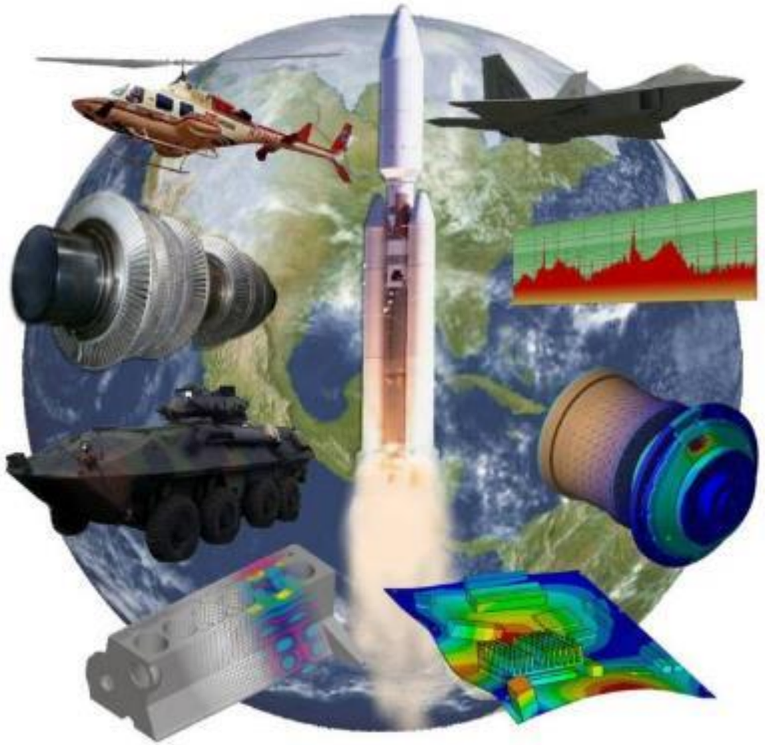
Advanced Gas Foil Bearing Design for Supercritical CO_2 Power Cycles

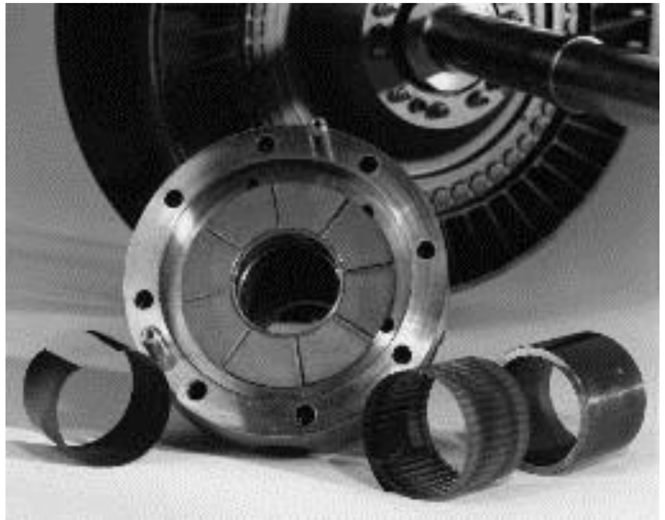


Peter A. Chapman, Jr., P.E.
Principal Engineer
Mechanical Solutions, Inc.
pac@mechsol.com
Albany, NY
Whippany, NJ
Denver, CO
www.mechsol.com
973-326-9920

The 6th International Symposium
Supercritical CO_2 Power Cycles
March 27-29, 2018
Pittsburgh, PA

Today's Presentation

- Project Background
- Overview of Hybrid Bearing Designs
  - Simple Concept
  - Radial Bearing
  - Thrust Bearing
- Analytical Approach
  - SCO2 Properties Evaluation
  - CFD Analysis
  - Structural Analysis
- Design Optimization
- Derivation of Bearing Coefficients
- Future Work



Project Background

- Funding provided by the Department of Energy (DOE) Office of Fossil Energy
- Goal: develop a reliable, high performance foil bearing system using sCO_2 as the working fluid
  - Temperatures up to 800°C
  - Pressures up to 300 bar
- Key elements of the design:
  - An advanced hydrostatically-assisted hydrodynamic (or hybrid) foil bearing with higher load capacity
  - An integral gas delivery system to distribute flow throughout the bearing
  - Addition of overload protection to handle large shaft excursions during severe system transients
  - Use of high temperature materials and coatings to prolong life and enabling sufficient start/stop cycles

Why Foil Bearings?

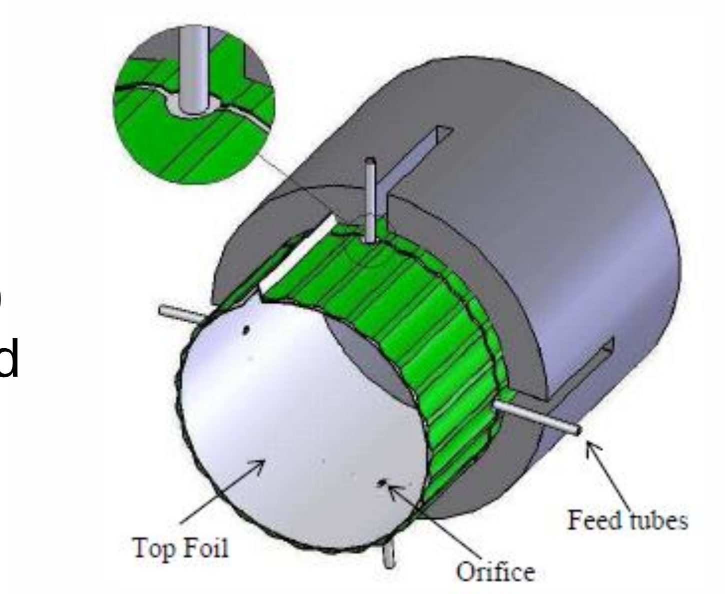
- High speed capability
- Extreme-temperature and/or oil-free environment
- Permits a hermetically-sealed system (eliminate end seals)
- Insensitive to system pressure
- Applicable to high energy density turbomachinery
  - Motors and generators are being designed to run faster and with more torque, with reduced size & weight
  - Direct drive is a trend
- Long, maintenance-free life

Traditional Foil Bearing Drawbacks

- Low load capacity
- Require thermal management (cooling)
- Relatively low direct stiffness
- Low damping (but low cross-coupling also)
- Difficult to quantify rotordynamic coefficients analytically
- Intolerant of low frequency overloads
- Rubbing wear during start-up/shut-down

Hybrid Bearing Concept

- Hydrodynamic load capacity often limits gas foil bearing use in some equipment, particularly larger machines running at lower speeds
- Supplementing load capacity and stiffness could enable broader use of gas foil bearings
- Adding a hydrostatic component is one method of enhancing a gas foil bearing
- Pressurized gas is injected directly into the bearing gap
- Evaluation of a simple orifice design (as shown on right) did not generate a significant amount of pressure around a large enough benefit
- Minimal force benefit gained, potential instability at high eccentricities



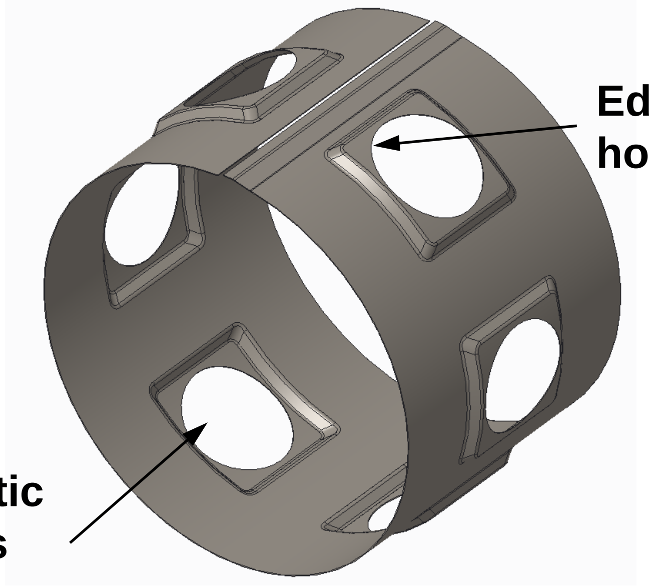
Source: Texas A&M University (Kumar[1])

---

1. Kumar, M., "Analytical and Experimental Investigation of Hybrid Air Foil Bearings," A Thesis submitted to the Office of Graduate Studies of Texas A&M University, August 2008.

