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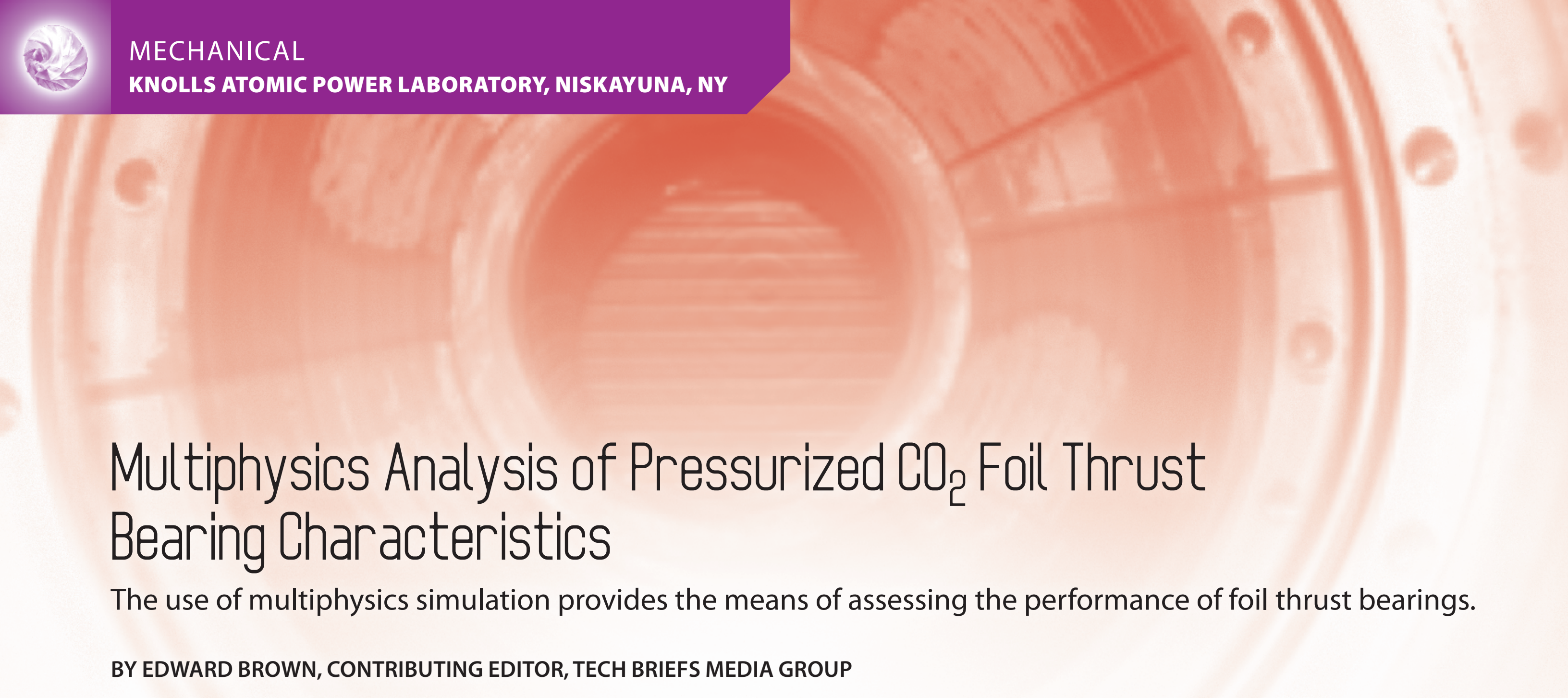

Multiphysics Analysis of Pressurized CO_2 Foil Thrust Bearing Characteristics

The use of multiphysics simulation provides the means of assessing the performance of foil thrust bearings.

