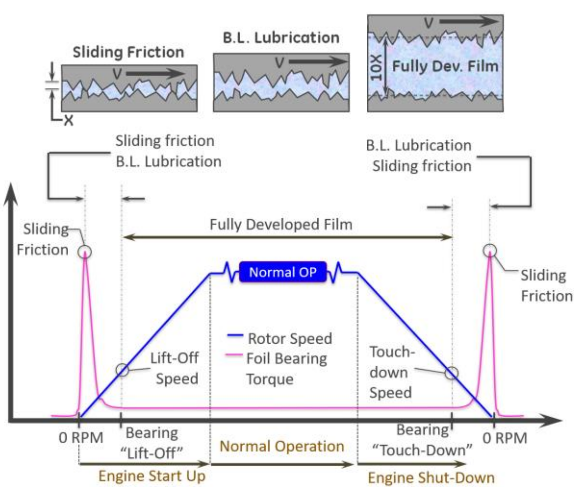

FIGURE 2. Torque-Speed Trend for Foil Bearings Showing Lubricating Regimes

Therefore, the first main technical hurdle for implementing gas lubricated bearings to larger equipment is load capacity. The second hurdle is damping. Damping is a mechanical force element that opposes vibratory velocity of dynamic mechanical systems. Figures 3-4 illustrate how damping influences vibration of rotating systems and how the requirement of damping increases as the stiffness coefficient of the system is increased.

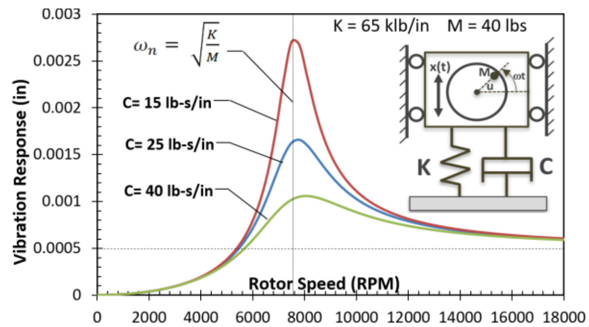

FIGURE 3. Synchronous Response to Rotor Unbalance for Varying Damping Coefficients While Keeping Stiffness Constant

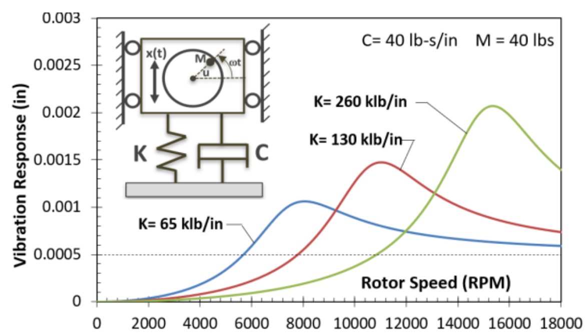

FIGURE 4. Synchronous Response to Rotor Unbalance for Varying Stiffness Coefficients While Keeping Damping Constant

Figure 3 shows a simple mechanical system with a rotating unbalance force as the forced excitation. In this figure damping is varied as stiffness is held constant. As one would expect the vibration level is reduced as the damping coefficient is increased. Figure 4 holds damping constant but varies the stiffness coefficient. In this case note that the vibration increases as the stiffness coefficient becomes larger in magnitude. The demand for damping goes up for the high stiffness case in order to retain the same amplification factor as the lower stiffness case. This simple example illustrates that damping alone is insufficient to define the damping capability of a spring damper system; what is required is the ratio of stiffness to damping. This is famously defined by the damping ratio (eq. 1), which relates the critical damping (eq. 2) to the absolute damping in the system. Where ζ is damping ratio, C is damping, C_c is critical damping, K is stiffness, and M is mass. Note that the critical damping is a function of stiffness.

$$\zeta = \frac{C}{C_c} \tag{1}$$

$$C_c = 2\sqrt{K \cdot M} \tag{2}$$

In summary, engineering a low stiffness in combination with high damping maximizes the damping ratio.

In the case for gas films found in bearings, the ratio of stiffness to damping is very high and therefore when compared to oil-based bearing systems the damping capability of gas films is significantly compromised. This is where the foil bearing concept takes advantage of the "soft" support to improve equivalent damping of the bearing system.

