The 5th International Supercritical CO_2 Power Cycles Symposium
March 29-31, 2016
San Antonio, TX

Application of Hydrostatic Bearings in Supercritical CO_2 Turbomachinery

Preuss, Jason L.
Barber-Nichols Inc.
Arvada, CO 80002
jpreuss@barber-nichols.com

Mr. Preuss is a project engineer at Barber-Nichols overseeing design and manufacture of specialty turbomachinery for aerospace and power generation applications. He received his B.S. (2001) and M.S. (2004) degrees in Mechanical Engineering from Texas A&M University. Prior to joining Barber-Nichols (2007), Mr. Preuss held positions at Honeywell (Torrance) performing aerospace turbomachinery design and analysis.

Abstract

Bearing selection and performance are critical for efficient and reliable operation of supercritical Carbon Dioxide (sCO_2) turbomachinery. Hydrostatic bearings, also known as externally pressurized bearings, offer unique benefits for these machines. The sCO_2 process fluid can be utilized as the bearing supply, making a hermetic machine design feasible. In addition, hydrostatic sCO_2 bearing performance can exceed oil or sCO_2 lubricated hydrodynamic bearings in regards to load capacity, stiffness, and damping. Non-contacting surfaces and relatively large film thickness can improve reliability and extend maintenance intervals. sCO_2 machinery can tolerate the high supply pressures and parasitic supply flow that typically limit application of hydrostatic bearings. Guidelines are presented to aid in sizing hydrostatic journal and thrust bearings during the conceptual design phase. These include rule of thumb estimates for load capacity, leakage rates, and dynamic coefficients. Limitations due to fluid compressibility and rotordynamic instability are discussed along with design features that can be used to mitigate these limitations. The guidelines presented are summarized with examples of hydrostatic bearing sizing and performance estimates.

Introduction

Supercritical Carbon Dioxide (sCO_2) power cycles offer potential economic benefits and higher power densities compared to traditional plants. The optimization of turbomachinery for these sCO_2 cycles can yield demanding requirements for the bearings. These include high bearing DN values, large loads, and the use of sCO_2 as the bearing lubricant. These factors combined with industrial life requirements favor selection of non-contacting fluid film type bearings. These bearings are generally subdivided as hydrodynamic or hydrostatic based on how load-supporting pressure is generated. sCO_2 hydrostatic bearings are presented here as an alternative to traditional hydrodynamic bearings.

Hydrodynamic bearings rely on the shaft rotation to draw lubricant into a wedge that generates a load supporting pressure on the shaft. Load capacity is roughly proportional to viscosity and speed for a given geometry and clearance. Alternatively, hydrostatic bearings direct an external pressure supply to the shaft removing the need for shaft rotation or a viscous lubricant. The load supporting pressure results from changes in flow restriction as shaft loads modify the bearing clearance. Therefore, hydrostatic bearings are well suited to effectively operate using low viscosity sCO_2. Hermetic turbomachinery designs are therefore feasible and eliminate complexity of sCO_2 shaft seals. The performance of sCO2 hydrostatic bearings exceeds most bearing types in load capacity, stiffness, and damping. Their principal disadvantage is the cost, complexity, and inefficiency of using an external supply. However, the robustness offered by these bearings can justify these tradeoffs in sCO_2 MW class turbomachinery. The following sections provide information to aid in trade studies and their application.

Bearing Supply and Discharge

The external supply should be identified early as it has the most influence on performance and bearing design. The low temperature discharge of the cycle feed pump or compressor is most desirable as it provides maximum pressure differential and the most incompressible fluid. Fluid compressibility can significantly reduce bearing performance and lead to a self-excited instability analogous to pneumatic hammer. The extent depends on the fluid bulk modulus, bearing pressure differentials, and bearing flow volumes. A typical hydrostatic bearing design can be utilized when the average bulk modulus is an order of magnitude larger than bearing pressure differentials. This constraint is met for sCO_2 bearings when the discharge pressure is regulated to a supercritical pressure. This pressure regulation provides an additional benefit in vapor cycles by preventing two phase flow in the bearing that might lead to cavitation erosion. For marginally supercritical conditions, the bearing recess volumes should be minimized to mitigate compressibility effects. Recess depths that are approximately ten times the minimum film clearance are recommended. Bearings based on purely gaseous sCO_2 supply are feasible but result in significantly reduced performance. These bearings are typically referred to as Aerostatic bearings with compressibility effects mitigated by elimination of bearing recesses. The guidelines in the following sections are limited to hydrostatic types.