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The Knolls Atomic Power Laboratory (KAPL) and Bettis Atomic Power Laboratory are testing a supercritical carbon dioxide (S-CO_2) Brayton power cycle system with a 100 kW electric power output (kWe). The 100 kWe Integrated System Test (IST) is a two-shaft recuperated closed Brayton cycle with a variable-speed turbine-driven compressor and a constant-speed turbine-driven generator using S-CO_2 as the working fluid. The main goals of the IST are to provide test data to verify the ability to model important thermodynamic characteristics of an S-CO_2 Brayton cycle system and to demonstrate S-CO_2 Brayton cycle system controllability in various operating modes and transients.

Supercritical CO_2 is a means of providing high energy conversion efficiency at moderate temperatures, as compared to a helium system. Near the critical point, carbon dioxide rises to a very high density — about 700 kg per cubic meter — close to the density of water. This means that the compressor acts more like a pump. The high density of the CO_2 working fluid is what makes it possible to use smaller components, rotating at higher speeds, with fewer turbine stages than for a comparably rated helium closed-loop Brayton power system (Figure 1). It takes far less work to run the compressor, so it requires less input energy to generate a given output. Note, it is not practical to build an S-CO_2 system for generating any power less than 100kWe because the turbomachinery would have to be too small and too fast.

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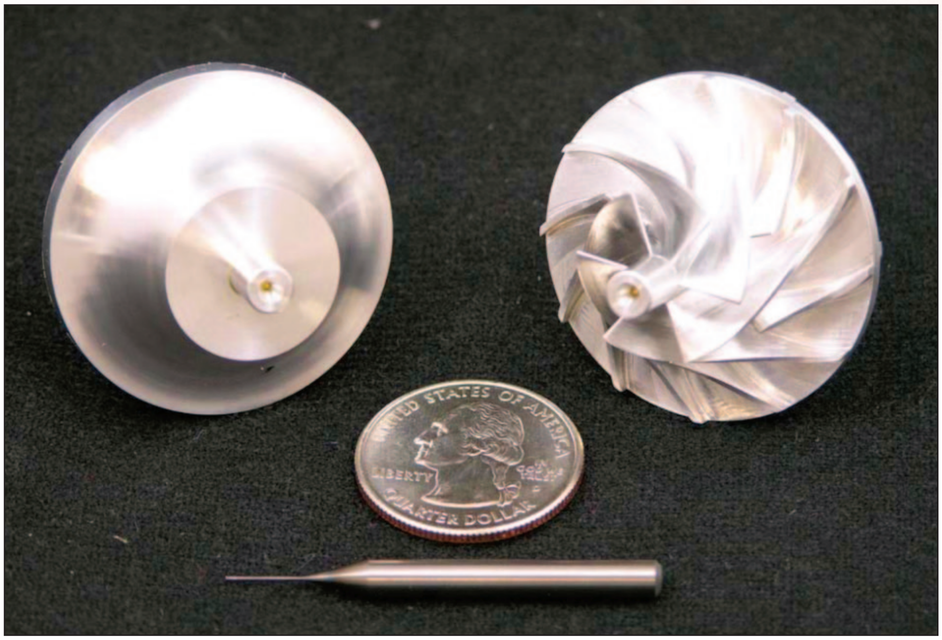

Figure 1: Representative size of IST compressor.

System Operating Conditions

A critical factor for system operation at high operating speed and high gas pressure is that the turbine shaft bearings must withstand a wide range of radial and axial loads. Gas foil journal bearings to handle the radial load were manufactured by Capstone Turbine Corporation. The thrust bearing (Figure 2), used for constraining the axial forces, was designed by NASA Glenn Research Center and manufactured by Barber Nichols, Inc. Under IST operating conditions, the bearings must eventually operate at 75,000 rpm in 200 psia CO_2. However, during component testing, undesirable thrust bearing wear patterns were observed at shaft speeds of less than 35,000 rpm (Figure 3), even with bearing temperatures that were maintained at well below the