Enhanced Hydrostatic Design

- To enhance the hydrostatic benefit, an array of discrete pockets were added to the top foil
- The working fluid (sCO_2) is supplied to each pocket through an orifice
- The pockets provide larger pressure areas to be created
- Significantly larger hydrostatic force can be generated

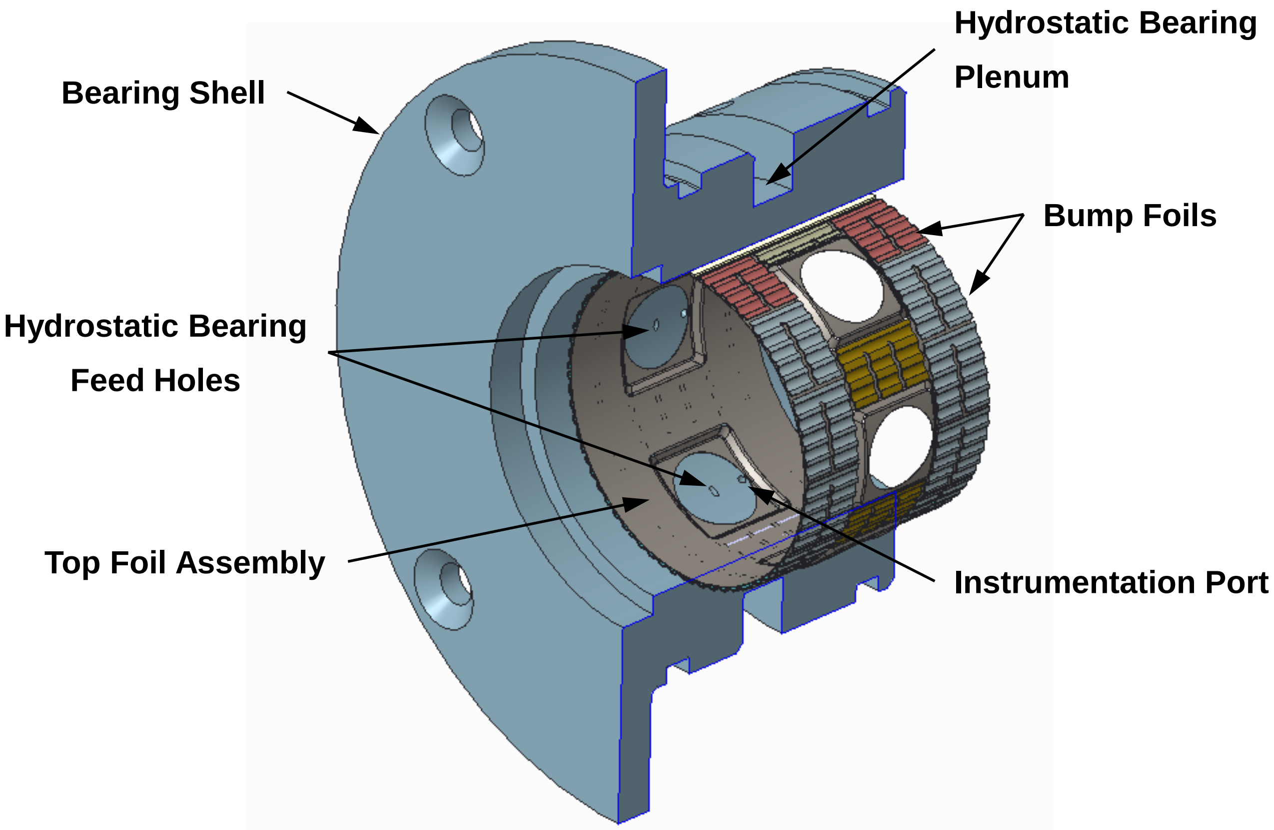


Edge is laser welded to housing to provide seal

Hydrostatic Pockets (qty. 6)

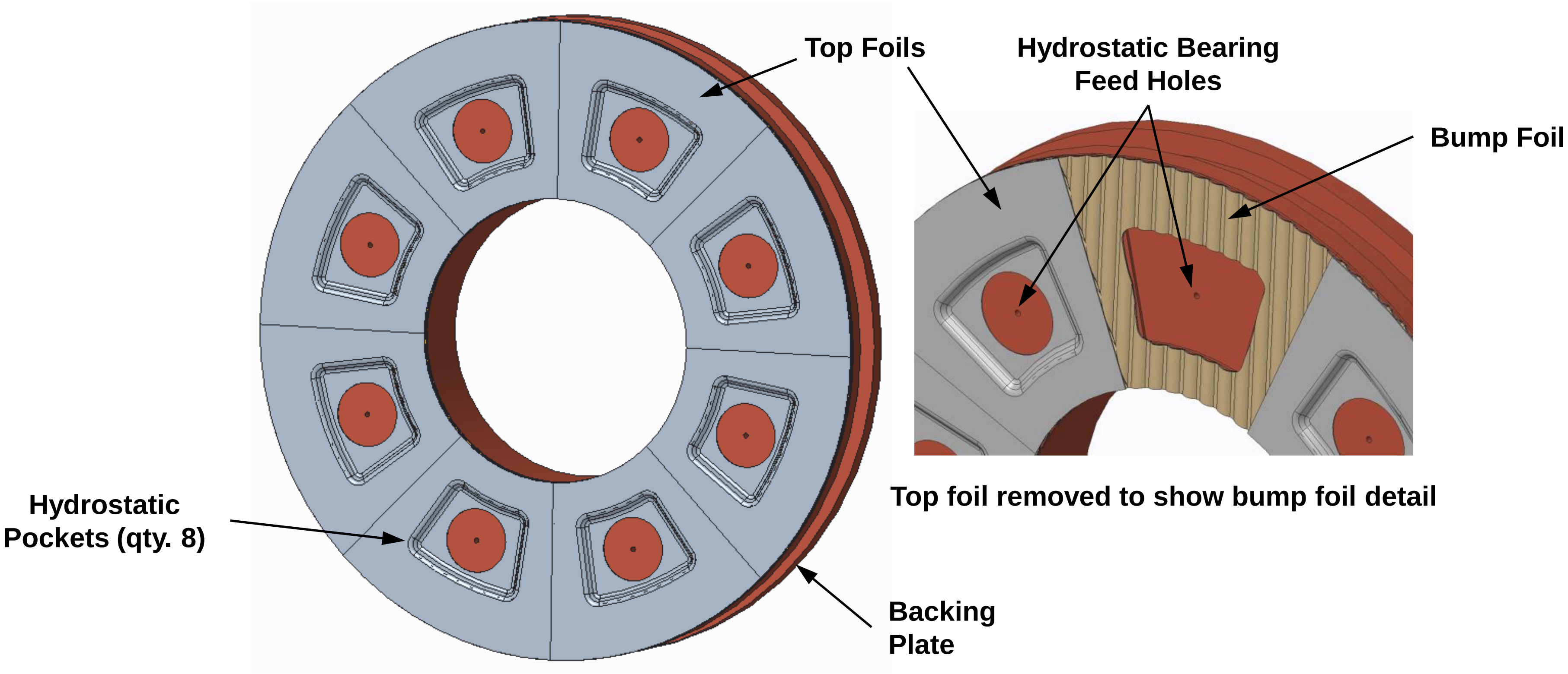
Radial Bearing Design

- Radial bearing consists of:
  - A top foil containing the hydrostatic pockets
  - A multi-layered array of bump foils
  - A bearing shell
- An annular plenum supplies each pocket through an orifice



Thrust Bearing Design

- Similar to the radial bearing, the thrust bearing consists of:
  - A top foil containing the hydrostatic pockets
  - A multi-layered array of bump foils
  - A backing plate
- An annular plenum supplies each pocket through an orifice



Top foil removed to show bump foil detail

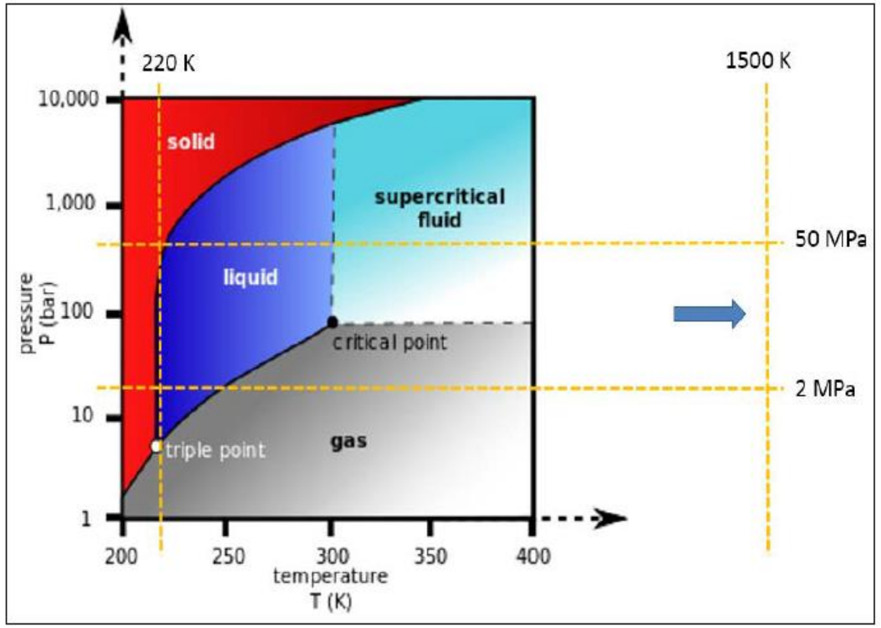
Analytical Approach

- A series of computational models were developed
- The following performance characteristics were sought
    - Load capacity
    - Direct stiffness
    - Cross-coupling stiffness
    - Damping coefficients (both direct and cross-coupling)
- The following steps were carried out in developing the modeling approach
    - Characterization of the sCO_2 fluid properties, particularly around the critical point
    - Optimization of the hydrostatic bearing geometry
    - Superimposition of the hydrodynamic effect on the model, including synergistic effects
    - Addition of the compliant foil sub-structure interaction

