MEEN 626 Project Proposal

Comparison of Experimental and Numerical Results of Various Bearing Designs

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The objective of this project is to compare experimental results of three different bearing designs with numerical predictions. A plane journal bearing, pressure dam bearing, and a flexure pivot bearing will be designed and manufactured for this project as well as a bearing housing for easy bearing replacement and installation on the test rig. The bearings and housing will be modeled in Solidworks® and manufactured with 3D printing techniques out of polymer. The bearing surface will be printed oversized and sanded to the final clearance; this will result in a bearing with the desired clearance and a smooth surface finish. Each bearing will be tested on a Bentley Nevada rotor kit. Numerical predictions using the bearing codes in the rotordynamic software suite XLTRC^2® will be compared to the experimental results.

The Bentley Nevada rotor kit, shown in Figure 1, is driven by a variable speed motor with the rotor supported by a bushing on the drive side and the bearing being tested on the non-drive end. Four proximity sensors measure shaft position at both drive, and non-drive ends in the horizontal and vertical directions. Load can be adjusted by a spring attached to a rolling element bearing on the shaft. The rotor journal has a diameter of 24.70 mm (0.9724 inch). Each bearing design as seen in Table 1, will have a clearance of 0.102 mm (4 mil) and a length of 10 mm (0.3937 inch) (L/D = 0.4048).

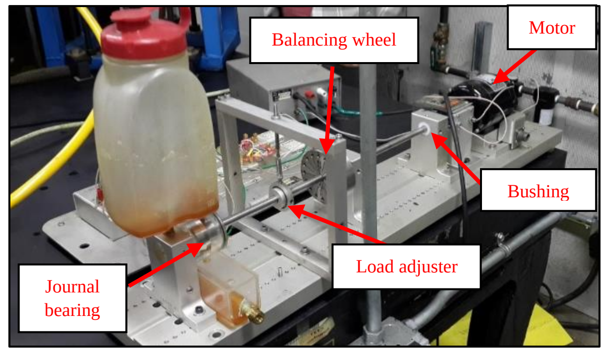
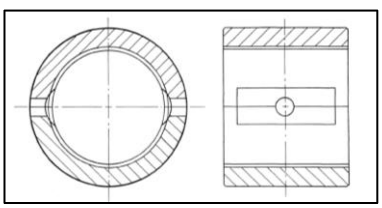
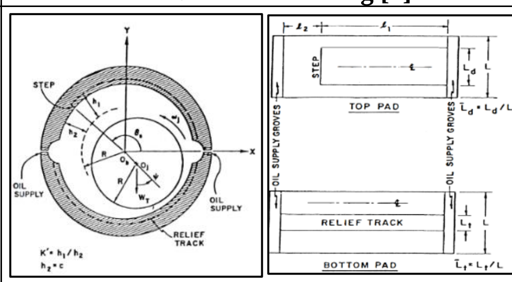
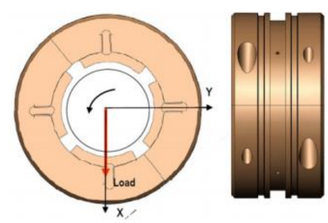


Figure 1: Bentley Nevada rotor kit.

Table 1 Dimensions of test bearings

Plain Journal Bearing[1,2]	Pressure Dam Bearing [2]	Flexure Pivot Bearing [3]
		
- Clearance = 0.102 mm (4 mil)	- Clearance = 0.102 mm (4 mil) - Film Thickness ratio, $h_1/h_2 = 6$ - Axial dam length ratio, $L_d/L = 0.7$ - Relief track length ratio, $L_t/L = 0.5$ - Dam Location, $\theta_s = 140^0$	- Clearance = 0.102 mm (4 mil) - Pivot offset = 0.60 - Preload = 0 - Number of pads and arc length = $4/72^0$

The lubricating fluid will be ISO VG 100 and fed via the oil reservoir located above the bearing. Oil is supplied by the reservoir on the non-drive end of the bearing and exits the bearing on the drive end. Each bearing will be tested at three speeds: 2000, 5000, and 8000rpm. Four target eccentricity ratios will be tested at each speed: 0(centered), 0.27, 0.53, and 0.8. These eccentricities for the plain journal bearing are estimated to have modified Sommerfeld numbers of ∞, 1.6, 0.5, and 0.09 respectively [4].

The following variables will be measured for each test condition: speed, load, flow rate, lubricant temperature in and out, and journal displacement in vertical and horizontal directions.

Directly measured variables including flow rate, temperature rise, journal locus, eccentricity, and attitude angle will be compared to numerical predictions. Power loss will be estimated from the flow rate and temperature rise and compared to the predicted values. Findings will be discussed including the effects of changing speed (if any) for a constant Sommerfeld number on the measured variables. Differences between the three bearing designs will be discussed such as flow rate, temperature rise, load capacity, and static stiffness. Cross coupled force effects between designs will be compared qualitatively by the path of the journal locus as the Sommerfeld number changes.

[1] Salamone D.J., *"Journal Bearing Design Types and Their Applications to Turbomachinery,"* Proceedings of the 13th Turbomachinery Symposium, Texas A&M University.
[2] J. C. Nicholas J.C., and Allaire P.E., 1980, *"Analysis of Step Journal Bearings-Finite Length, Stability,"* ASLE Transactions, 23(2), pp. 197-207.
[3] San Andrés, L., 2006, *"Hybrid Flexure Pivot-Tilting Pad Gas Bearings: Analysis and Experimental Validation,"* J. Tribology, 128(3), pp. 551-558.
[4] J. E. Shigley and C. R. Mischke, Standard Handbook of Machine Design, McGraw-Hill, 1986, pp 28.30.