

Effect of varying Stiffness on the Performance of Air Foil Journal Bearing

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Abstract: Air Foil bearings (AFBs) are compliant surfaces, self-acting hydrodynamic bearings typically constructed from several layers of sheet metal foils. These foils rest on elastic support structure and their performance largely depends on this support structure. These type of foil structure consists of bump strips and the smooth foil rests on top foil. AFBs are environmentally friendly and used in many commercial applications, both terrestrial, turbomachinery system and in aerospace. The compliant foil structure is modelled using the Catia. The foil deflections calculated from the structural model using finite element analysis are coupled to the hydrodynamic gas film. In the present work of AFBs, the stiffness of the foil is varied at different speed conditions of the rotor.

keywords: Air foil bearing, Load carrying capacity, Stiffness.

1. Introduction

Air Foil Bearings (AFB's) have gained more attention than any other types of bearings because of their multifunctionality of application. AFB's eliminate the oil system that leads to reduce overall system weight & maintenance cost. AFB's are self-acting hydrodynamic bearings that use ambient air as their lubricant fluid. AFB's are made from sheet metal foil which is comprised of two or more layers. The top foil layer supports a gas pressure film which supports load. The bump foil is a flexed foil which induces friction to the bearings. AFB's have more advantages compared to oil lubricated systems. AFBs offer significant advantages over rolling element bearings like requiring fewer parts to support the rotating assembly without the necessity of oil lubrication supply. Oil-free bearings are environmentally friendly which might be used for applications like optical grinding machine, Digital Printing, helicopter system, air cycle machine in aircrafts, Compressor, IC engines, turbofan engine & optical inspection equipment.

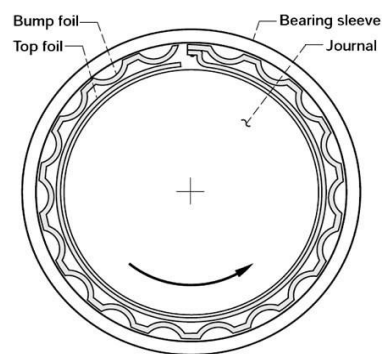


Fig. 1 Compliant air-foil journal bearing

These sort of foil bearing hydrodynamic action allows for an oversized and high speed operation in comparison with the oil bearing system. Additionally, oil lubricants cause larger power losses thanks to friction at the interface between the fluid and bearing. AFB's are elastically supported bearings which resist variations. During operation of the foil bearing, the rotation of the rotor generates the pressurized gas film that blows the highest foil out radially & detaches the highest foil from the surface of the shaft. The pressure during this air film is proportional to the relative surface velocity between the rotor & foil. Thus, faster the rotor rotates, the upper the pressure &

more is that the load the bearing can support. When the rotor starts to rotate, the highest foil and also the rotor surface are in reality until the speed increases to some extent where the pressure within the air film is sufficient to push the highest foil removed from the rotor and supports the load. Whereas the rotor slows all the way down to some extent the speed is insufficient to support the rotor weight, the highest foil and rotor again are available in contact.

2. Literature Survey

In late 1950s the effect of flexibility on the hydrodynamic film was first observed by Blok and Von Rossum. It almost took 20 years to find the first commercial application in the air cycle machine. In 1970s Agarwal's paper is a nice compendium on the development of FB in which the author provides the development statistics and forecasts the possibility of a host of applications in aviation gas turbine engine, cryogenic turboexpanders and auxiliary power unit for aircraft operating at high temperature with enhanced efficiency and reduced cost. So far, developed the models of Air foil bearings considering the reorientation. We hope to be finally able to develop a constitutive model to satisfy the above-mentioned requirements. As a main goal, based on the developed constitutive model we will increase the load carrying capacity of the AFB's and efficient computational tool which can be used in design, analysis of various AFB's structures.

Studying new structural configuration of bump foils and its analysis

Since the effective characteristics of the AFBs are greatly influenced by the behavior of the foils, researchers have experimented with different types of bump foils apart from the conventional corrugated structure to mitigate the structural non-linearity of the bearing.

Among the various types, metal mesh FBs by San Andres and Chirathadam is noteworthy. The authors analyzed the metal mesh bearing by integrating the bump foil uniform stiffness with the finite element (FE) model of the highest foil and paired the FE model with the equation model of the gas film. In their work, the RE was treated using the control volume scheme with a precise flow advection model, and therefore the simulation results were verified by comparing with available experimental measurements. Feng et al. (2016) analyzed the metal mesh analytically by considering the metal mesh substructure as assembled springs with dry friction joints and performed parametric analysis considering several factors like wire diameter, geometrical size and radial interference of metal mesh substructure. The simulation results revealed that viscous damping coefficient and denseness of metal mesh have a pronounced influence on the bearing's dynamic performance.