SCO_2 Properties Evaluation

- Through non-project funding, MSI acquired detailed real gas properties (RGP) tables for CO_2
  - Pressure range: 2 to 50 MPa
  - Temperature range: 200 K (-53°C) to 1500 K (1227°C)

- The tables support the following states:
  - liquid
  - vapor
  - supercritical
  - metastable states near saturation
    - superheated liquid
    - subcooled vapor



- Metastable states occur when the vapor cools below the local saturation temperature due to rapid expansion (likely at the hydrostatic nozzle)

CFD and Structural Analysis Studies

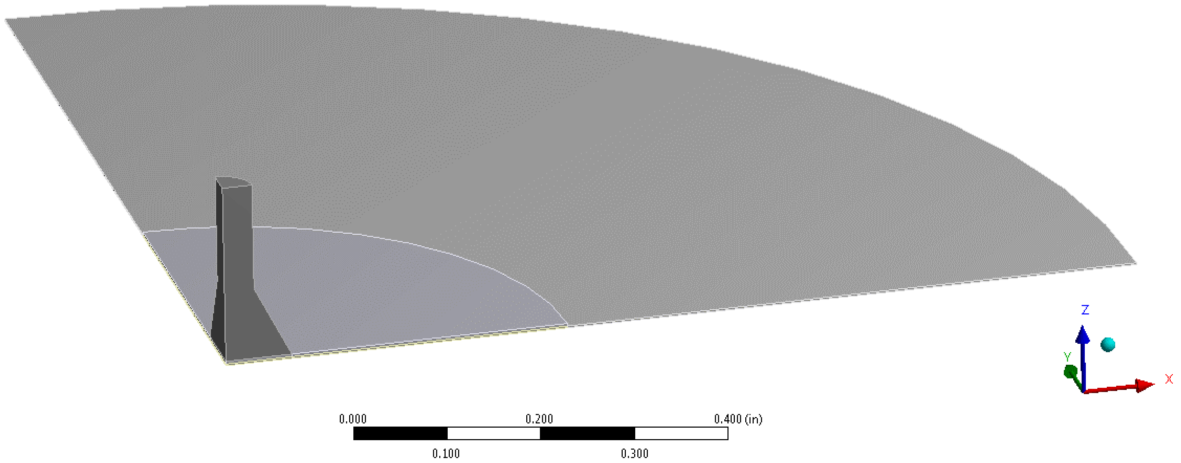
- A series of CFD and structural FEA studies were completed
- The intent was to understand the flow characteristics within the bearing and the variables that control them
- Four primary steps of the study consisted of:
    - Optimization of the hydrostatic nozzle and pocket through simplified geometry
    - Generation of a full 3-D model of the bearing geometry
    - Transfer of the CFD-generated pressures to the structural model to determine stresses and deflections
    - Iteration of the CFD model and deformed structural model to obtain a converged solution

Optimization of Nozzle and Pocket Geometries

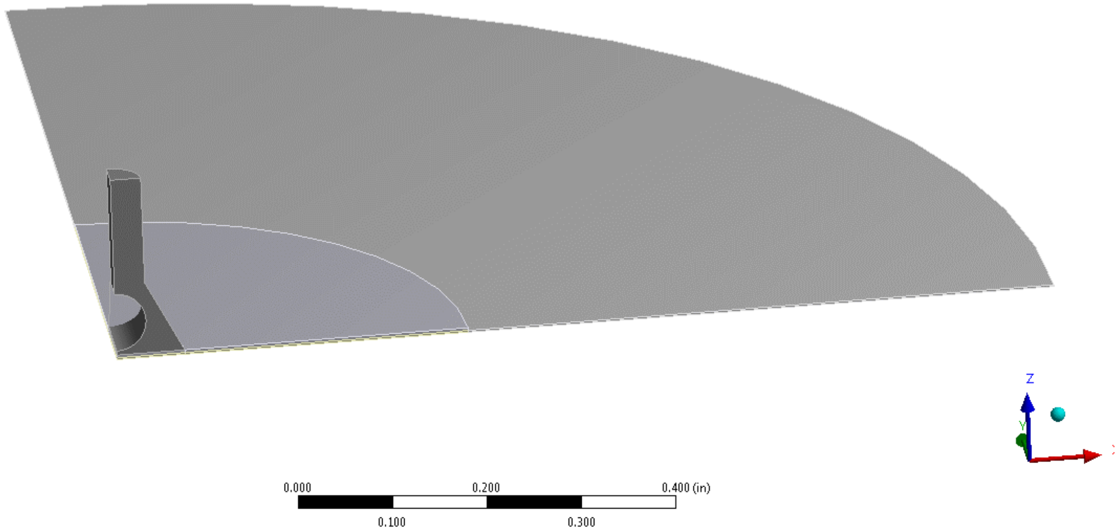
- CFD sector models were used to evaluate the effect of geometry on hydrostatic bearing performance
  - nozzle diameter
  - bearing hydrodynamic nominal clearance
  - pocket size
  - pocket depth
  - different diffuser geometries
- The goal of the study was to find the optimum geometry that would:
  - maximize static pressure differential between opposing pockets in the direction of load (thereby maximizing the force)
  - minimize the flow rate through the bearing (thereby maximizing system efficiency)

Optimization of Nozzle and Pocket Geometries

- A 90-degree sector model was used to evaluate different nozzle and diffuser geometries
  - Initial results indicated the jetting force on the shaft was high