Thrust Bearings

Assuming a sCO_2 supply as described above, the simple orifice fed, rectangular recessed type is recommended and presented here for both thrust and journal bearings. It is cost effective, robust, and provides a majority of the achievable performance for a given fluid supply. Figure 1 depicts a typical layout for annular multi-recess thrust bearings.

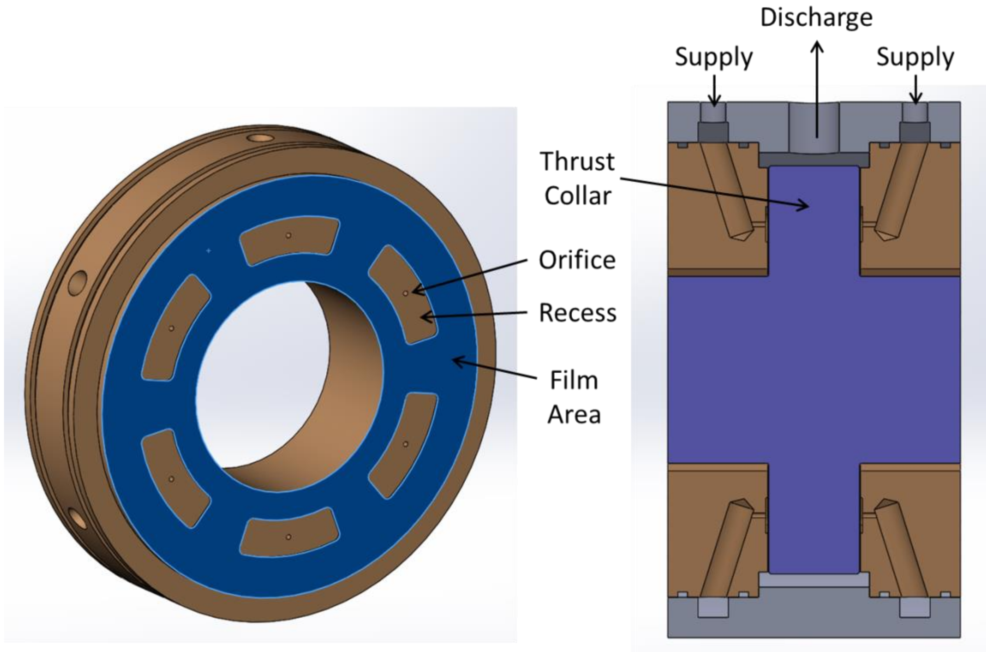

Figure 1: Annular Recess Hydrostatic Thrust Bearing

A key characteristic of hydrostatic thrust bearings is flow restriction vs load. At large clearances the orifice dominates, limits flow, and the recess pressure approaches the discharge. As thrust load increases, the face clearance is reduced and provides additional restriction. This restriction reduces the pressure drop in the orifice and the resulting increase in recess pressure provides load capacity. The use of a constant supply flow (positive displacement pump or control valve) in lieu of orifices results in a similar effect.

Load capacity and power loss are primary parameters of interest for thrust bearing sizing and trades. The power loss is comprised of the leakage flow pumping power and thrust runner windage. An approximate thrust bearing load capacity estimate is calculated as follows.

$$Thrust\ Load\ Capacity = \frac{1}{3} \times Pressure\ Differential \times Bearing\ Area$$
$$= \frac{\pi}{12}(P_s - P_o)(D_0^2 - D_i^2)$$

Pumping power is calculated using the pressure differential determined above and bearing leakage flow. The leakage flow depends on many parameters and generally the result of detailed analysis using commercial codes or listed references. The fluid conditions and geometry are constrained here to simplify the leakage estimate. These constraints are Do/D_i = 2, 100F supply temperature, 1200 psia discharge pressure, .001" minimum film thickness, 6 recesses with one orifice each, and optimized recess location for maximum load capacity. A range of leakage flow can be estimated using a compressible flow orifice calculation. The discharge coefficient typically ranges from .65 to .8 and a value of .7 is suggested. The total orifice area is estimated as follows based on detailed designs with D_o ranging from 4 to 8 inches and the constraints listed previously.

$$A_O = .0014D_o + .0086\ [in\char`^2]$$

The range of possible leakage flows corresponds to the minimum and maximum recess pressure. The maximum recess pressure is equal to the discharge pressure when the clearance is large at low loads. The minimum recess pressure corresponds to a recess pressure ratio of .7 that is defined as follows.

$$R_{PR} = \frac{P_r - P_o}{P_s - P_o}$$

Generally, a double acting thrust bearing will be implemented with total flow estimated as a summation of the minimum and maximum leakages.

