

DYNAMIC CHARACTERIZATION OF ORIFICE TYPE AEROSTATIC BEARING

Varun. M¹, M. M. M. Patnaik², Arun Kumar. S³, A. Sekar⁴

¹Varun. M, Student, M.Tech (Machine Design), K. S. Institute of Technology, Karnataka, India

²M. M. M. Patnaik, Associate Professor, K. S. Institute of Technology, Karnataka, India

³Arun Kumar. S, Scientist/Engineer-SE, ISRO Satellite Centre (ISAC), Karnataka, India

⁴A. Sekar, Scientist/Engineer-SG, ISRO Satellite Centre (ISAC), Karnataka, India

Abstract -In this work, the dynamic characteristics of rectangular orifice type aerostatic bearing are determined both by FE Analysis and experimentation. Initially, a modal analysis performed in ANSYS (FE tool) gives the resonant frequencies of the elements of the dynamic test set-up. The stiffness obtained from the static test is used to formulate the air column as a spring mass element. To study the dynamic behavior, other DOFs are arrested except in z-axis, by suitably building the experimental set-up. Frequency response curves for varying load conditions are obtained from Hypermesh (FE tool). These results are validated by dynamic experimentation. The test set-up with varying load conditions is vibrated on Electrodynamic shaker with 0.25g amplitude and frequency of 5 to 100 Hz. It is found that in all the cases the first mode resonance started after 25 Hz and predominant resonance occurred beyond 90 Hz. They are in good agreement with the FEA results. Hence it is concluded that the air bearing behaves as stiff member up to a frequency of 25 Hz.

Keywords: Aerostatic bearing, air column, dynamic test set-up, resonance

1. INTRODUCTION

Non-contact bearings of aerostatic type are being extensively used to overcome the problems caused by the conventional bearings. They make use of a very thin film of pressurized air to maintain a non-contact gap. This pressurized air is responsible to take up the loads acting on the bearings. They also possess advantages like running at high speeds, greater load carrying capacity, vibration and shock resistance.

The feeding restrictors like orifice and porous pads play an important role in determining static and dynamic characteristics. Orifices are simple, easy to construct, and result in accurate air film thickness and hence they are used in the aerostatic bearing.

Ye Yixi et al. [1] investigated the dynamic performance of annular orifice aerostatic bearings experimentally. A hammer exciter induced excitations on the bearing and the effect of air film thickness was such that the load carrying capacity decreased at higher clearances showing a nonlinear behavior. Iruikwu. Det al. [2] designed a dynamic loading system using piezo actuators to test the air bearing. At low frequency of 1 Hz, it was seen that both dynamic load and air film height varied sinusoidally with respect to time. It was also concluded that the performance of the bearing system was satisfactory at lower frequencies and amplitude of dynamic force dropped down at higher frequencies.

1.1 Application of Aerostatic Bearings

Air bearings find its application in many areas where frictionless motion coupled with precision in motion is required. One such application in space technology is the measurement of Moment of Inertia of Spacecraft. An air bearing table aids in providing frictionless oscillations, thus reducing damping to determine accurate moment of inertia. Air bearings also find its application in Coordinate Measuring machine (CMM), Dentist drill etc.

1.2 Air column as a spring element

The thin layer of air film present between the bearing surfaces is considered as an equivalent spring damper system. During its operation, the thickness of air film changes due to variation in loading conditions. It is

observed that the response of an air bearing system is identical to the system comprising of a purely elastic spring in series with a viscous damper. Since spring stiffness is equal to the static stiffness and it can be determined using the knowledge of physical dimensions of the system, loading conditions and working pressures.

The stiffness is thus determined by testing the air bearing in static test set up.

1.3 Dynamic study of aerostatic bearings

The dynamic characteristics play a vital role in predicting the overall performance of the structures. The resonant frequency of the bearing determines the speed at which the bearing can be operated in different conditions. One of the methods of inducing dynamic conditions in the structure is by using an Electrodynamics Shaker. It is a mechanical vibration exciter used to induce excitations in the structure. They are contact (Intrusive) type devices extensively used to obtain the structural response in vibration testing. Also, various types of signals such as sine, random and transient inputs can be applied [3].

2. DESIGN OF ELEMENTS FOR DYNAMIC TESTING OF AIR BEARING

An orifice type air bearing is used to study the air bearing characteristics. The experimental apparatus consists of an Aluminium base plate (Supporting surface to the bearing), an Air bearing pad, Spacer blocks-4 (support member for the strip), Shim/Strip-4nos (allows bearing motion only in vertical direction), Balancing mass (for loading and obtaining response), Stud (to fix the balancing masses rigidly). Fig -1 shows the experimental set-up.

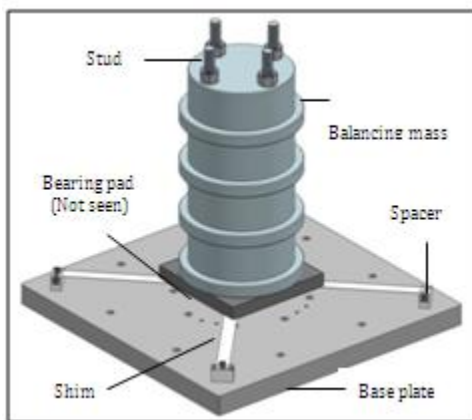


Fig -1: Experimental set up

Initially, a modal analysis using FE tool ANSYS is done on all the above components to make sure that the excitation frequency is away from the natural frequency of the system. The following figures (Fig -2 to 5) give the first mode shapes of the elements of the test set-up. All the DOFs are constrained at the holes.