Exploration stiffness and damping parameters

Stiffness and damping play a very important role within the stability of bearings. Walowit et al. (1973) first introduced a model to quantify the structural stiffness of the compliant surface of FBs by assuming that the foil's structure encompasses a uniform stiffness. Later on, Heshmat et al. (1981) experimentally analyze the effect of coating on the foil surfaces, LCC and stability of supported rotor. During a comprehensive theoretical model, Ku and Heshmat (1992), for the primary time, introduced the frictional interaction forces between bumps and with sleeve surface to research the deformation mechanism of the bump strip employed in thrust bearings. They showed the existence of variable stiffness within the bump strip and provided an experimental proof for his or her theoretical model. The identical authors subsequently extended their theoretical model to see the dynamic stiffness and viscous damping coefficients where small displacement or velocity perturbation was applied to the steady-state operating conditions and results were experimentally verified. Rather than differentiating the load numerically, Peng and Carpino (1993) first introduced small pressure perturbations into the Reynolds equation and subsequently extended the work to incorporate the Coulomb damping effects thanks to frictions within the sub-foil. During a follow-up work, Peng and Carpino (1997) predicted the dynamic coefficients for a broad range of bearing configurations using the finite element approach for modeling both the fluid and structure. They incorporated the results of the bearing number, bearing compliance, sub-foil Coulomb friction, and foil membrane stiffness on the dynamic coefficients of the bearing. Carpino et al. (1994) studied the disarranged effect of the

journal on the performance of the FB. They also reported an analysis of an concentrated foil bearing with provision for the effect of membrane stress resulting from combined bending, membrane and elastic foundation effects.

3. Modeling of Afb

Figure shows the two different types of foil bearings which are tested to provide reliable information on the foil bearing parameter. The top foil structure is modelled as a beam element without the effect of curvature incorporating the bump strip layer as a series of springs which is not connected with each other. The hydrodynamic pressure generations and therefore the film thickness have also reflected, however the deflection of the foil which undergoes by the action of hydrodynamic pressure is by and huge overseen by the character of foil structure considered. The compliant foil structure which provides flexibility to Afb hence are often modeled in several ways from simple model.



Figure 2. Top foil with small gap

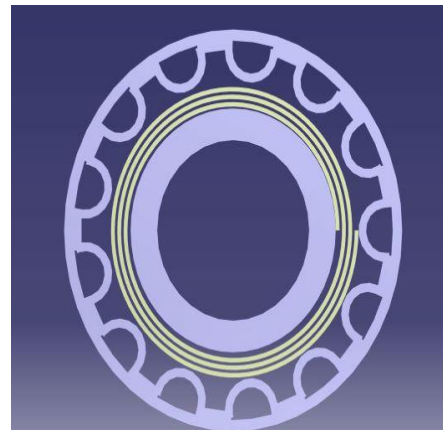


Figure 3. Top foil with spiral spring

This foil bearing design constrains the direction of shaft rotation to only 1 direction. Due to the film created by rotor spinning, the highest foil expands leading to a bigger film thickness than in conventional rigid wall bearings. At set out, the rear of the foil is in reality with the bump foils and therefore the outer side of the foil is in reality with the journal. Because the rotor spins to a sufficiently high speed (i.e. when depart occurs), the highest foil expands as air is dragged into a skinny annular film between the foil and therefore the shaft.

Table 1. Geometry and Operating Condition of AFB

Bearing radius, $R=D/2$	19.05 mm
Bearing length, L	38.1 mm
Bearing radial clearance, C	20 μm
Top foil thickness t_t	101.6 μm
Bump foil thickness, t_b	101.6 μm
Bump pitch, s	4.572 mm
Half bump length, l_0	1.778 mm
Bump foil Young's modulus, E_b	214 GPa
Top foil Young's modulus, E_t	214 GPa
Bump foil Poisson's ratio, ν	0.29

Operating conditions	
Atmospheric pressure, pa	105 N/ m ²
Gas viscosity, η	2.98×10^{-5} N-s/ m ²

3.1 Sub-Structuring

The schematic diagram of the rotor-bearing-support system is shown in the Fig. It can be observed from the Fig. that the rotor-bearing-support system consists of three components, namely the motor, rotor, and axial AFB. The structural characteristics of the isolated individual components, except the axial AFB, is identified in light of experimental data as shown in Fig. All the components are subsequently assembled to form complete rotor-bearing-support system.