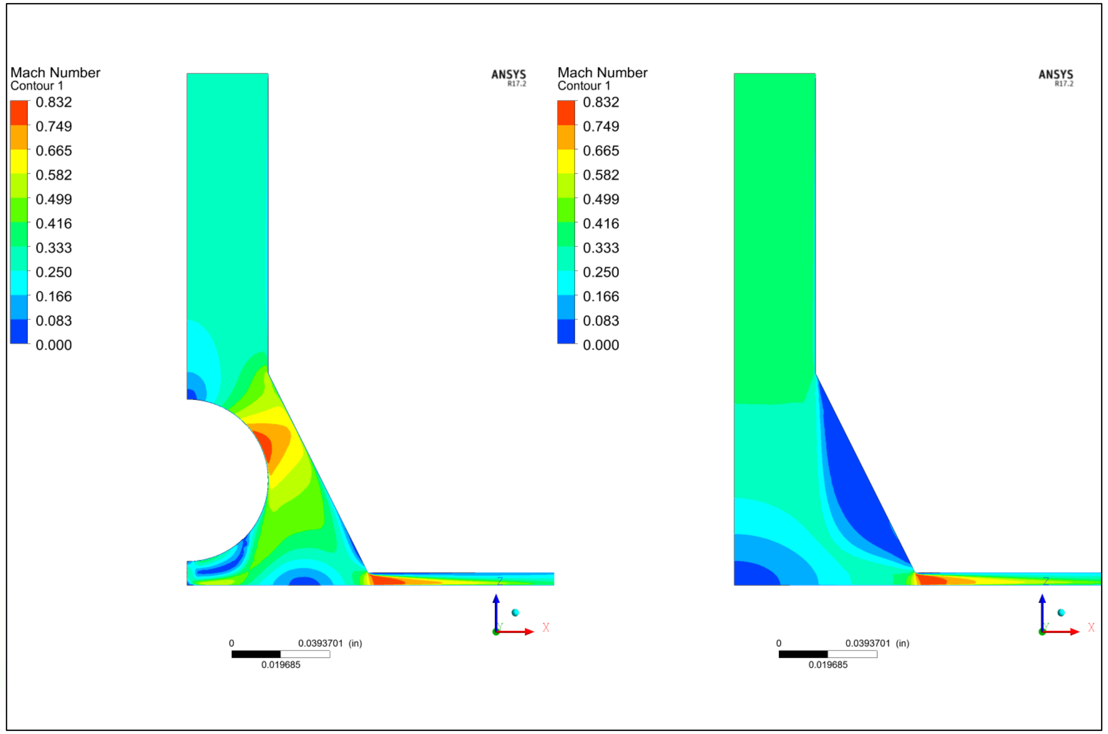
Simple Conical Diffuser



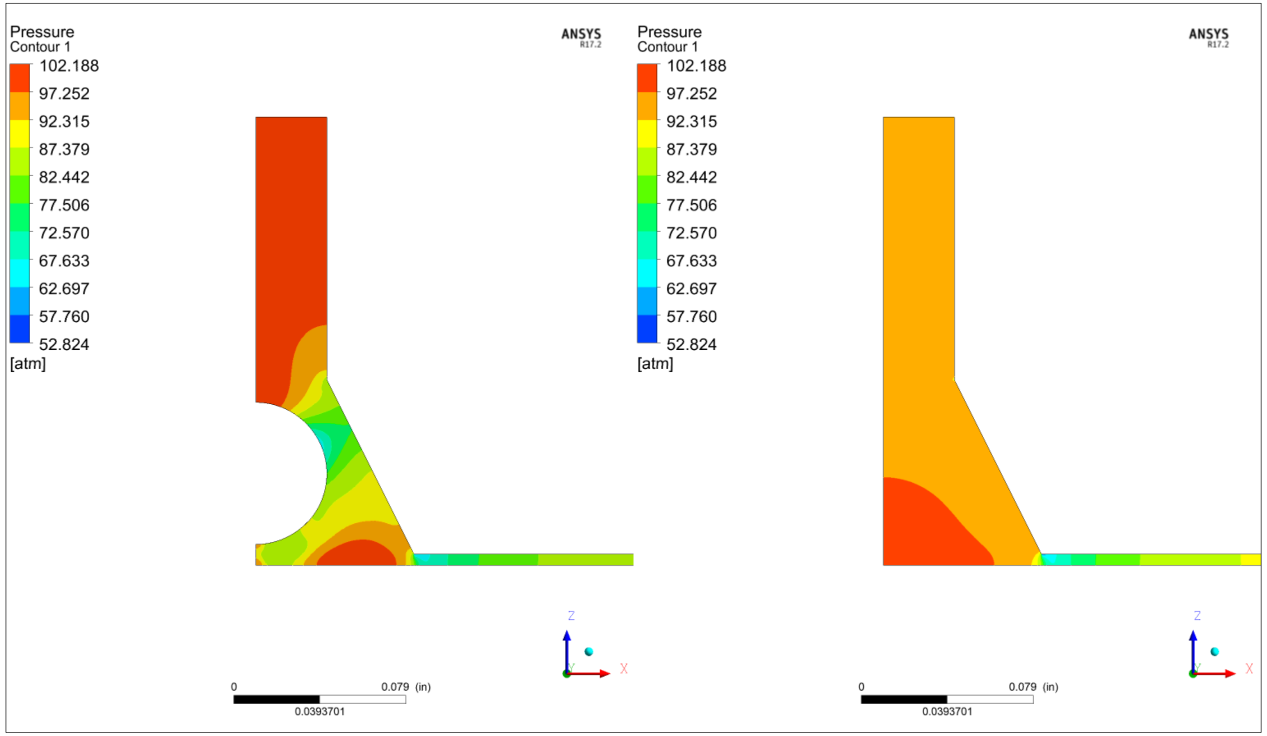
Conical Diffuser with a Small Obstruction

(Intended to reduce jet impingement on the shaft)

Optimization of Nozzle and Pocket Geometries



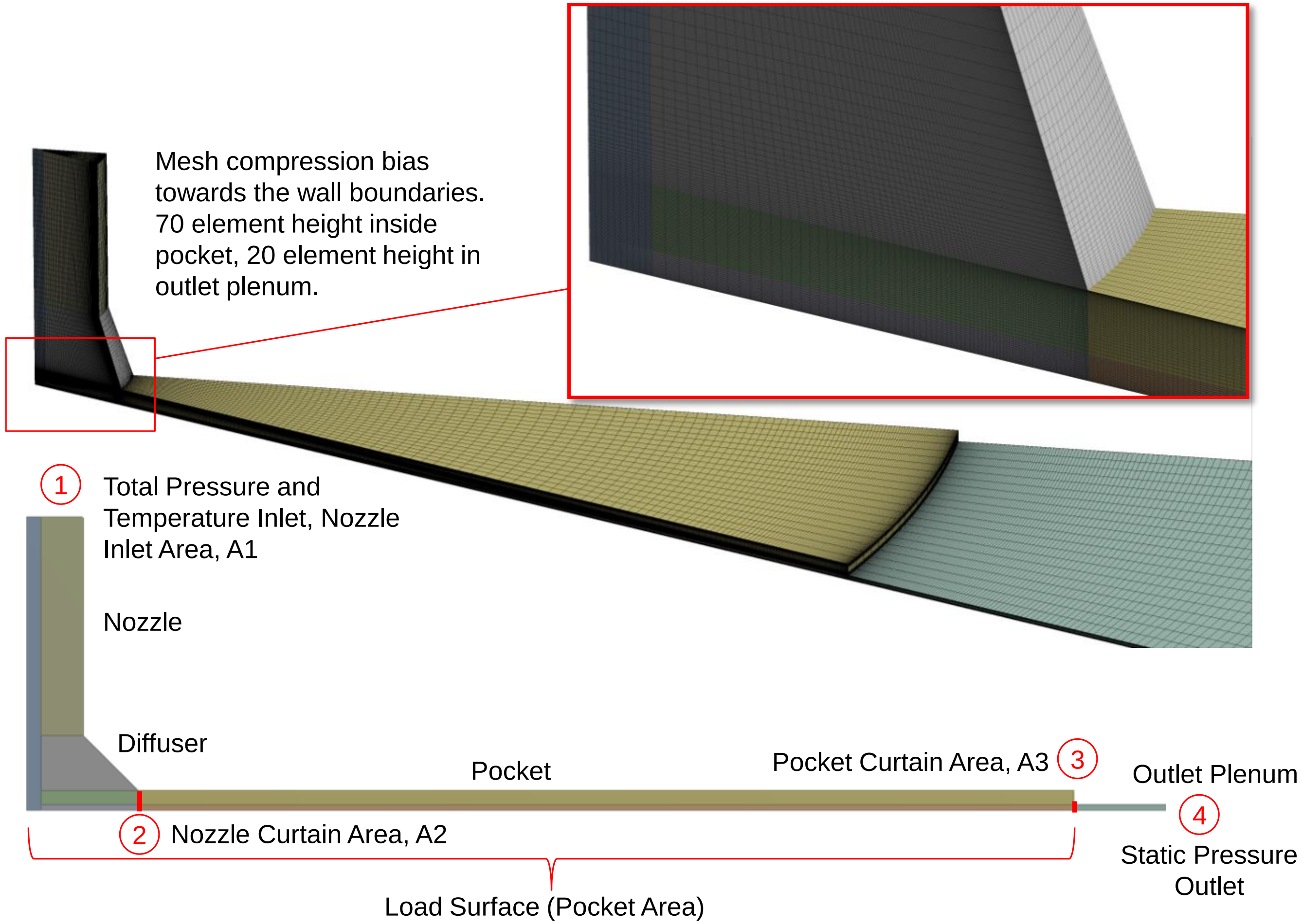
Velocity Profile (Mach Number)



Static Pressure Distribution

Optimization of Nozzle and Pocket Geometries

- A 20-degree sector model was used to reduce solver time so multiple configurations could be evaluated quickly



Mesh compression bias towards the wall boundaries. 70 element height inside pocket, 20 element height in outlet plenum.