The windage power loss can be estimated using the equation below reported by reference 3.

$$W = \frac{k}{1.44E11} \rho_d N^3 {D_o}^4 (D_o + 5B)$$

where k is an empirical friction type coefficient that is calculated as:

$$k = 18.23Re^{-.644} + .004 \text{ and } Re = \ \pi \rho_d {D_o}^2\ N/(60\mu_d)$$

This approach corresponds to windage of a disk surrounded by a fixed clearance. A clearance to diameter ratio of .1 is built into the equations. The windage reduces with decreasing face clearances but the effect is minor with sCO_2. The use of discharge clearance and density are recommended for use in the equations above. The thrust collar width must be sufficient to keep the deflection under load small compared to the minimum film clearance. A minimum value of 1/4 inch per inch of diameter is generally sufficient for a diameter ratio $D_o/D_i = 2$.

Figure 2 summarizes example outputs for 4, 6, and 8 inch sCO_2 hydrostatic thrust bearings. The results are in-line with predictions by XLHydrothrust, a computational analysis code available from Texas A&M. XLHydrothrust predictions have been validated by measurements described in reference 7. The results in Figure 2 support the general conclusion that large load capacities are feasible but limited by leakage flows and windage. These leakage flows are generally prohibitive for smaller scale machines and become tolerable in the megawatt class. The design driver for the leakage is selection of a minimum film thickness based on alignment capabilities. The flow is particularly sensitive to this and varies by the cube of the film thickness. A value of .001 inches is typical in oil lubricated bearings and assumed in estimates presented here. Windage is significant due to the relatively high density of sCO_2. It roughly scales with speed cubed and diameter to the fifth power. Thrust bearing sizing is typically a result of increasing the bearing diameter until windage losses become excessive.

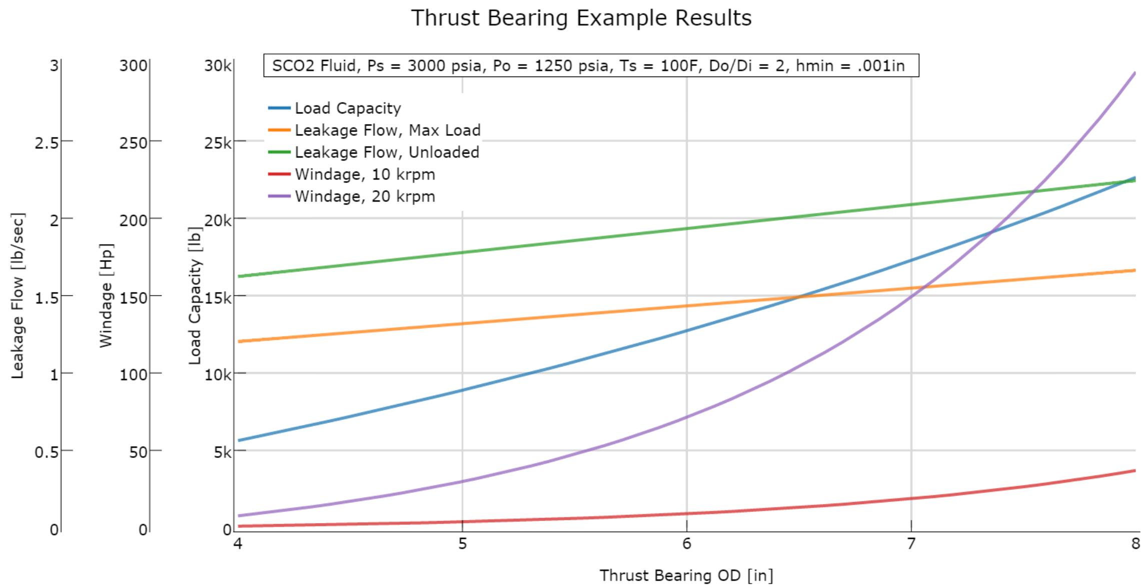

Figure 2: Thrust Bearing Example

Journal Bearings

Hydrostatic journal bearings share the same operating principles described above for thrust bearings. Figure 3 depicts a typical layout for the multi-recess hydrostatic journal bearing.