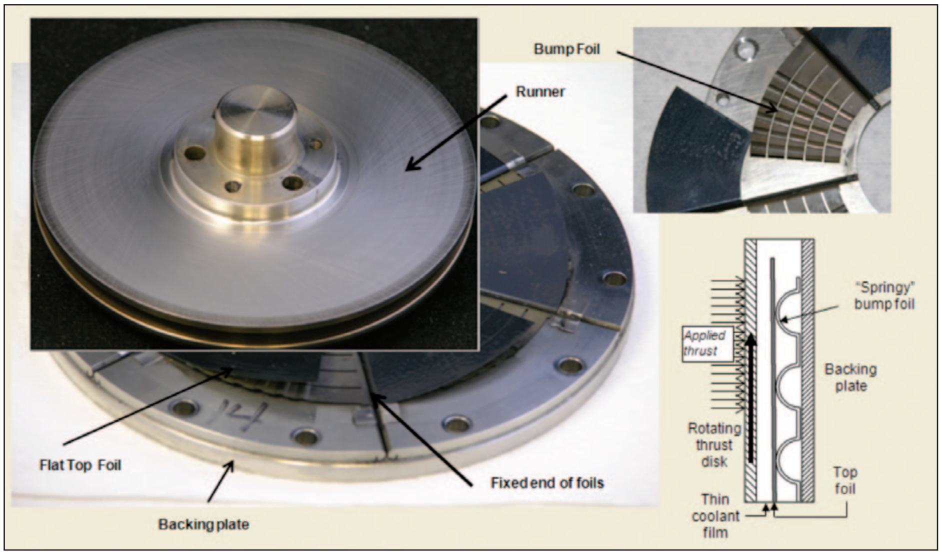

Figure 2: Parts of a thrust foil bearing.

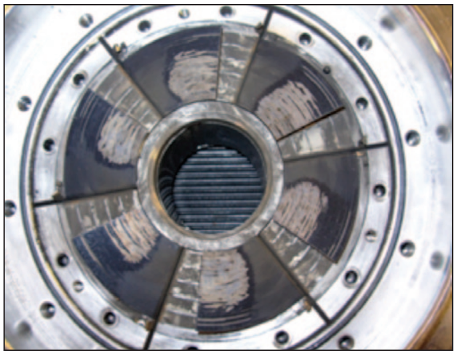

Figure 3: Foil thrust bearing wear after a short time in service.

imposed 400 °F operational limit. In some cases, complete destruction of the foils occurred. Multiphysics analysis was performed by KAPL to provide insights into identifying potential failure mechanisms. COMSOL Multiphysics was used to model the thrust bearing over the speed range of operation.

Thrust Bearing Structure

The thrust bearing, as shown Figure 2, consists of four components. From top to bottom:

- The runner, which is attached to the rotating turbomachine shaft. This is what imparts the thrust load to the bearing. Its spinning surface forms the upper boundary for the CO_2 lubricating film.
- The flat top foil, which provides the lower boundary for the development of the thin film that generates the hydrodynamic pressure to counteract the thrust load.
- The bump foil, which is a spring-like structure with bumps that are deflected under load, providing the bearing compliance.

The backing plate provides support for the top and bump foils.

Modeling

Trevor Munroe of Knolls Atomic Power Laboratory, Niskayuna, NY, performed an analysis for understanding the failure mechanism of the bearing. Turbomachinery designed for sealed systems must contend with viscous power loss attributed to bearings and rotor windage. This power loss can reduce turbomachinery performance and necessitate additional cooling mechanisms due to fluid heat build-up. COMSOL Multiphysics can couple built-in structural mechanics, fluid mechanics, and heat transfer options. The model should account for:

- Contact and rubbing (friction) between the top foil and the bump foil as well as between the bump foil and the backing plate. Axial displacement and rotation of the runner.
- The interaction between the fluid (supercritical CO_2) and the structure (particularly in the thin film region).
- Viscous heat generation, particularly between the runner and the flat top foil along with the accompanying heat transfer by conduction and convection.

