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Rotor Dynamic Stability Analysis of Hybrid Support System

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***Abstract*—Bump foil bearing is a self-acting hydrodynamic bearing which uses bump foils as a spring to fulfill the functions of a bearing. They can be fixed into the rotor radially to support the rotor and permit the relative motion. Permanent Magnetic Bearings (PMBs) are simple lubricant free options for high speed rotors. They can be used to support the shaft in either axial or radial direction but they offers negative stiffness in perpendicular direction. Only for permanent magnet bearing it's not possible to support the rotor. So to support the rotor in both the directions a foil bearing is used in conjunction with permanent magnet bearing where foil bearing supports the negative stiffness of magnet along with dead weight and rotor dynamic loads. The objectives of this project work is to develop a set of foil bearings of different stiffness and measure their clearance gap between top foil and shaft and also variation of stiffness under operating loads. Carry out the rotor dynamic experimental analysis of hybrid support system rotor to determine its stability under given operating conditions for the operating speed range up to 50,000 rpm. Then selecting the promising bearing configuration from the developed bearings for experimentation. Perform the experiments to validate the stability of the system. From the experimentation it is observed that the bearing configuration is stable up to the desired speed range.**

***Index Terms*— Rotor dynamics, Stability, Air bearing, magnetic bearing, Stiffness, Critical speeds, Whirling, Clearance, Lift off**

I. Introduction

Air Foil Bearings (AFBs) or gas foil bearings uses hydrodynamic principle that operates on gas or air for the purpose of rotation of shaft as operational fluid. It is called foil bearing, as it uses foils (sheets of metal) as spring for fulfilling the functions of the bearing. Hydrodynamic theory of lubrication is the basic working principle of foil bearings. An air film formed by hydrodynamic principle during operation separates the rotating shaft from encircling foil or top foil of bearing, at starting and shut-down as the shaft speeds is inadequate to generate gas film pressure causes friction between top foil of the bearing and shaft surfaces. Once the shaft or journal attains enough whirling speed the working fluid (air) drives the foil away from the shaft making it levitate without contact. Passive Magnetic Bearings (PMB's) uses permanent magnets and do not require any input power and it uses permanent magnets as a means to support the rotating object. In this project work stability of the hybrid support system (Combination of foil and magnetic bearing) is analyzed up to the speed of 50000 rpm experimentally and critical speed analysis is done in Samsef field software to predict the stability of the system.

II. Literature Survey

Dellacorte et al [1] has given, "In order to improvise the aircraft propulsion systems, the technology used for the very foundation, the shaft support and lubrication system must be fundamentally changed".

Concept of foil bearing was introduced by Blok and Rossom [2] in the year 1953. Blok and Rossom have pointed out that a foil bearing film thickness, thicker than the rigid gas bearing, could improve the operational reliability and it can provide a solution for problems related to thermal expansion of both journal and its bearing.

Kevin Radil, Samuel Howard and Brian Dykes [3] worked on radial clearances and stated "radial clearance plays an important role in the performance of compliant foil air bearing". With an insufficient amount of radial clearance, the bearing imparts a high preload on the shaft and excessive radial clearance can reduce the load carrying capacity of the bearing. Systems using foil bearings with excessive clearance may experience rotor dynamic instabilities.

A numerical algorithm is developed by Heshmat et al., Peng and Khonsari [4] to predict the load carrying capacities of thrust and journal type of foil bearings. With a pertinent interest Dellacorte and Valco [5] estimated the load carrying capacity of foil bearing by an empirical formula which is derived on the basis of first principle and the experimental data.

Erik Swanson, Sorin Weissman and Chris D. Powell [6] worked on rotary machineries critical speeds and who practically made it understood the terminology of the system and behaviour of shaft vibration at critical speeds along with examining issues which affects vibration. At critical speeds rotor influences vibration and split into forward and backward whirl components that respectively increase and decreases the frequency of vibration.

Guillaume Filion, Jean Ruel and Maxime R. Dubois [7] studied the characterizations of Permanent Magnetic Bearing and concluded that reduction of the force applied on the plain bearing can be achieved by using the intrinsic property of the magnetic field of permanent magnets which consequently reduces the friction induced by the rubbing contacts. This reduction of friction is done while maintaining the bearing

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functionality. Along with this bearing stiffness greatly influences the dynamics of the system.