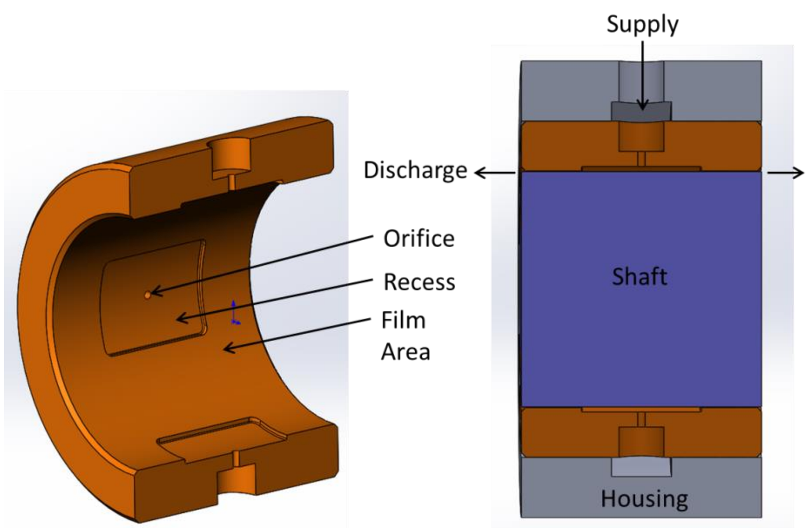

Figure 3: Hydrostatic Journal Bearing

The key parameters of interest are radial load capacity, leakage, pneumatic hammer, and rotordynamic performance. The load capacity estimation procedure is similar to that for thrust bearings and presented below.

$$Radial\ Load\ Capacity = \frac{1}{3} \times Pressure\ Differential \times Projected\ Area = \frac{1}{3}(P_s - P_o)(D_j L)$$

Similar to the thrust bearing above, a compressible flow orifice calculation can be used to estimate leakage with modification to the orifice area and recess pressures as described below. The fluid conditions and geometry are again constrained to simplify the orifice area relation. The constraints include use of SCO2 fluid, P_o = 1250 psia, L/D_j = 1, 4 pads, and optimized recess geometry. Bearing clearance is the design driver for leakage flow. Similar to oil lubricated bearings, a value of .002 inches per inch of journal diameter is recommended. Orifices are then sized to achieve 50% of the pressure drop across the orifice for a centered unloaded rotor. The total orifice area is estimated per the equation below based on the prior listed constraints.

$$A_0 = .016D_j - .019$$

Prediction of bearing dynamic coefficients is necessary to determine if rotordynamic stability margins are acceptable. The viscosity and density of SCO2 are large enough that significant hydrodynamic forces are generated. This offers the advantage of hybrid type performance where hydrodynamic effects are additive and load capacity is increased over hydrostatic operation alone. The disadvantage is that sCO_2 hydrostatic journal bearings suffer the same stability limitations as plain oil lubricated journal bearings. A whirl frequency ratio of .5 is typical which translates to a limit speed of twice the first critical speed. Therefore, bearing design will tend towards maximizing stiffness and the resulting first critical speed. A significant stability improvement is suggested in reference 4 by use of angled orifice injection. The angled injection reduces cross coupling forces by directing flow against shaft rotation as depicted in Figure 4.