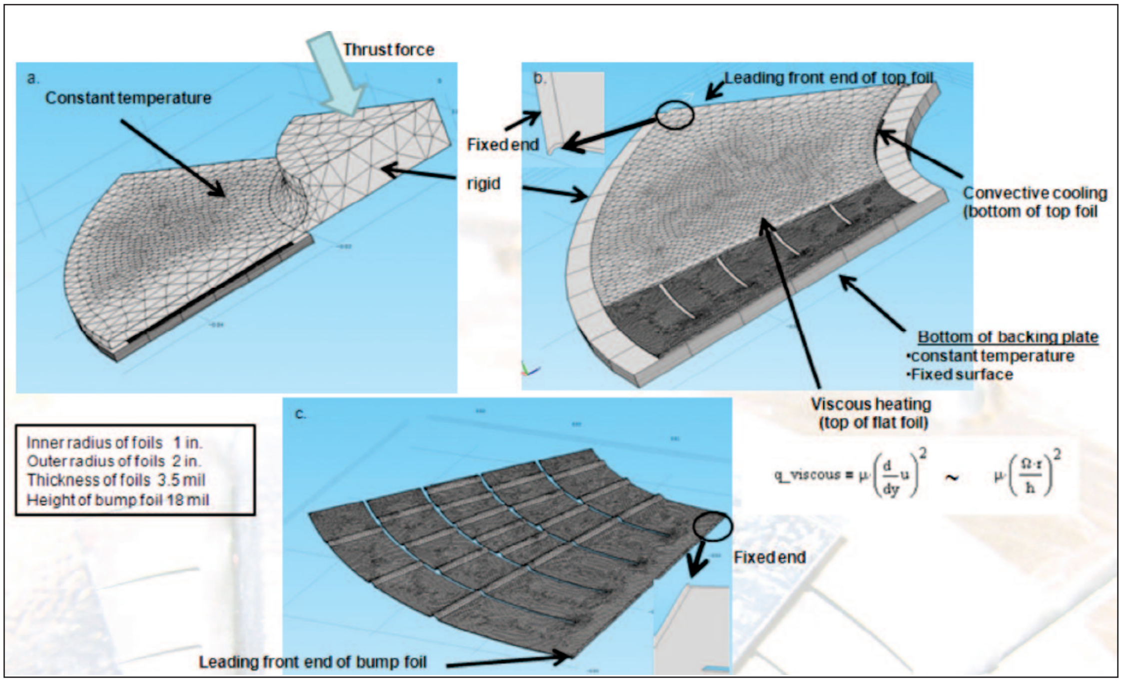

Figure 4: Foil thrust bearing model boundary conditions.

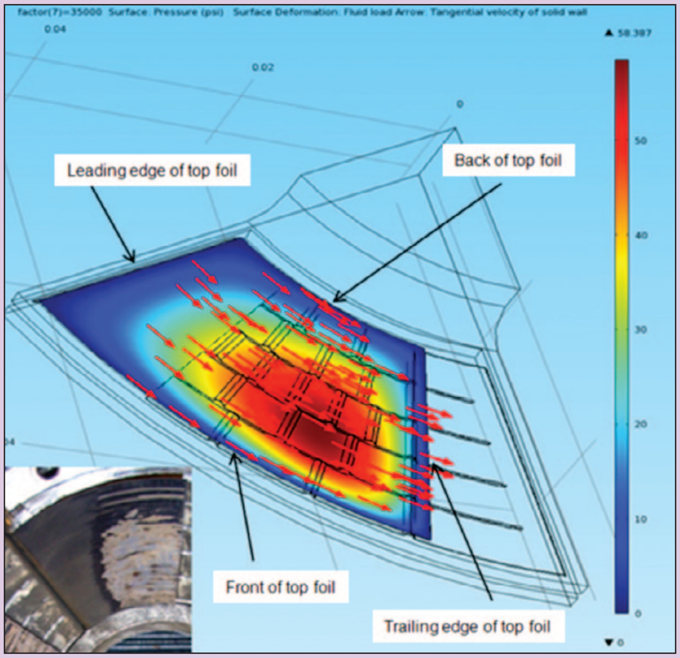

Figure 5: Top foil pressure distribution at 35,000 rpm. (red indicates higher pressure)