A study states that nearly $162.3 billion per year is invested on lubrication and another study (bearing news) has reported the bearing market would reach $101 billion in sales. So there is a requirement of air bearings which can eliminate the lubrication system and reducing the overall weight of all systems. Hybrid bearings are new technological answer for elimination of lubrication and complexities related to lubrication and lubricating systems. As foil bearings are less complex, easier for maintenance it reduces the total weight of system increasing its efficiency and reduces the cost. Hence development of these bearings for specific applications will be advantageous [8].

III. PROBLEM STATEMENT

Permanent magnetic bearings are simple lubricant free options for high speed rotors. They can be used to support the shaft in either axial or radial direction with penalty of negative stiffness in perpendicular direction. It is impossible to support the rotor only with permanent magnet bearings. In order to develop total lubricant free rotor system alternate bearing support is required to be used in conjunction with magnetic bearings.

In an attempt to make the lubricant free support system for high speed rotors, CSIR-NAL developed a concept of hybrid support system with foil bearing for radial support and for axial support permanent magnetic bearing. As the permanent magnetic bearing used in axial direction develops the negative stiffness in radial direction, foil bearing has to cater for this load as well along with dead weight and rotor dynamic loads. Both the foil and permanent magnetic bearings have lesser damping and the shaft has to operate at speeds above 50,000 rpm.

IV. FOIL AND MAGNETIC BEARING DEVELOPMENT

A. Foil Bearing Design

Foil bearing design procedure involves designing of the bearing casing, bump foils of sheet metals, rotating shaft and top foils of below given materials.

- Casing and Shaft- EN 24 high strength steel
- Bump Foils and Top Foils - Beryllium-Copper (Be-Cu) and Inconel X-750

Fig. 1. Bump foil

B. Design Specifications

This part involves manufacturing of 4 sets of foil bearings with each set having a pair of foil bearings.

Fig. 2. Foil bearing final set

TABLE I
DESIGN SPECIFICATIONS

Foil Material	Casing Name	Casing Inner dia. (mm)	Shaft dia.(mm)	Bump rad. (mm)
Be-Cu	B1	22.5	15.04	3.3
Be-Cu	B2	22.5	15.04	3.3
Inconel	C1	21.5	15.04	3.3
Inconel	C2	21.5	15.04	3.3
Be-Cu	D1	19.5	15.04	2
Be-Cu	D2	19.5	15.04	2
Be-Cu	E1	17.5	14.87	1
Be-Cu	E2	17.5	14.87	1

C. Magnetic Bearing Development

For current work Passive Magnetic Bearing (PMB) is used. PMB can be realized using radially, axially or perpendicularly magnetized circular magnets. Used PMB is axially magnetized as shown in Fig. 3. Arrangement of magnets changes bearing's stiffness and load carrying capacity.

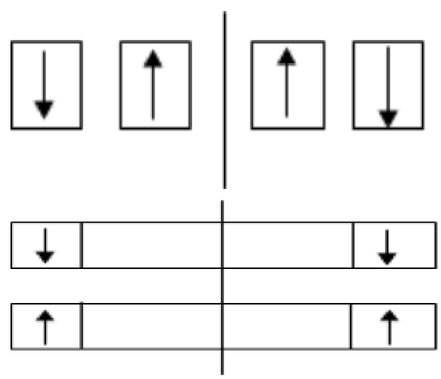

Fig. 3. Axial magnetization

V. CHARACTERIZATION OF FOIL AND MAGNETIC BEARING

A. Clearance Test

The purpose of this test is to measure the clearance space between top foil and shaft of the foil bearing.

Clearance test setup is made as shown in below figure. In this setup two displacement sensors are used, one in horizontal and one more in vertical direction to measure the displacement of the bearing with varying load and a tachometer is used to measure the speed of rotation. Data Acquisition System used here is the NV gate software system to acquire the data.

Fig. 4. Clearance test set up

Bearings have the clearances from 21 microns to 291 microns for varying loads from 0.5 kg to 1.5 kg. Higher the clearance needs lesser speeds to lift off the shaft and lower the clearances needs higher speeds to lift off the shaft.