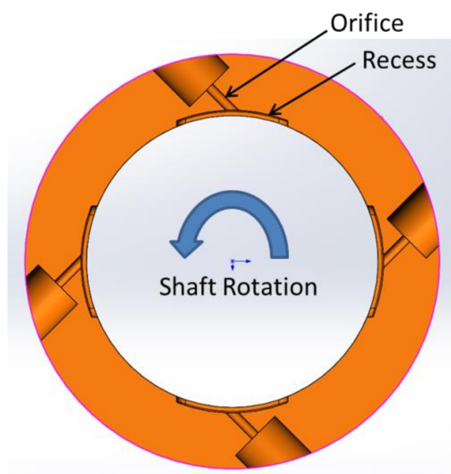

Figure 4: Cross Section of Hydrostatic Journal With Angled Orifice Injection

Fluid compressibility is another potential limitation for rotordynamic performance. Compressibility can significantly reduce damping and lead to a self-excited instability known as pneumatic hammer (Reference 5). This instability is different from rotor instability driven by cross coupled forces. The problem becomes more prevalent at low values of bulk modulus, high excitation frequencies (shaft speed), high bearing pressure differentials, and large recess volumes. Reference 5 offers a design parameter to gauge severity of compressibility based on these factors. Similar to thrust bearings, regulating the discharge above the critical pressure will help mitigate the problem with higher values of fluid bulk modulus. Additional mitigation includes minimization of the recess volume using recess depths approximately 10 times the bearing radial clearance.

Example load capacity and leakage results are presented in Figure 5. XLHydrojet, a computational analysis code available from Texas A&M, was utilized to generate the example rotordynamic coefficients for two, three, and four inch bearings as depicted in Figures 6 and 7. XLHydrojet predictions have been validated by measurements described in reference 4. The results are based on constraints listed in the figures and no angled injection. The values can be utilized in rotordynamic models to determine if sufficient critical speed and stability margins are feasible. If operation below the first critical is not feasible, sufficient rigidity in the shaft will typically result in heavily damped modes for which zero critical speed margin can be considered. Stability limitations discussed prior should also be considered for operation above the first critical.