EVOLUTION OF GAS BEARING SYSTEMS

This section builds on the main two technical challenges highlighted in the previous section and gives a cursory overview of the approach used to evolve gas bearings to mitigate the load capacity and damping challenges. As power rating of the turbomachine increases, the rotor diameters and weight becomes larger. Figure 5 shows a high levels chart illustrating the landscape for hydrodynamic versus hydrostatic bearing systems.

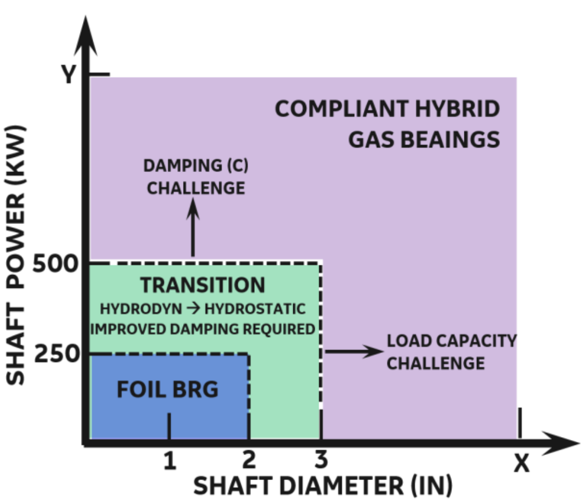

FIGURE 5. Radial Gas Bearing Design Space

As the shaft diameter increase the mass of the rotating assembly also increases. With mass increasing hydrostatics are required to support rotor loads. Hydrostatic pressurization involves using an external pressure source to lift the rotor and can be achieved through several configurations and gas delivery methods. Increased shaft power also implies higher levels of fluid forces between the rotating assembly and stationary casing of the machine. These fluid forces, either from the turbomachinery stages or seals, generate destabilizing dynamic forces. To mitigate high vibration generated from destabilizing forces, one must introduce higher levels of damping. The key to generating high levels of damping in gas bearings resides in utilizing a soft bearing support in series with the gas film. Compared to rigid geometry gas bearings, either hydrodynamic or hydrostatic, soft mounting the gas film on a flexible support that possesses a favorable stiffness to damping ratio generates superior damping capability to the rotating system. Soft-mounting a bearing system is not a new concept as aircraft engine systems utilize flexible bearing supports with rolling element bearings [3]. Soft-mounting the bearing system transforms rotor bending modes to rigid body modes, reduces transmitted dynamic bearing loads, and can significantly reduce rotor vibration magnitudes. Figure 6 illustrates how a soft support would maximize the equivalent damping of the bearing system. The simple model in Figure 6 shows that the ratio of gas film stiffness (K_a) and bearing support stiffness (K_b) have strong influence on the equivalent damping of the overall system (C_{eq}). As the bearing support stiffness (K_b) is reduced, the equivalent damping (C_{eq}) approaches the bearing support damping (C_b).

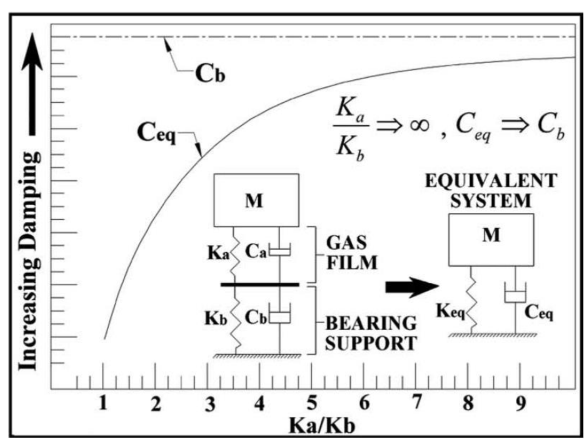

FIGURE 6. Maximizing Equivalent Damping Using Soft Bearing Supports

This bearing and support configuration can allow for high damping even with a relatively low valued gas film damping (C_g) coefficient.