B. Foil Bearing Stiffness Test

The bump stiffness can be calculated for the given bump geometry with following expressions [9].

$$K_f = \frac{6G}{{r^3}_b} * \text{width of the bump MN/m}$$

Where, G indicates the spring flexure stiffness per unit corrugation length and is given by,

$$G = \frac{E{t^3}_b}{12(1-\gamma^2)}\ \text{Nm}$$

E= 140 GPa for Be-Cu and E= 210 GPa for Inconel X-750

C. Stiffness Characterization of Passive Magnetic Bearing

In this project axially magnetized PMB's are used. The magnets are made up of NdFeB material. Results indicate that the bearing can support the maximum force of 66.8 N with axially offsetting 6.5 mm and offering the stiffness of 22.3 KN/m when rings are at the center [9] which is indicated in Fig. 5.

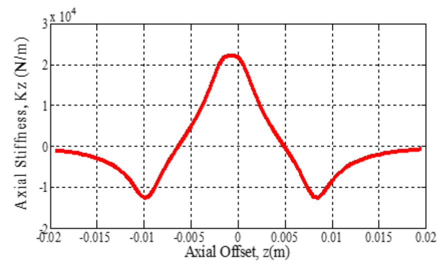

Fig. 5. Stiffness variation with axial offset

TABLE II
DESIGN SPECIFICATIONS

Casing	Bore gauge dia. (mm)	Shaft dia. (mm)	Nominal clearance (microns)	Clearance under weight. (kg)	(micron)
B1	15.11	15.04	70	0.5	193
				1.0	217
				1.5	266
B2	15.09	15.04	50	0.5	25
				1.0	66
				1.5	126
C1	15.29	15.04	250	0.5	107
				1.0	199
				1.5	291
C2	15.24	15.04	200	0.5	102
				1.0	180
				1.5	255
D1	15.09	15.04	50	0.5	20
				1.0	58
				1.5	120
D2	15.07	15.04	30	0.5	58
				1.0	145
				1.5	178
E1	14.97	14.87	100	0.5	56
				1.0	99
				1.5	110
E2	14.9	14.87	30	0.5	21
				1.0	44
				1.5	78

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TABLE III
FOIL BEARING STIFFNESS TEST RESULTS

Casing	Sheet thickness (mm)	Theoretical stiffness (MN/m)	Diff. Load	Deflection (mm)	Exp. Stiffness (MN/m)
B1	0.3	0.6934	NA	-	-
			0.5	24	0.204
			1	73	0.134
B2	0.3	0.6934	NA	-	-
			0.5	41	0.120
			1	101	0.097
C1	0.2	1.0582	NA	-	-
			0.5	92	0.053
			1	184	0.053
C2	0.2	1.0582	NA	-	-
			0.5	78	0.063
			1	153	0.064
D1	0.3	3.1154	NA	-	-
			0.5	38	0.129
			1	100	0.098
D2	0.3	3.1154	NA	-	-
			0.5	87	0.056
			1	120	0.082
E1	0.3	24.923	NA	-	-
			0.5	43	0.114
			1	54	0.182
E2	0.3	24.923	NA	-	-
			0.5	23	0.213
			1	57	0.172

D. Stiffness Calculations of Hybrid Support System

Stiffness invoked in the hybrid support is the combination of stiffness of foil bearing and magnetic bearing which influences the shaft movement in both radial and axial direction. Negative magnetic bearing stiffness influences resultant stiffness in radial direction this has to be countered by the stiffness of foil bearing. The negative stiffness experienced by foil bearing is one fourth of the magnetic bearing's axial stiffness.

Resultant stiffness in radial direction of hybrid bearing,

K = Stiffness of foil bearing – (0.25 * Stiffness of magnetic bearing)

$K = Kf - (0.25 * Km)$

Stiffness results of hybrid support system are tabulated in Table-4.

E. Lift-Off Speed Test

Lift off speed test has conducted to study at what speed of rotation of shaft bearing is able to create the hydrodynamic force to lift the shaft along with the load on it. This parameter is very important to know and predict the load carrying capacity of the bearing and optimum range of speed a bearing is sufficient to lift the shaft. Higher the loads on the shaft eventually take the bearing to lift the shaft at higher speeds and lower the load it takes less speed to lift the loads. Lift off speed test results are tabulated in Table-5.