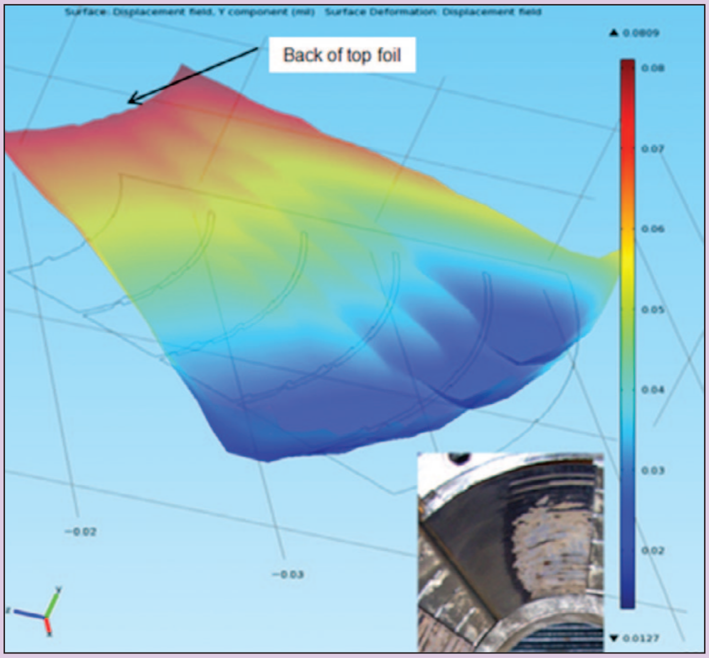

Figure 6: Top foil distortion at 35,000 rpm. (red indicates greater thermal growth)

The model must take account of the effects of thermal expansion of the foils.

Analysis

Munroe imported a three-dimensional Pro/ENGINEER® model of the bearing's physical geometry as shown by Figure 4. He then set up the boundary conditions as indicated. He made some simplifying assumptions to minimize the necessary computer resources. The first compromise was to use the Thin-Film Reynolds equation option (lubrication model) in the CFD Module to model pressure in the thin-film region. This avoids the complication of meshing an extremely long aspect ratio volume. Heat transfer boundary conditions were applied to model the cooling flow external to the thin film region. The next simplifications concerned the structural and thermal elements. The heights of the runner and backing plate were reduced in order to minimize the number of mesh elements needed for the model. The modulus of elasticity was increased to compensate for the reduced thickness in order to ensure that these components would remain relatively rigid. Future analyses could be improved by applying the thin-film boundary layer option within the thin film region and full Navier-Stokes equations option within the cooling flow regions.

Results

The results for a case with a thrust load of 30 lbs and a range of rotational speed of 5,000 rpm to 35,000 rpm are presented in Figures 5 and 6. Figure 5 shows the pressure distribution on the top foil. The arrow shows the rotational direction of the runner. As would be expected, the maximum pressure is towards the region of diminishing film height (ranges from 0.12 mm to 0.28 mm over the range of speed) or away from the leading edge of the flat foil. The maximum pressure of 58.88 psi is shown. Integration of the pressure distribution over the surface of the top foil or runner results in an overall force of 30 lbs., which balances the imposed force.

Figure 6 shows the distortion of the bump foil under non-isothermal conditions. Under non-isothermal conditions the distortion increases with rotational speed. For the isothermal case, the distortion is not noticeable since the thrust load is being kept constant over the range of rotational speed. This distortion of the bump foil is related to the effect of thermal expansion. The inserted images in Figures 5 and 6 show that more wear of the flat foil seems to occur in the area of high pressure and higher thermal growth.

Conclusions

COMSOL Multiphysics has the essential features necessary to assess the performance of foil thrust bearings. The results indicated that thermal expansion of the bump foil due to viscous heating is an area of concern. COMSOL provides additional capabilities to model flow and heat transfer beyond those used for this work. Future analyses should incorporate a 3D fluid flow model outside the region of the lubricating film to address the cooling and flow distribution issues throughout the bearing and optimize foil geometry to improve bearing performance. ■

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