Considering the previously discussed approaches for obtaining higher load capcity and damping, considerable research has been performed on compliant hybrid gas bearings. Figure 7 shows the evolution of compliant hybrid gas bearings over the last decade. The bearing concept involves using hydrostatic pressurization in combination with a flexible support possessing a robust damper. Initially the generation 1 (Gen1) bearing used a simple square hydrostatic recess to deliver pressurized fluid between the bearing surface and journal. The GEN1 concept [4] further implemented an oil-free metal mesh damper into the bearing support. Although these two novel additions to existing hydrostatic bearings improved load capacity and damping when compared to foil bearings, there were still limitations in supporting edge loading and mitigating vibration response on heavy rotor systems. From this conclusion the development aimed to improve the gas delivery method and the damping element in the bearing support. The GEN2 [5] concept addresses these shortcomings through implemented a distributed gas delivery system using multiple holes or porous media to enhance the edge loading capability of the bearing pad. The GEN 2 design improved the damping capability through the development of hermetic squeeze film dampers rather than having metal mesh dampers in the bearing support. The concept proved to have significantly higher damping levels when compared to friction based damping elements like foils or metal mesh. However, the GEN2 design used modular fluid dampers and resulted in over 500 parts in the assembly; making the design impractical for industrial applications. The GEN3 concepts were developed [6-8], which used additive manufacturing or diffusion bonding to integrate complex functionality into a single piece bearing design. The integral designs, especially the additive approach, yielded a cost effective and compact configuration showing good promise for implementation into high-power turbomachinery. However, key challenges in the wear-couple between the bearing and journal remain.