① Total Pressure and Temperature Inlet, Nozzle Inlet Area, A1

Nozzle

Diffuser

Pocket

Pocket Curtain Area, A3 ③

Outlet Plenum

④ Static Pressure Outlet

② Nozzle Curtain Area, A2

Load Surface (Pocket Area)

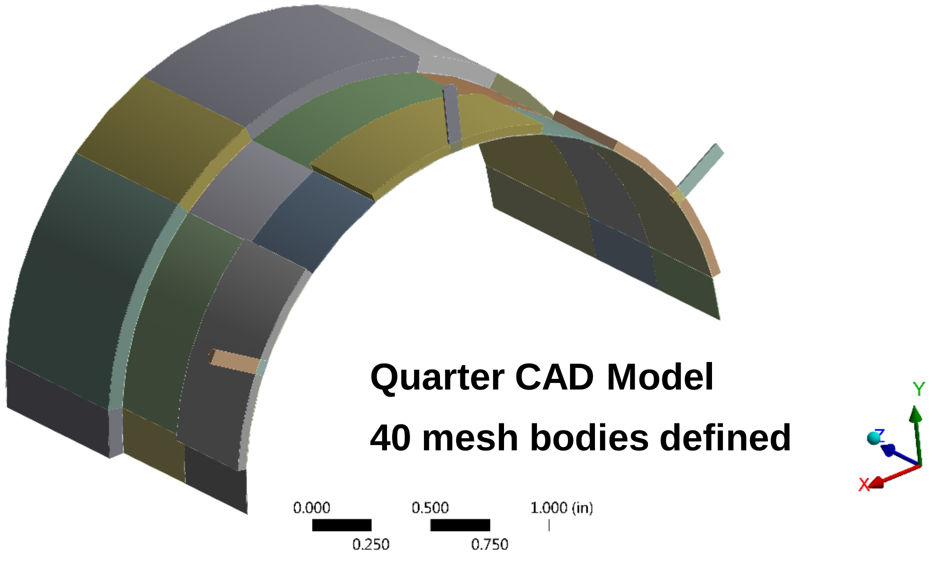
Optimization of Nozzle and Pocket Geometries

- The following conclusions were drawn from this study
  - The nozzle diffuser had no significant performance benefit
  - The obstruction increased the restoring force by less than 4 percent, but increased flow by 25 percent
  - Pocket size had the greatest effect on maximizing restoring force
  - Increasing pocket depth reduced flow velocity (jetting) and avoids supersonic flow at high pressure ratios
  - Nozzle diameter has less than a primary effect on force, but
    - quadratically changes flow rate
  - Too small a nozzle can cause a phase change as the static pressure drops below the critical pressure line

Full Bearing CFD Analysis

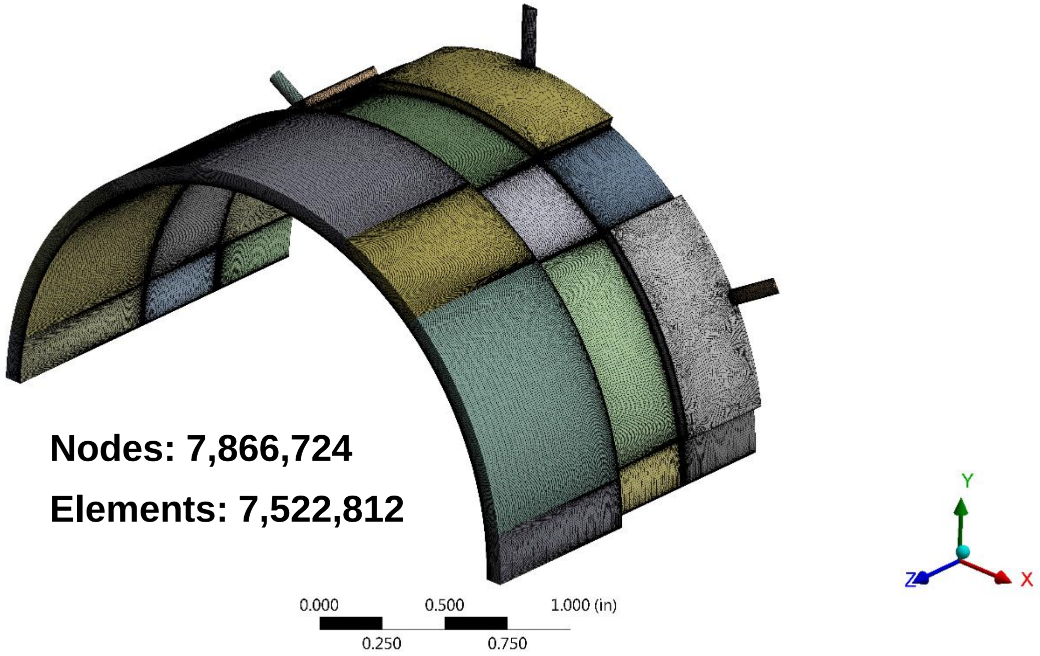
- Radial bearing geometry selected for analysis:

Bearing diameter:	2.50 inches
Bearing length:	1.50 inches
Radial clearance:	0.002 inch
Pocket size:	0.875 inch
Number of pockets:	6



Quarter CAD Model

40 mesh bodies defined



Nodes: 7,866,724

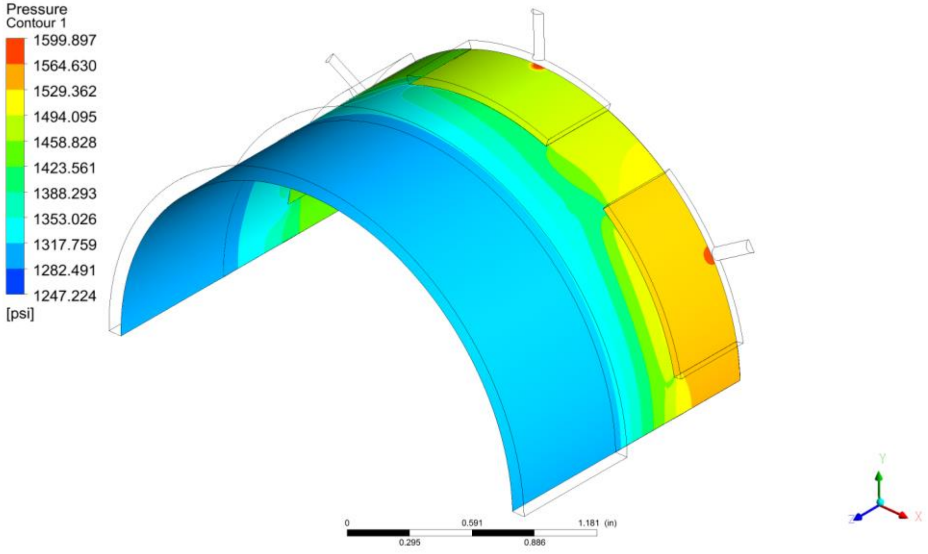
Elements: 7,522,812

Full Bearing CFD Analysis

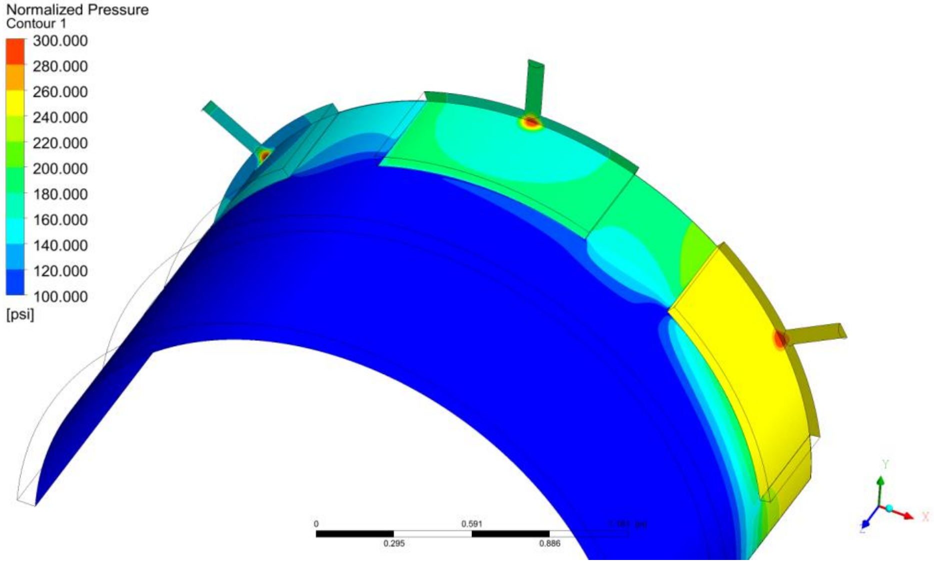
- The analysis was run for the following conditions:
  - Inlet pressure: 1600 psi (11 MPa)
  - Discharge pressure: 1300 psi (9 MPa)
  - Inlet temperature: 275°F (125°C)
  - Shaft eccentricity: 25%, 50%, 75%
  - Rotational speed: 0, 30k, 40k, and 50k rpm
- The following variables were tracked in post-processing the results:
  - Forces on the shaft
  - Mass flow rate
  - Minimum domain pressure (to avoid phase change)
  - Maximum domain Mach number (to avoid sonic flow)
  - General pressures and temperatures at the flow boundaries

Radial Bearing Results: Non-Rotating

- The bearing had the following performance at 50% eccentricity, zero speed:
  - Restoring force: 310 lbf (10x hydrodynamic bearing)
  - Stiffness: 310,100 lbf/inch
  - Min. static pressure: 1247 psi (no phase change)
  - Max. Mach number: 0.463
  - Total flow rate: 0.42 lbm/sec