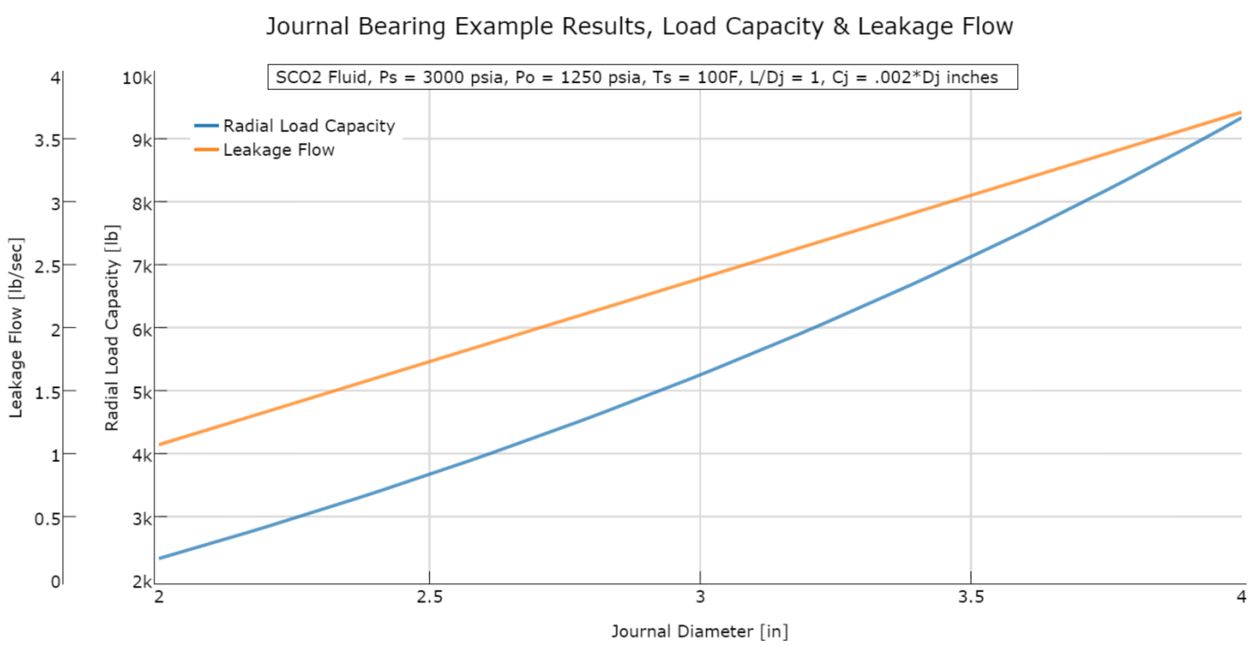

Figure 5: Journal Bearing Load Capacity & Leakage Flow vs Size

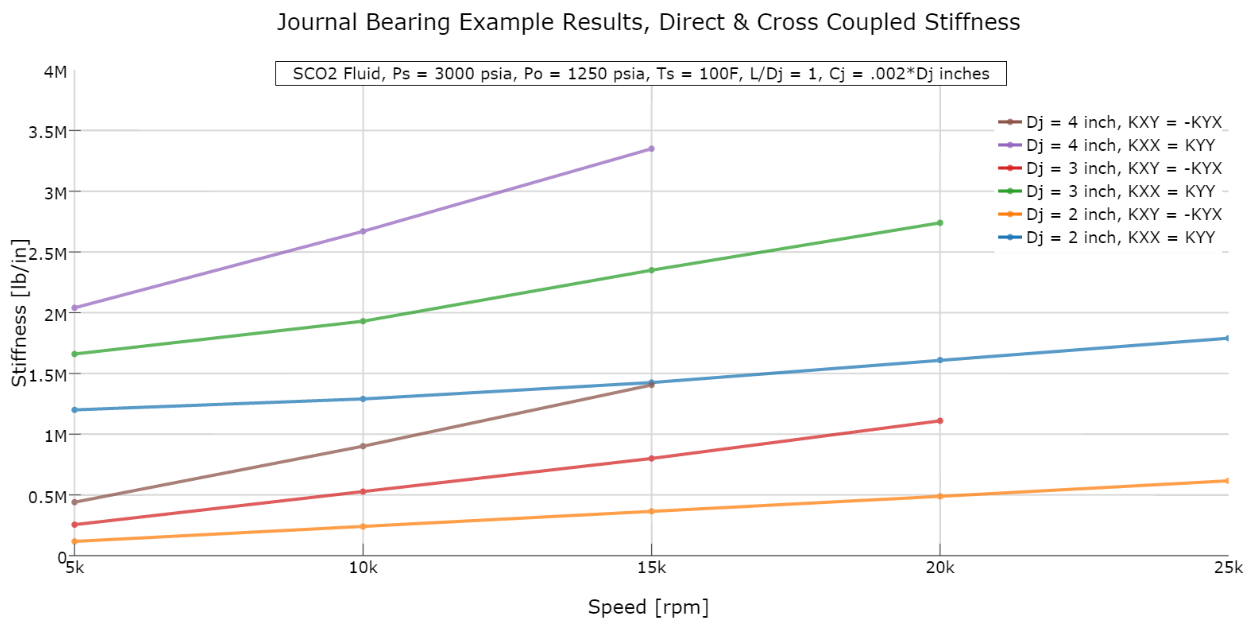

Figure 6: Journal Bearing Direct & Cross Coupled Stiffness

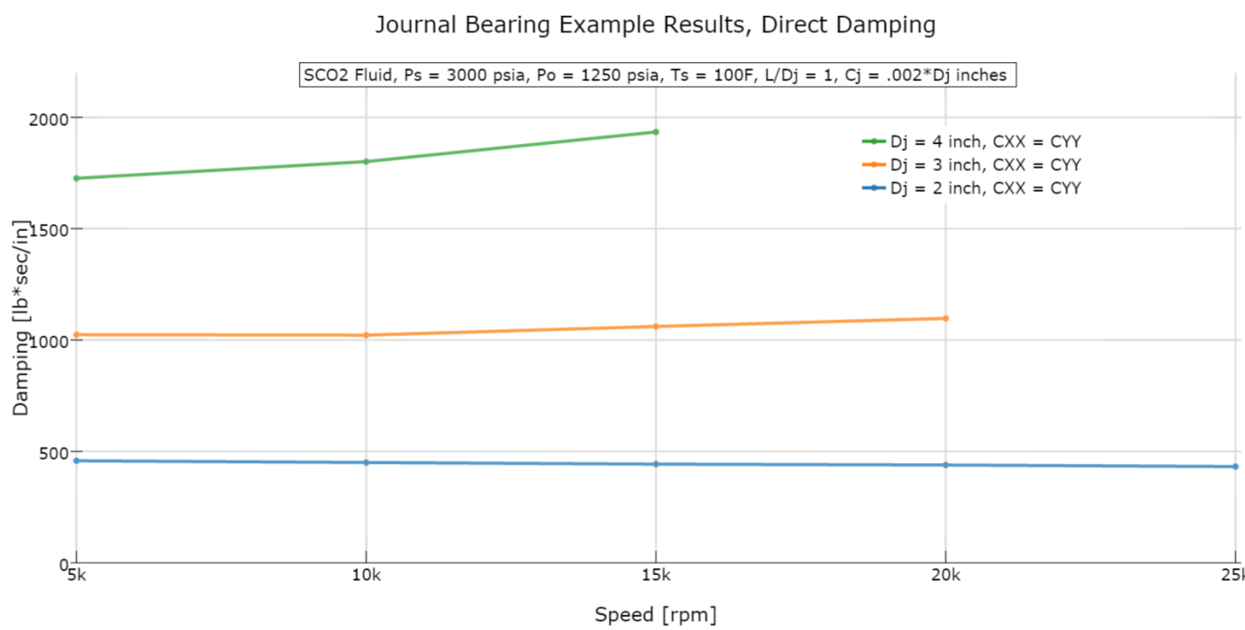

Figure7: Journal Bearing Direct Damping

Conclusion

sCO_2 hydrostatic bearings offer large load capacities and are suitable for consideration in MW class machines where their parasitic losses may be tolerable. In addition, journal bearings offer very high stiffness and damping values that result in excellent rotordynamic performance. Thrust bearing size will generally be limited by windage losses that are a strong function of diameter (D^5). Journal bearing design and sizing will generally be driven by rotordynamic considerations. In particular, sCO_2 viscosity is significant enough that whirl frequency ratio limitations exist similar to plain hydrodynamic bearings (whip). Supply and discharge conditions should be in the supercritical region to avoid detrimental effects of fluid compressibility. The guidelines presented here are intended to help the reader determine feasibility, balance design trades, and proceed to detailed design.

Nomenclature

$N = Rotational\ Speed\ [rpm]$
$P_s = Supply\ Pressure\ [psia]$
$P_r = Recess\ Pressure\ [psia]$
$P_o = Discharge\ Pressue\ [psia]$
$T_s = Supply\ Temperature\ [°F]$
$\rho_d = Density\ at\ Bearing\ Discharge\ [lb/ft\text{^}3]$
μ_d = Viscosity at Bearing Discharge [lbm/in*sec]
$D_o = Thrust\ Runner\ OD\ [in]$
$D_i = Thrust\ Runner\ ID$ [in]