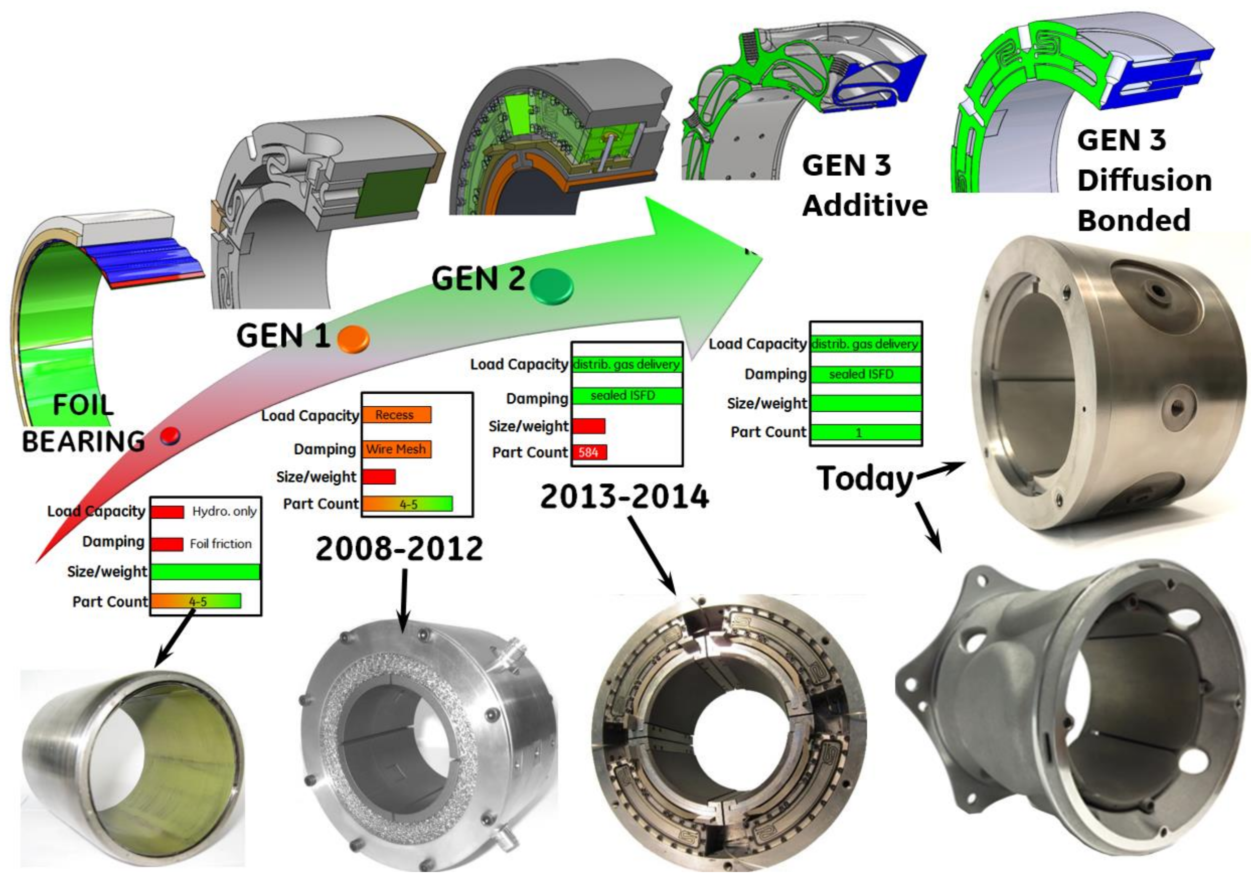

FIGURE 7. Evolution of Compliant Hybrid Gas Bearings

REMAINING CHALLENGES

In addition to the need for higher damping and load capacity for mission critical high-performance turbomachinery, there is a requirement of the gas bearing system to sustain unintended touch-down conditions during off-design points. For example, aircraft engines require the ability of the bearing system to absorb blade-out forces and bearings in centrifugal compressors during surge and stall must survive loads an order of magnitude higher than the 1G reaction load. Under these circumstances it is most likely that the bearing surface will experience a rub at high surface speeds and high loads. Coating technology or the tribological wear-couple between the rotor and bearing becomes of paramount importance.

One highly promising technology that can be implemented in these max-load off-design cases are porous carbon media bearings. Porous carbon bearings (Figure 8) have a unique crystalline structure that allows for easy slip of parallel planes in the structure due to sliding motion while retaining high compressive stiffness in the normal to the motion but in line with the load. Recent testing of porous carbon media for over-load conditions show superb ability to sustain high loads during solid dry friction contact while at high rotor speed [5,9]. Although porous carbon bearings are utilized in several application today, they still have yet to penetrate the turbomachinery industry. A natural evolution would involve the integration of porous carbon with a GEN3 type compliant hybrid bearing to address the risk of bearing touch down at high speeds and loads.

FIGURE 8. Porous Carbon Gas Bearing

CONCLUSION

The following paper discussed the evolution of gas bearing technology keenly focused on enabling the oil-free operation of high-power turbomachinery. With the recent advancement in load capacity and damping, compliant hybrid gas bearings show great promise in achieving the oil-free turbomachinery milestone. Next step in the development involve developing reliable coatings or wear-couples that allow for the bearing system to sustain rubs at high speeds and loads. Porous carbon bearing systems are potential candidates for this next step in the evolution of gas bearings and demand further research to mature the technology for high-power turbomachinery.

ACKNOWLEDGEMENTS

Thanks to Newway air bearings for graciously allowing the use of example pictures of porous carbon bearings.

REFERENCES

[1] Agrawal, G., 1990, “Foil Gas Bearings for Turbomachinery,” SAE Technical Paper No. 901236.

[2] Lubell, D. R., Wade, J. L., Chauhan, N. S., and Nourse, J. G., 2008, “Identification and Correction of Rotor Instability in an Oil-Free Gas Turbine,” ASME Paper No. GT2008-50305.

[3] Magge, N., 1975, “Philosophy, Design, and Evaluation of Soft-Mounted Engine Rotor Systems,” J. Aircr., **12**(4), pp. 318–324.

[4] Ertas, B. H., 2008, “Compliant Hybrid Journal Bearings Using Integral Wire Mesh Dampers,” ASME J. Eng. Gas Turbines Power, 131(2), p. 022503.

[5] Ertas, B., and Delgado, A., 2018, “Compliant Hybrid Gas Bearing Using Modular Hermetically Sealed Squeeze Film Dampers,” ASME J. Eng. Gas Turbines Power, 141(2), p. 022504.

[6] Mook, T. J., Ertas, B.H., Bellardi, J. J., “Bearing”, US Patent No. 9746029, Aug. 29, 2017.

[7] Ertas, B.H, Mook, J. T., Bellardi, J. J., “Additive Thrust Gas Bearing”, US Patent No. 10036279, July 31st, 2018.

[8] Ertas, B.H., 2019, “Compliant Hybrid Gas Bearing Using Integral Hermetically-Sealed Squeeze Film Dampers”, Proc. Gas Turbine Tech. Conf. Exp. ASME Turbo Expo, Phoenix, AZ, June 17-21, Paper No. GT2019-90865.

[9] San Andres, L., Cable, A., T., Zheng, Y., De Santiago, O., and Devitt, D., 2016, “Assessment of Porous Type Gas Bearings: Measurements of Bearing Performance and Rotor Vibrations,” ASME Paper No. GT2016-57876.