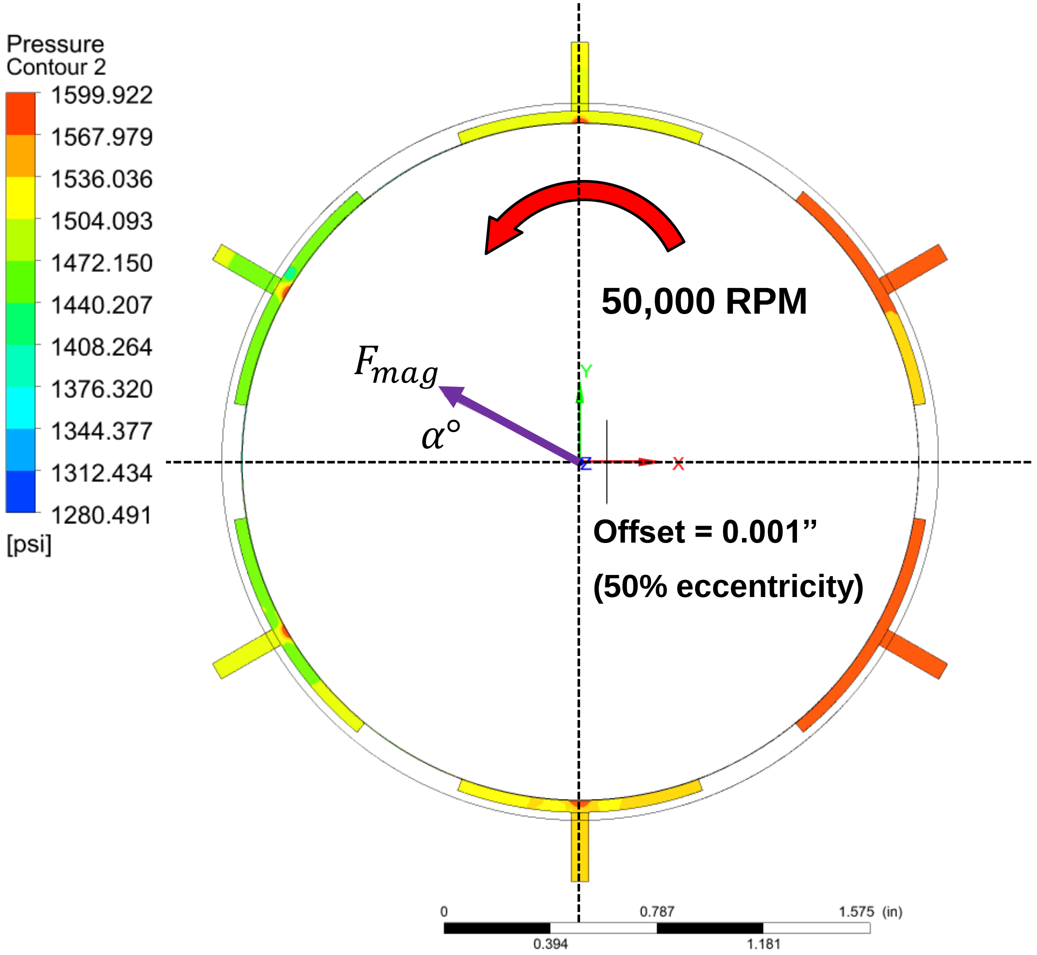
Static Pressure (absolute)



Static Pressure (normalized)

Radial Bearing Results: Rotating

- The model was expanded to a full 360° to include rotational effects

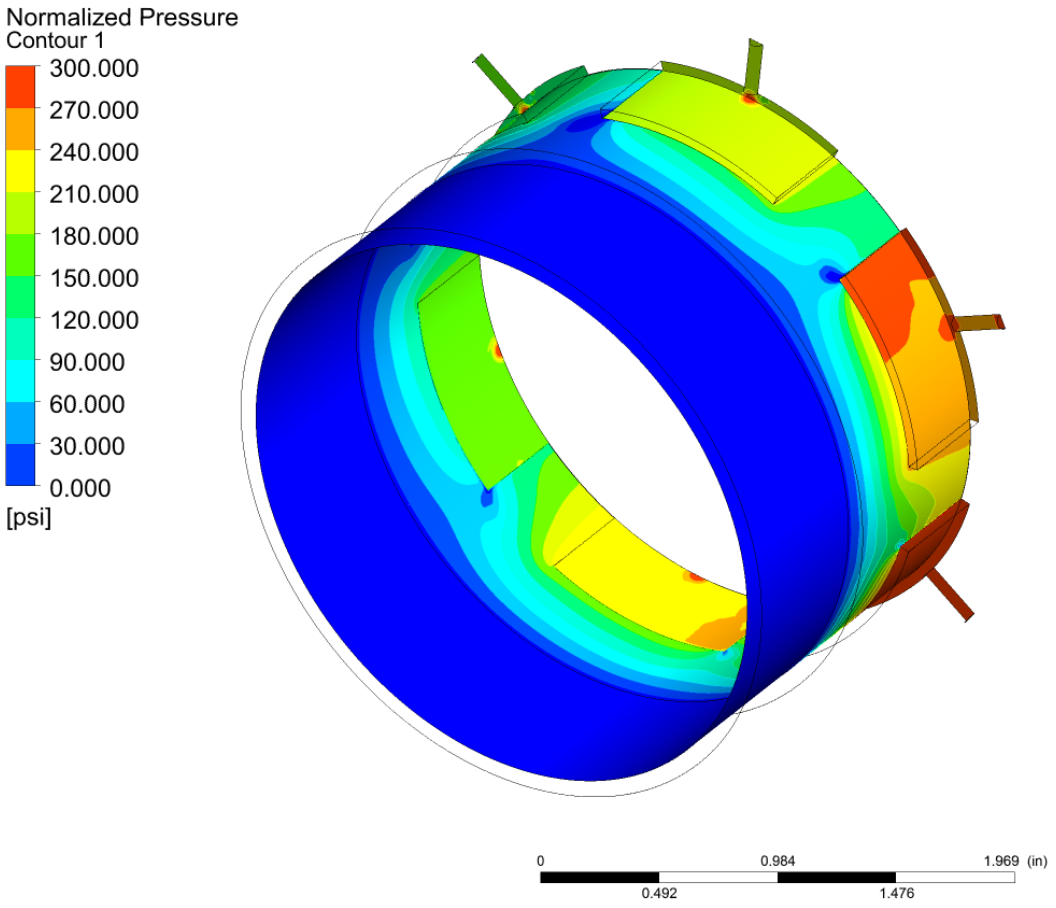


50,000 RPM

Offset = 0.001”

(50% eccentricity)

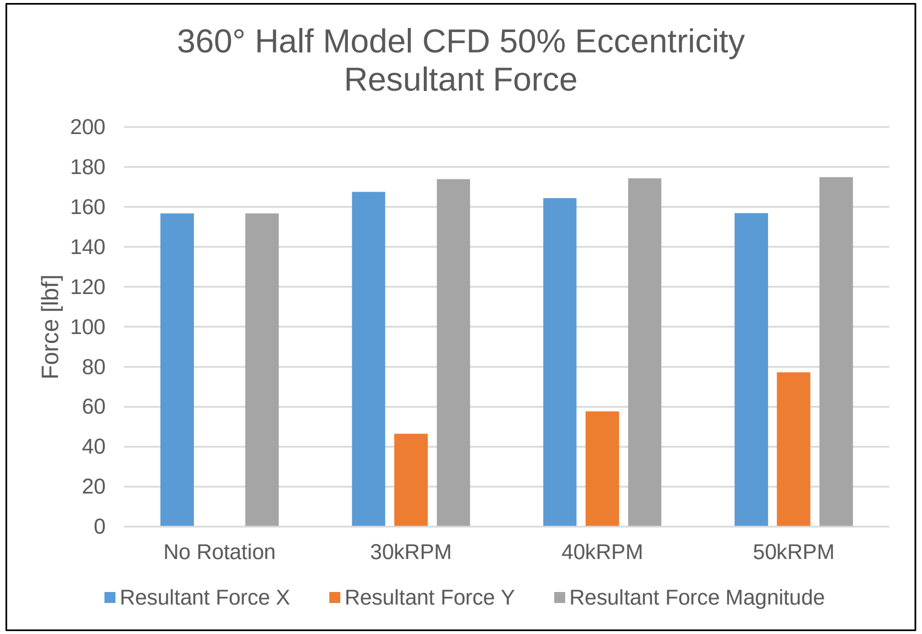
Static Pressure (absolute)