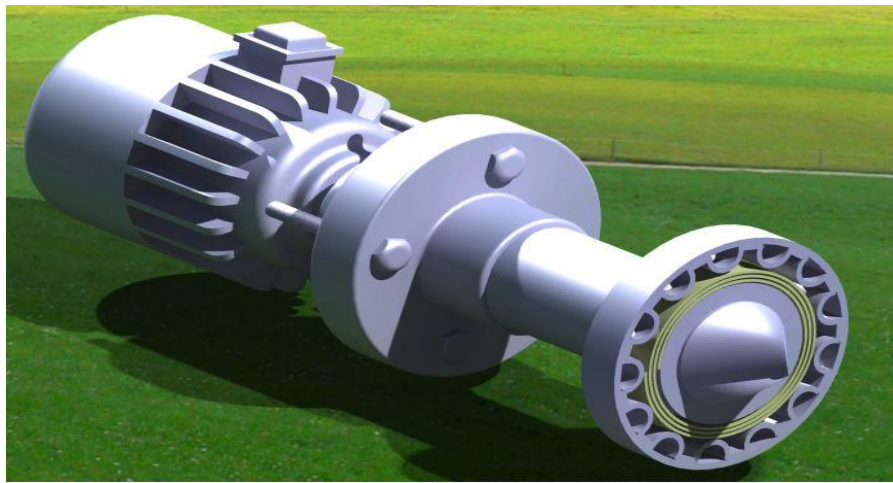


Figure 4. Sub-structuring of Air foil bearing

The torque required to rotate the AFB's supported rotor is greatest when the rotor is to bear with the stator before there's sufficient speed to get a fluid film separating the 2 surfaces. To initiate rotation sufficient torque must be applied to the rotor to beat the sliding frictional forces within the AFBs, the rotor inertia and also the required rotational acceleration. Once sliding begins the torque requirement to sustain rotation drops since the kinematic coefficient of friction is often but the static coefficient of friction. The rotor was designed to own its first typical frequencies above the operating speed of the system. The schematic diagram of rotor is shown in the fig. It can be observed from that the rotor is hollow. The measurements on the rotor were disbursed at three different locations spaced equally along the length of the shaft.

4. Analysis

Air foil bearing analysis is conducted to find the Load carrying capacity of the foil. This analysis is carried out on the two different models. FSI analysis is carried out to find the Pressure exerted on the foil which is also calculated by varying the eccentricity. Static structural analysis is carried to find the Deformation of the foil. During operation of the foil bearing supported machine, the rotation of the rotor generates a pressurized air film that pushes the highest foil out radially and separate the highest foil from the surface of the shaft. The pressure within the air film is proportional to the relative surface velocity between the rotor and top foil. Thus, faster the rotor rotates, the upper the pressure, and also the more is that the load the bearing can support. When the rotor first begins to rotate, the highest foil and also the rotor surface are in touch until the speed increases to some extent where the pressure within the air film is sufficient to push the highest foil far away from the rotor and support its weight. This threshold speed is usually referred as liftoff speed. Likewise, when the rotor slows down to some extent where the speed is insufficient to support the rotor weight, the highest foil and rotor again are available contact.

4.1 Pressure exertion

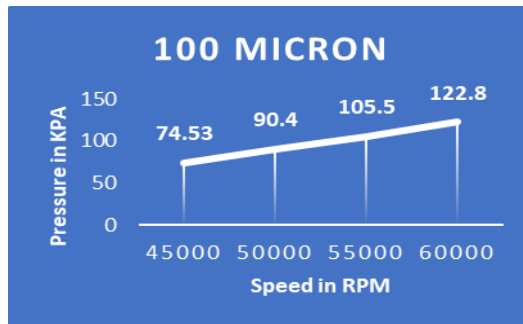


Figure 5 Pressure vs speed (100 micron gap)

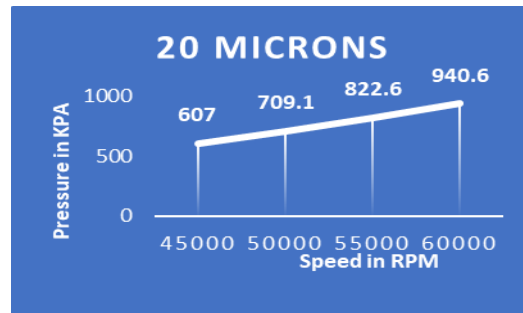


Figure 6 pressure vs speed(20 micron gap)

The graphical representation shows that the pressure exerted for different micron gap created between the rotor and the foil. By increasing the speed of the rotor from 45000 to 60000rpm with 5,000 rpm increments. The graph shows that with increasing the speed of the rotor the load carrying capacity will also increases therefore deformation of the foil will also increases. Similar analysis is carried out with reducing the stiffness of the foil i.e., with 50 & 20 microns.

4. Results

Fluid structure interaction (FSI) & Structural analysis of foil bearing has been carried out to check the deformation & load carrying capacity of the foil. This is carried out by varying the speed of the rotor as well as by varying the stiffness of the top foil. This frequency is used to identify the stiffness of the axial bearing. It can also be observed from that the value of stiffness from 5,000 rpm till 45,000 rpm remains constant and therefore the stiffness decreases. As per the analysis that varying the eccentricity of the foil with increasing the speed of the foil as well as reducing the gap between the rotor & foil we found that the load carrying capacity is more.

Table 2 Comparison between the varying eccentricities of the foil

Speed	Pressure in Kpa (50 Microns)	
	0	50
45000	176.6	255.3
50000	209.9	303
55000	247.4	357.2
60000	287.4	402

The result of analysis shows that the stiffness value is very high during the static position of the rotor and the value of stiffness decreases whereas while increasing the speed of the rotor. It can be observed that during the static condition, there is contact between the rotor and the bearing because of that the stiffness is very high as the rotor pick up the speed fluid medium develops between bearing and rotor which causes the decrease in the stiffness.

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