SIGNIFICANCE

Thrust hydrodynamic bearings (TBs), oil or process fluid lubricated, are vital components in rotating machinery. Axial loads in a turbomachine arise from pressure fields on the front shroud and back surface of an impeller, hence are shaft speed (and load) dependent. To date, prediction of aerodynamic induced thrust loads is still largely empirical. Hence, the need to design and manufacture proven thrust bearings into any turbomachinery. This proposal addresses to the shortcomings in TB technology by delivering reliable experimental results to validate predictive tools that will better the engineering design of TBs.

DESCRIPTION OF THRUST BEARING TEST RIG

Figure 1 depicts a cross sectional view of the test rig with water lubricated hydrostatic bearings, thrust and radial. The test rig was constructed in 2007 with USAF funds [1]. A motor, through a diaphragm coupling and quill shaft, drives a rotor comprised of a 197 mm long 316 stainless steel shaft with two 316 stainless steel thrust collars. The shaft diameter at the location of the radial bearings is 38.1 mm, while the thrust collars have a diameter of 108 mm. Two radial hybrid bearings (4-pad flexure-pivot with diameter=38.1 mm and radial clearance=0.089 mm) support the test rotor, whose center of mass locates at mid span between the two bearings. The test rig has two thrust bearings (8 pockets with inner diameter $D_i=40.6$ mm and outer diameter $D_o=76.2$ mm); one is a test bearing and the other is a slave bearing, both facing the outer side of the thrust collars on the rotor. The slave TB is affixed rigidly to a bearing support, as shown on the right of the figure. Through a non-rotating floating shaft, a load system delivers an axial load (static and/or dynamic) to the test TB. Two aerostatic bearings support the axial load shaft with minute friction. The test TB displaces to impose a load on the rotor thrust collar. The slave TB reacts to this axial load.

Figure 1. Cross sectional view of hydrostatic thrust bearing test rig [1].

SUMMARY OF WORK IN 2014-15

Technical report TRC-B&C-03-15 [2] describes the revamping of the thrust bearing test rig as well as measurements taken during its preliminary operation. This past year, graduate students Michael Rohmer and Scott Wilkinson completed the revamping of the test rig that included manufacturing two new rotors, repairing the quill shaft and diaphragm coupling, repairing the housing, aligning the motor shaft and test rotor, designing a static load system, upgrading the water delivery manifold system, installing instrumentation, and developing means of data acquisition. Presently, on its maiden operation, the revamped coupling and rotor system have turned to a (low) rotational speed of 5 krpm.
Measurements of TB static load performance were obtained with water at room temperature, 24 °C, delivered at an increasing supply pressure ($P_S$), max. 4.14 bar (g). In the tests, conducted under a pure hydrostatic condition, i.e., without shaft speed, the floating load shaft pushes the test TB towards the thrust collar and eddy current sensors measure the axial clearance at three locations. A strain gauge sensor records the pressure ($P_R$) in one of the bearing pockets. Fig. 2(a) shows the average axial clearance ($c$) decreases as the applied load per unit area ($W/A$) increases. Note $A = 32.6 \text{ cm}^2 = \frac{1}{4} \pi (D_o - D_i)^2$. Increasing $P_S$ into the TB produces a larger axial clearance for the same load; however, note the specific load ($W/A$) is a fraction of the supply pressure. Fig. 2(b) depicts the pocket pressure ratio ($P_R/P_S$) decreases steadily as the operating clearance increases, i.e., the applied load decreases. Not shown, as the clearance decreases, the flow rate into the TB also decreases because of the increase in viscous drag in the film lands of the TB.

![Fig. 2. Water lubricated hydrostatic thrust bearing (a) axial clearance vs. specific load ($W/A$), and (b) pocket pressure ratio vs. axial clearance. Operation without rotor speed. Water at pressure $P_S = 2.76$, 3.45, and 4.14 bar(g) supplies the thrust bearings. Water at 3.45 bar(g) feeds the journal bearings. Error bars indicate minimum and maximum clearances recorded on the face of the thrust bearing. Tests conducted at room temperature, 24 °C.](image)

Prior work shows that the test rotor and its quill-shaft coupling must be considered as a single unit for a sound rotordynamic analysis. Without the test and slave TBs lubricated with water, impact loads are exerted on the quill shaft and lateral rotor displacements measured towards identifying the fundamental natural frequency and damping ratio of the rotor-coupling-radial bearings system. In these tests, water at an increasing pressure supplies the support radial bearings. Figure 3 shows the results with an inset depicting the predicted fundamental mode-shape of the rotor-coupling-bearings system. The natural frequency, varying from 90 Hz to 100 Hz, increases little with a raise in the magnitude of water supply pressure since the quill shaft flexibility (note the inset) commands the frequency placement whereas the rotor displaces as a rigid body in conical motion. The damping ratio ($\xi$) is surprisingly low, yet nearly doubling from 4.5 % to 7% as the water supply pressure increases from 2 bar(g) to 6 bar(g). The low $\xi$ is mainly due to the extreme flexibility of the quill shaft that has no external damping. Incidentally, Fig. 3 displays XLTRC predictions obtained with hydrostatic radial bearing force coefficients supplied by the model of San Andrés [3] and a carefully crafted structural rotor model. The predictions are in good agreement with the measured natural frequency and identified damping ratio.

![Figure 3. Measured (a) lateral natural frequency and (b) damping ratio for rotor-coupling-radial bearings system vs. water supply pressure into journal bearings. Data derived from impact load tests on a stationary rotor and inactive thrust bearing. Predictions from XLTRC with bearing coefficients from Ref. [2]. Inset shows mode shape of rotor-coupling at its natural frequency. Tests conducted at room temperature 24 °C.](image)
**PROPOSED WORK 2015-16**

On year III, the main objective is to measure the performance of a water lubricated hybrid thrust bearing (eight pocket). The tasks to be performed are:

- Complete design and installation of mechanism for applying static and dynamic axial loads into thrust bearing.
- Measurement of axial clearance vs. thrust load (max. $W = 670$ N [2.0 bar specific load]) for a range of rotor speed (max. 9 krpm) and (max. 6 bar(g)) water supply pressure into the thrust bearings.
- Measurement of rotor axial response from dynamic loads exerted by a shaker for a range of rotor speed (max. 9 krpm), supply pressure (max. 6 bar(g)), clearance (min. 25 μm), and excitation frequency (max. 150 Hz).
- Parameter identification for estimation of axial stiffness, damping and inertia force coefficients for the test thrust bearing.

The proposed work will benchmark a predictive tool for hybrid thrust bearings [4] thus leading to improvements in design, manufacturing and operation of thrust bearings in rotating machinery. The products of the research are important for compressors –barrel and integrally geared, turbochargers and turbo expanders, blowers, etc. In subsequent years, an oil lubricated thrust bearing or a gas face seal or a porous surface gas bearings could also be tested, depending on the interest from the TRC members.

### BUDGET FROM TRC FOR 2015-2016  Year III

<table>
<thead>
<tr>
<th>Item</th>
<th>Cost</th>
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<td>Support for graduate student (20 h/week) x $ 2,100 x 12 months</td>
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<td>Fringe benefits (2.7%) and medical insurance ($255/month)</td>
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<td>Tuition three semesters ($ 363 credit hour x 24 ch/year)</td>
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<td>Re-circulation pump</td>
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<td><strong>Total Cost:</strong></td>
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### REFERENCES