Lift off speed test results observed from the run indicates bearings with different design specifications are able to lift off and levitates the shaft in the speed ranges from 22,000 rpm to 33,000 rpm. Bearing B lifts off the shaft at lower speed ranges of 22,000 rpm as it contain appreciable clearance space. Bearing E lifts off the shaft at higher speeds of the range 32,000 rpm and reason for this is its lower clearance gap and higher stiffness of the bumps.

VI. EXPERIMENTATION WITH HYBRID SUPPORT SYSTEM

A. Hybrid Support System Assembly

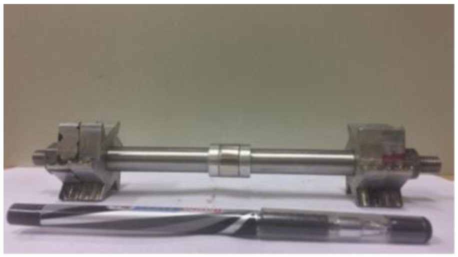

Fig. 6. Test rotor with air turbines

TABLE IV
HYBRID BEARING STIFFNESS TEST RESULTS

Casing Name	PMB Position (m)	Foil bearing stiffness (MN/m)	PMB's –ve stiffness (MN/m)	Hybrid bearing stiffness (MN/m)
B1	0	0.6934	0.01115	0.6878
	0.04		0	0.6934
	0.018		-0.006	0.6964
B2	0	0.6934	0.01115	0.6878
	0.04		0	0.6934
	0.018		-0.006	0.6964
C1	0	1.0582	0.01115	1.0526
	0.04		0	1.0582
	0.018		-0.006	1.0612
C2	0	1.0582	0.01115	1.0526
	0.04		0	1.0582
	0.018		-0.006	1.0612
D1	0	3.1154	0.01115	3.109825
	0.04		0	3.1154
	0.018		-0.006	3.1184
D2	0	3.1154	0.01115	3.109825
	0.04		0	3.1154
	0.018		-0.006	3.1184
E1	0	24.923	0.01115	24.9174
	0.04		0	24.923
	0.018		-0.006	24.926
E2	0	24.923	0.01115	24.9174
	0.04		0	24.923
	0.018		-0.006	24.926

TABLE V
LIFT OFF SPEED TEST RESULTS

Casing	Lift off speed of foil bearing (RPM)
B1	22000
B2	28000
C1	25000
C2	26500
D1	30000
D2	32600
E1	32000
E2	33000

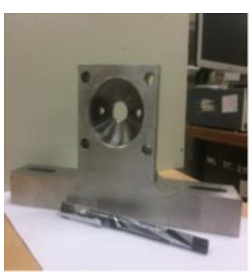
Fig. 7. Magnet holder


Fig. 8. Cover plate

As hybrid support system consists of two main bearings. Radially controlled foil bearing and axially controlling passive magnetic bearing. The solid model of the assembly and demonstration are shown in the figures.