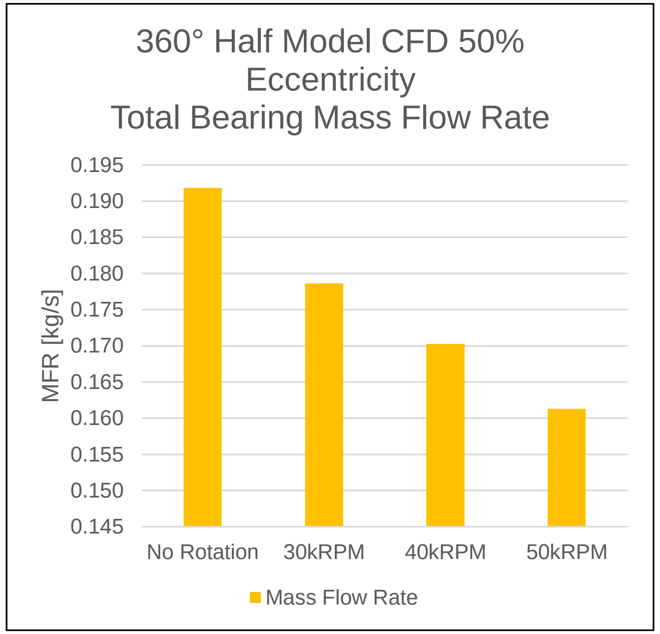


Static Pressure (normalized)

Radial Bearing Results

- The addition of shaft rotation resulted in:
  - a modest tangential force, or cross-coupling force
  - a reduction in the total mass flow rate through the bearing


360° Half Model CFD 50% Eccentricity Resultant Force


360° Half Model CFD 50% Eccentricity Total Bearing Mass Flow Rate

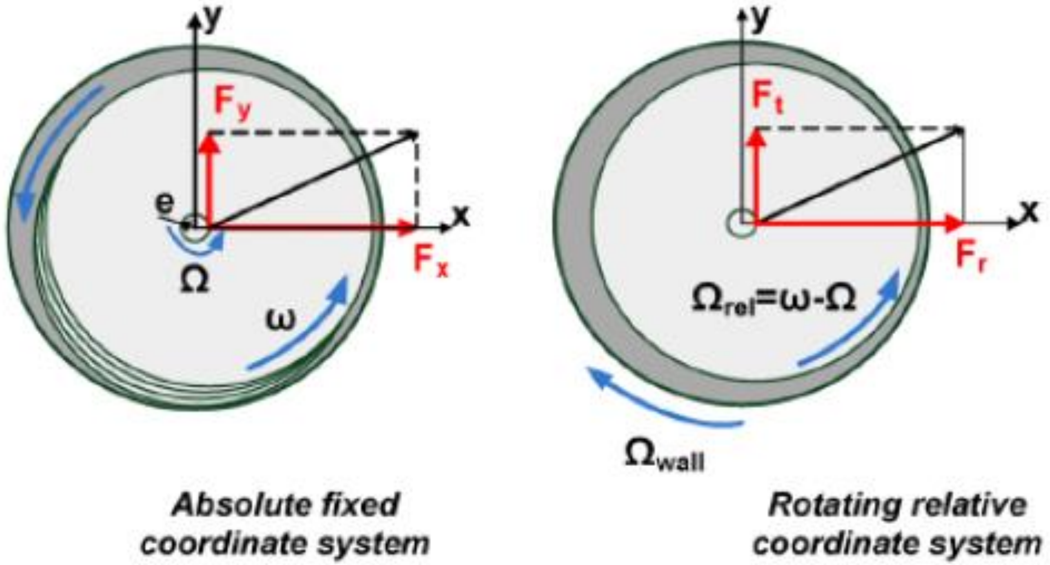
Derivation of Stiffness and Damping Coefficients

- Characterization of the bearing coefficients was accomplished using a modified frame of reference method[2, 3]
- Constant direct stiffness, direct damping, and cross-coupling coefficients were derived assuming that rotor whirl would be in the range of 25% to 60% of the rotating speed
- The bearing geometry is not axisymmetric and therefore the method is not exact
  - though it is cyclic-symmetric
- Values were compared to Someya[4] and found to be reasonable

2. Athavale, M. M., and Hendricks, R. C., "A Small Perturbation CFD Method for Calculation of Seal Rotordynamic Coefficients," International Journal of Rotating Machinery, 2(3), pp. 167-177, January, 1996.
3. Wagner, N.G., Steff, K., Gausmann, R., Schmidt, M., "Investigations on the Dynamic Coefficients of Impeller Eye Labyrinth Seals," Proceedings of the Thirty-eighth Turbomachinery Symposium, pp. 53-69, September 2009.
4. Someya, T., "Journal-Bearing Databook", pp. 179-180, Springer-Verlag Berlin, 1989.

Derivation of Stiffness and Damping Coefficients

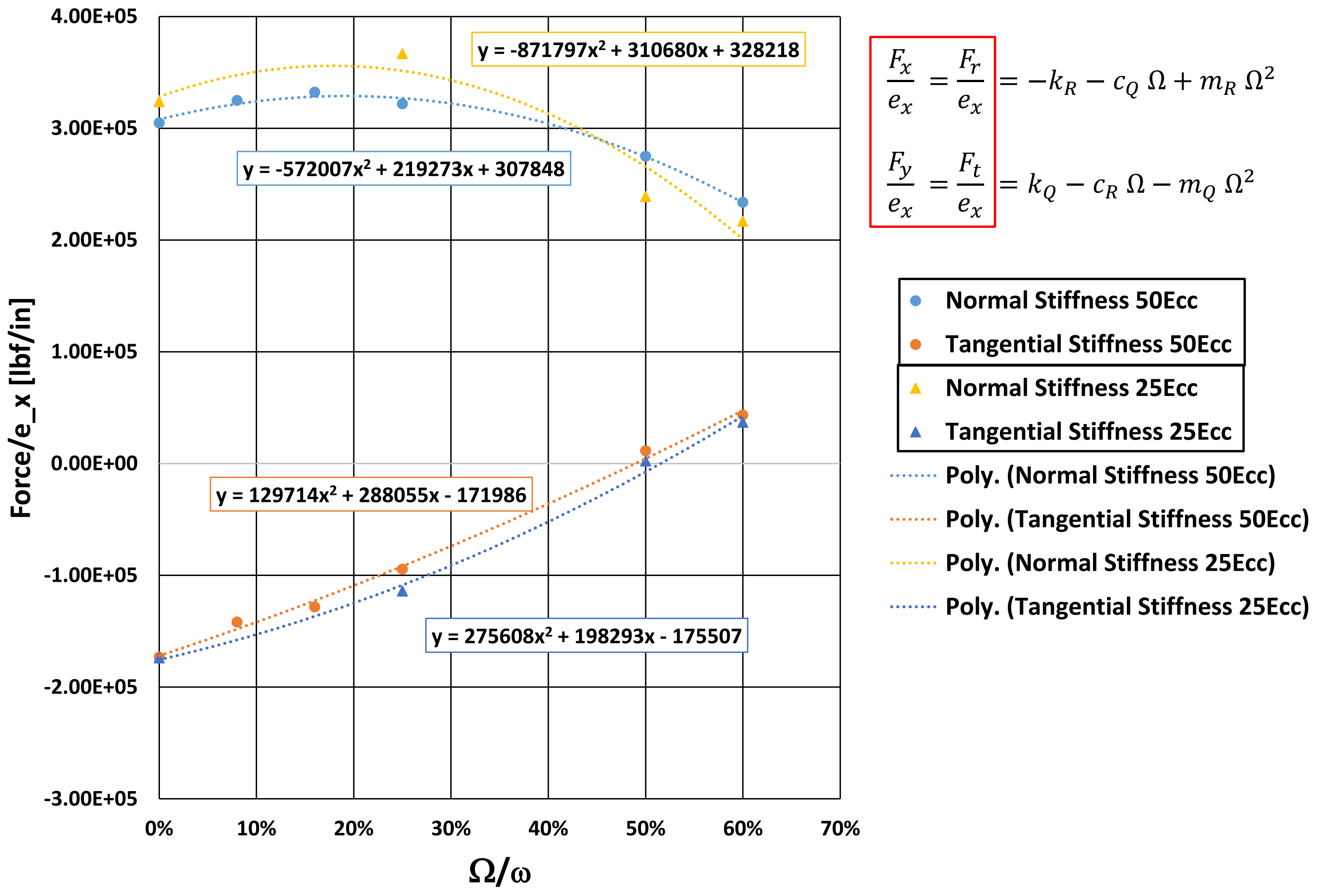
- The applied technique is an attempt at simulating sub-synchronous whirl at fractions of operating speed
- The technique considers the transient problem of a rotor with a spinning frequency of ω rotating around the center of the bearing with a whirl frequency of Ω



