Topic Code: L

Design and Development of Passive Magnetic Bearings for High-Speed Turboexpander

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Abstract: The high-speed applications such as turboexpander uses aerodynamic gas bearing for contamination free system. The major issue with aerodynamic gas bearing is during the phase of starting and stopping the rotor. Above issue can be prevented by use of the passive magnetic bearing. This paper explains about design and development of passive magnetic bearing for the turboexpander rotating at 80,000 rpm. Two pairs of ring magnets using material Neodymium (NdFeB) alloy magnet of Grade N42 are simulated, and the distance between them is predicted to support the axial load. The two pairs of ring magnets are used in a turboexpander setup and tested their performance.

1. INTRODUCTION

The issue with the aerodynamic bearings is at the start and stop phase of the turbomachine due to direct contact between the bearing surface and runner[1]. To avoid direct contact and minimize the wear rate a pair of passive magnetic rings can be used as axial or radial bearings. These bearings do not need any sensors or controller, unlike active magnetic bearings. They are found to be compact and quite suitable for the current application.

2. DESIGN OF PASSIVE MAGNETIC BEARING

The passive magnet bearing pair is designed for the static conditions to support the dead weight of the rotor during start and stop of the turboexpander. One ring magnet is fixed with the rotor and other with the dynamic seal as shown in Fig. 1. Both the rings are axially polarized with opposite direction to create a repulsive force between the rings. An analytical approach is adopted to predict the distance between the rings for carrying the designed load.

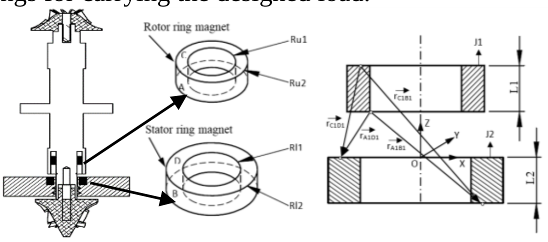

Fig. 1. Arrangements of ring magnets and their parameters as axial passive magnetic bearings.

2.1. Mathematical modeling of active magnetic thrust bearings

The parameters are shown in Fig. 1 is used to calculate magnetic forces between four surfaces A, B, C and D. The elemental magnetic force is expressed in Eq. (1) [2].

$$\vec{F}_{A1B1} = \frac{J_1 J_2 S_{A1} S_{B1}}{4\pi\mu_0 r^3_{A1B1}} \vec{r}_{A1B1} \quad (1)$$

Where J and S are the magnetic polarization of the rings and surface area respectively. The elemental forces between elements of surfaces A-D, C-B, and C-D is calculated using the similar notation of Eq. (1).

2.2. Calculation of magnetic forces

Two pairs of magnetic bearings of material NdFeB, Grade N42 ($\mu0$=10-6 N/A^2 and J= 1.4 Tesla) are used to design the bearing. The pair1 has Ru_1=5 mm, Ru_2=7.5 mm, L_1=5 mm Rl_1=5 mm, Rl_2= 7.5 mm, L_2=5 mm. The pair2 is modified with Rl_1=6.35 mm, Rl_2=9.53 mm, L_2=3.18 mm. The required disnace is predicted from the analytical methodology.

3. DEVELOPMENT AND TESTING

Two pairs of magnetic bearings are developed (Fig. 2) and tested in a turboexpander test facility to study their performance. The FFT spectrum in Fig 3. shows the smooth start of the turboexpander.

Fig. 2. Fabricated rotor and passive axial magnetic bearing for turboexpander.

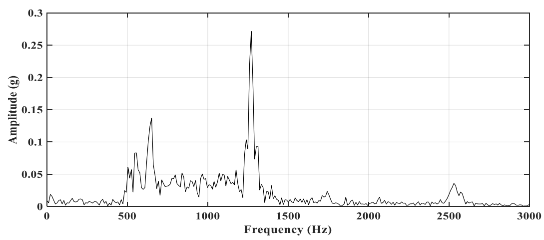

Fig. 3. FFT Spectrum at a speed of 80000 rpm speed.

4. CONCLUSIONS

The developed passive magnetic bearings are found to be compact and reduce the friction during start and stop phase of the rotor of 2.64 N dead weight.

REFERENCES

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