$D_j = Journal\ Bearing\ Diameter$ [in]
$C_j = Journal\ Diametral\ Clearance$ [in]
$B = Thrust\ Runner\ Width\ [in]$
$L = Journal\ Bearing\ Length\ [in]$
$C_d = Orfice\ Discharge\ Coefficient\ [Dimensionles]$
$h_o = Minimum\ Film\ Clearance\ [in]$
$R_{PR} = Recess\ Pressure\ Ratio\ [-]$
$W = Windage\ power\ loss\ [hp]$
$KXX, KYY = Direct\ Stiffness\ [lb/in]$
$KXY, KYX = Cross\ Coupled\ Stiffness\ [lb/in]$
$CXX, CYY = Direct\ Damping\ [lb * sec/in]$

References

[1] Rowe, W. B. Hydrostatic, Aerostatic, and Hybrid Bearing Design. Amsterdam: Elsevier, 2012

[2] Khonsari, Michael M., and E. Richard. Booser. Applied Tribology: Bearing Design and Lubrication. 2nd ed. New York: John Wiley, 2008

[3] Hamm, H. W. "Determining Fluid Friction or Windage of Rotating Discs", Allis-Chalmers Electrical Review, Fourth Quarter, 1962

[4] San Andrés, L., and D. Childs, "Angled Injection - Hydrostatic Bearings, Analysis and Comparison to Test Results", ASME Journal of Tribology, (1997), 119, 1, pp. 179-187.

[5] San Andrés, L., "Fluid Compressibility Effects on the Dynamic Response of Hydrostatic Journal Bearings", L. San Andrés, WEAR, (1991), 146, pp. 269-283

[6] San Andrés, L., 2010, *Modern Lubrication Theory*, "Hydrostatic Journal Bearings," Notes 12b, Texas A&M Digital Libraries, http://oaktrust.library.tamu.edu/handle/1969.1/93197, (11 Mar. 2016)

[7] San Andrés, L., Phillips, S., and Childs, D., (2016), "A Water Lubricated Hybrid Thrust Bearing: Measurements and Predictions of Static Load Performance", ASME Turbo-Expo Conference, Seoul, June 13-17, ASME Paper No. GT2016-56349