Derivation of Stiffness and Damping Coefficients

Normal and Tangential Stiffness Coefficients vs. Whirl Ratio

eccentricities of 25 and 50 percent

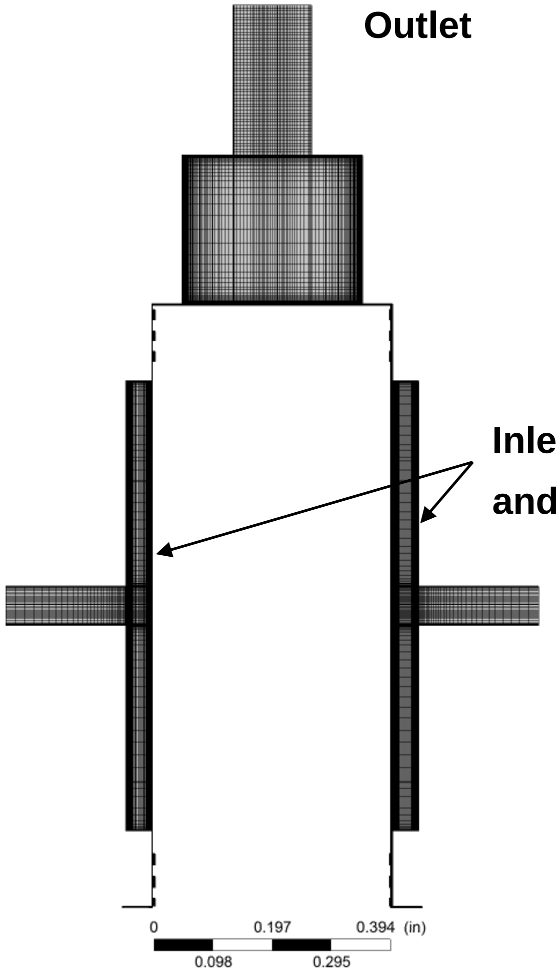


$$\frac{F_x}{e_x} = \frac{F_r}{e_x} = -k_R - c_Q\,\Omega + m_R\,\Omega^2$$

$$\frac{F_y}{e_x} = \frac{F_t}{e_x} = k_Q - c_R\,\Omega - m_Q\,\Omega^2$$

Thrust Bearing Design

- Radial bearing geometry selected for analysis:
  - Outer diameter: 4.00 inches
  - Inner diameter: 2.00 inches
  - Axial clearance: 0.002 inch per side
  - Pocket size: 0.875 inch
  - Number of pockets: 8

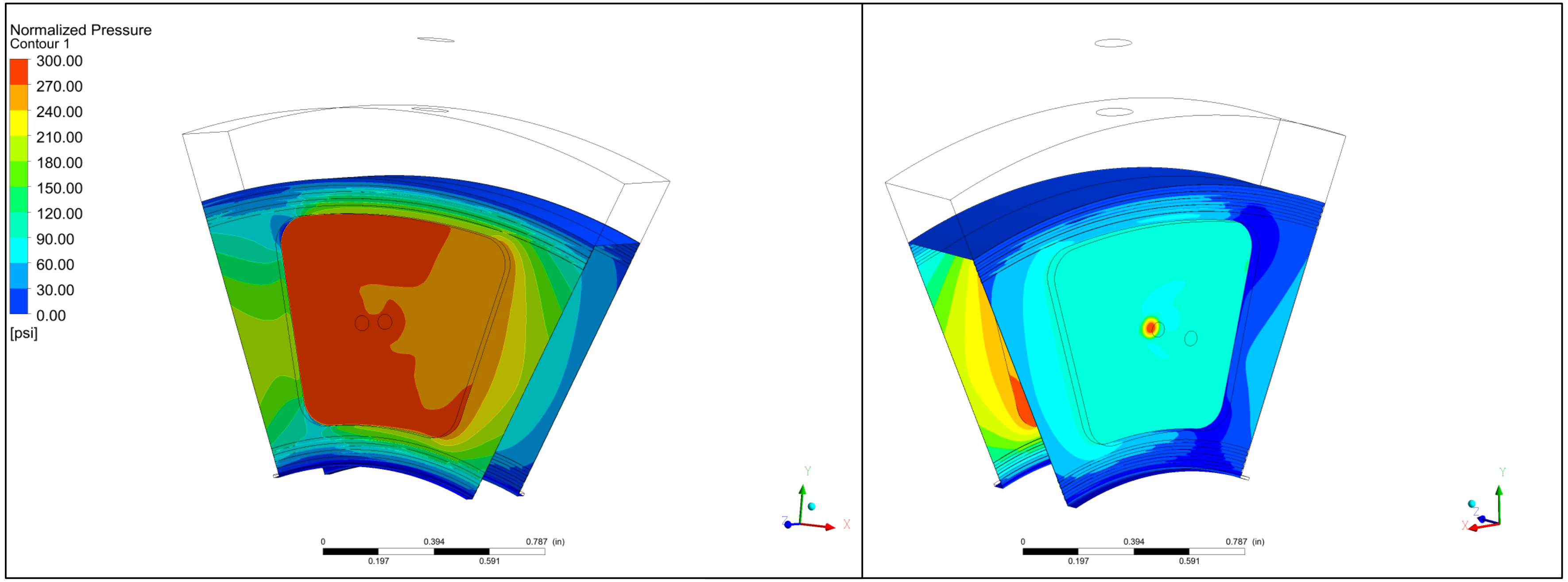


Outlet

Inlet Nozzles and Pockets

Thrust Bearing Performance

- Performance with a 0.001” axial offset at 50,000 rpm:
  - Total axial force: 1,286 lbf
  - Total flow rate: 0.74 lbm/sec
  - Total torque: 2.99 lbf-ft
- Normalized pressure distribution on both sides of thrust disk



0.001” Gap Side

0.003” Gap Side

In Conclusion:

Future Work

- Current efforts are focused on finalizing manufacturing drawings for the bearings and test rig components
- Hardware procurement will be ongoing over the next several weeks
- Testing will be conducted at MSI (in air) and at Sandia National Labs (sCO_2 environment)
- The bearings will be instrumented to measure local temperatures and pressures throughout
- Data will be compiled and compared to the theoretical models

Summary

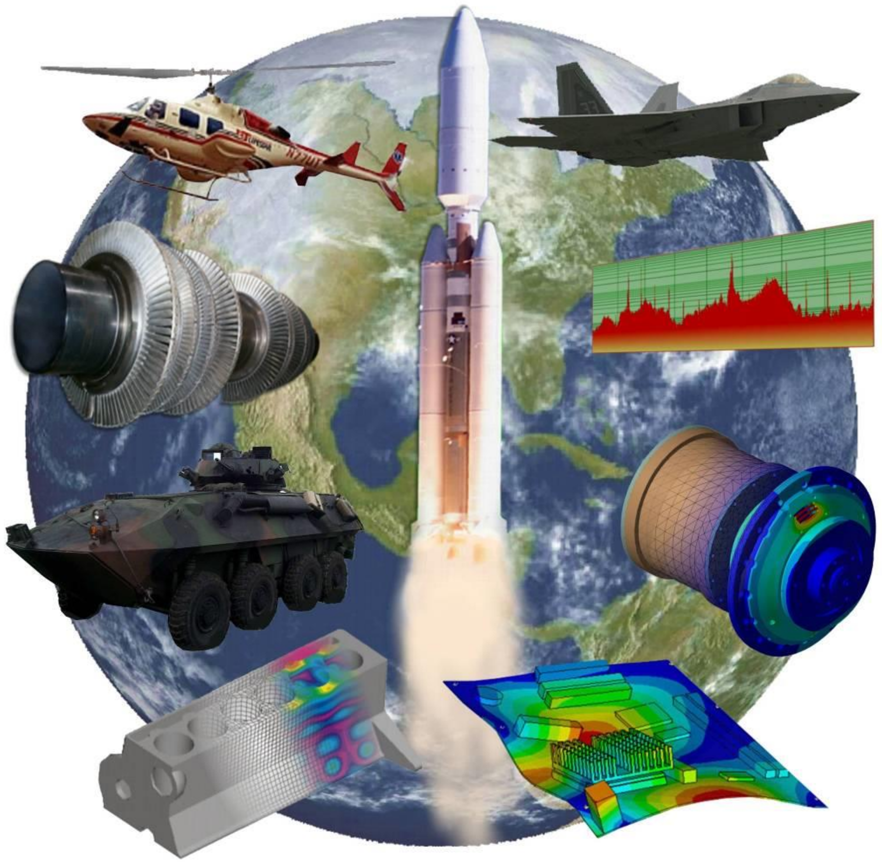
- The hybrid foil bearing designs show great promise using sCO_2 as the working fluid (and likely other fluids)
- To date, the predictions show that the hydrostatic assist can generate enough load capacity to provide an effective bearing design for sCO_2 turbomachinery

Acknowledgment

This material is based upon work supported by the U.S. Department of Energy, Office of Science, Office of Fossil Energy, under Award Number DE-SC0013691.

Disclaimer

Visit Our Web Site:
www.mechsol.com



Peter Chapman 973-326-9920 ext. 144; pac@mechsol.com