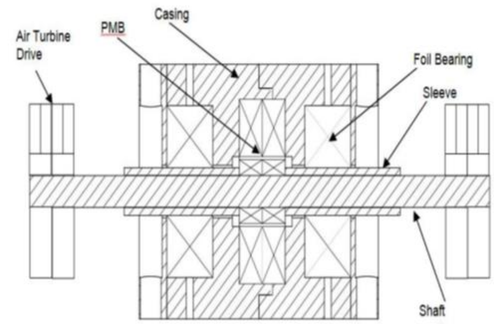

Fig. 9. Test rig schematic

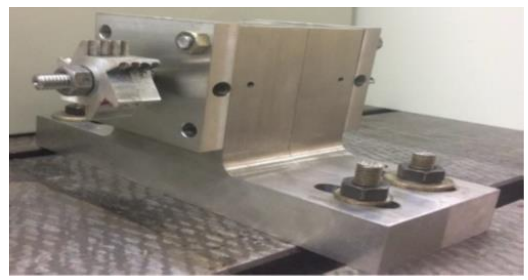
Fig. 10. Assembled test rig

B. Hybrid Bearing Test

Hybrid bearing test is carried out up to the speed of 50000 rpm. The signal indicating change in vibration levels with speed is shown in Fig. 11. This data which is obtained from DAQ system is processed in MATLAB to determine the shaft orbital motion at varying speeds. During coast up it is observed that there exists a metal to metal friction up to 29000 rpm and shaft becomes air borne once it crosses this speed, called lift off speed. Once the shaft becomes airborne there is no resistance for the motion and rotor reaches the maximum rpm within short time and continues to rotate at that speed until air mass flow over the air turbines reduces. During coast down it is observed that the shaft sits on the bearing at speed of 26000 rpm at which distortion in the orbit shape is observed. Orbit plots at lift off and break off are shown in Fig. 6.7. Orbit plot at operating speed of 48000 rpm is shown in Fig. 6.8. From figure it is observed that the sensor is saturating hence plot is appearing flat at the bottom.

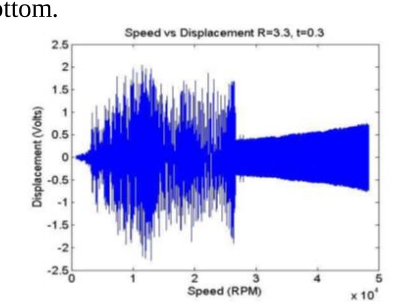

Fig. 11. Variation in overall vibration with speeds

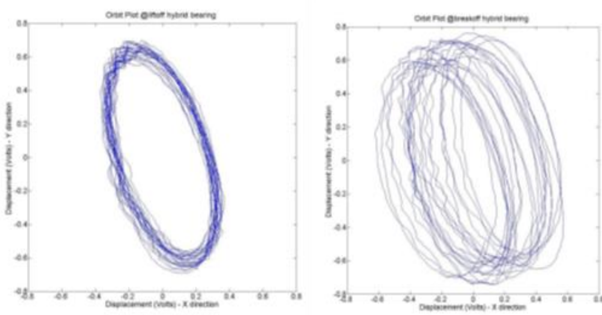

Fig. 12. Orbit plot at lift off and break off speeds

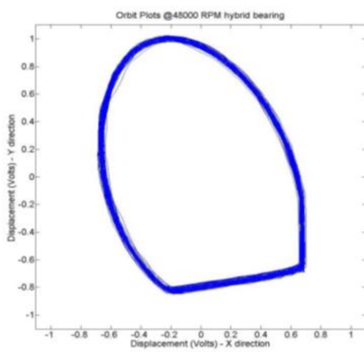

Fig. 13. Orbit plot at 48000 rpm

VII. Results And Discussion

As in the experimental test, clearances obtained are in optimum range which is helpful for bearing to develop the dynamic force at high speeds to lift the shaft and load. Clearances play an important role for allowing a space for lubrication and air to pressurize itself in the gap to lift the load. Stiffness indicates the strength of the system which provides support to heavy loads. Stiffer the support is, higher the load carrying capacity of the system. Less stiffer system with low damping may leads to instability.

Lift-off speed test shows hybrid support system is capable to lift the load at speed ranges 30000 rpm to 35000 rpm with varying design specifications. Once the system achieves the lift off speed the system is completely airborne, allow it to work without any friction between the rubbing surfaces and stable working.

Experimental stiffness depends on the nominal clearance of the bearing. Bumps of higher radius can offer theoretical stiffness with lesser loads and can be flattened with fewer loads. Lesser radius bumps have higher calculated stiffness and they need higher loads to make them flattened. It is observed that the peak to peak vibration amplitude is within 150 microns. It is also observed that the foil bearing lift off speed is higher for higher bump stiffness.

VIII. Conclusion

A. In the present work stability of hybrid support system is successfully analyzed and tested up to a speed of 50000 rpm. The study presented here indicates that the hybrid support system works satisfactorily for the desired application. The foil bearing and PMB can be designed for high speed and low load applications with small variation in lift off speed.

Solutions obtained from the experimental analysis and orbital plots are indicating it's a stable system. Also the system needs to be constrained in both radial and axial positions to keep a rotating system stable.

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