1. Aluminium base plate

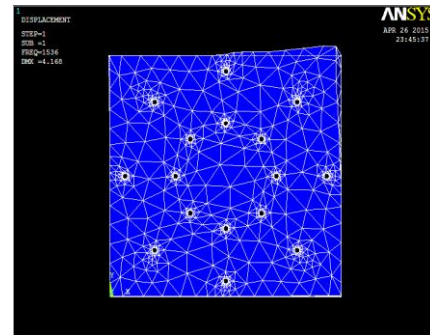


Fig -2: First mode shape of base plate fixed at 6" & 12" PCD

2. Air bearing pad

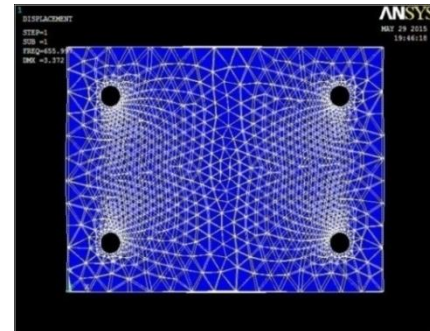


Fig -3: First mode shape of air bearing pad

3. Spacer

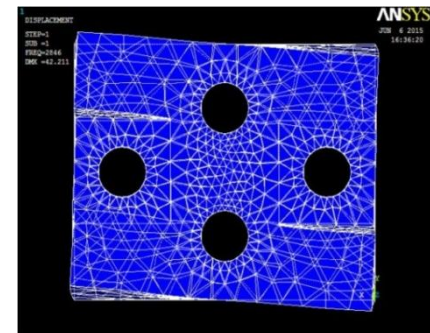


Fig -4: First mode shape of spacer

4. Shim/Strip

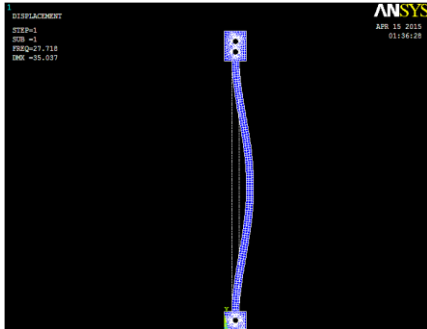


Fig -5: First mode shape of shim

It is concluded that the first mode frequencies of all the above components are greater than 100 Hz.

2.1 Modal analysis of shim assembly with air spring

The shim assembly modeled by taking 4 strips is connected at a common node. The lumped mass is applied at this node. Also, all DOFs except translation in z-direction are constrained. The air column is considered as a spring element and modal analysis is performed.

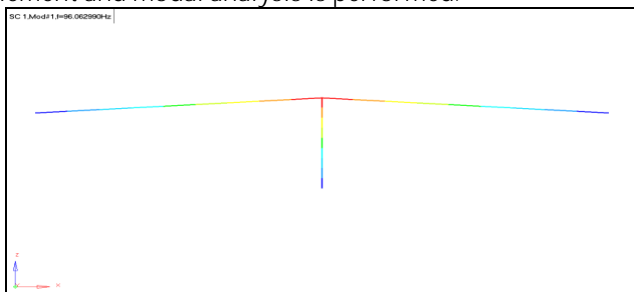


Fig -6: Mode shape of shim assembly with air spring for 1.505kg (Front view)

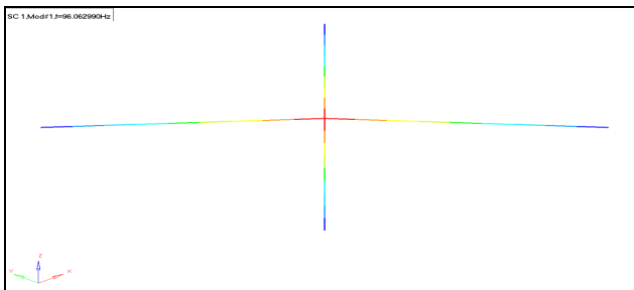


Fig -7: Mode shape of shim assembly with air spring for 1.505kg (Isometric view)

Figures 6 and 7 show the front and isometric views of the first mode shape of shim assembly with air spring for 1.505kg. Further, the analysis is carried out under different loading conditions. The first mode frequency obtained is over 100 Hz and hence the frequency range selected for dynamic experimentation is 5-100 Hz.

3. EXPERIMENTATION AND TESTING

3.1 Static load test

Here, the test arrangement consists of a shim assembly (4 shims placed diagonally) are fixed between the bearing pad and spacers on the Aluminium base plate. Due to this, the bearing motion is constrained in x and y directions and free to move **only in 'z' (vertical) direction**. The air bearing is loaded with balancing masses in steps and the air film gap is measured using a digital dial gauge.

Shims of 5 mm width with thickness of 0.3mm and a length of 200mm is chosen after the other strip dimensions failed to lift the bearing satisfactorily. The loading system comprised of four balancing mass of 4.4 kg each. A mass plate of 1.505kg is used to support the loading system. The film thickness was measured in all the load conditions by digital dial gauge. The static test gave satisfactory results and the corresponding stiffness is determined from the slope of Load vs Film thickness graph. It is further used to formulate the air column as a spring element in FE analysis.

3.2 Dynamic Experimentation

The sine vibration experiment is carried out on an Electro dynamic shaker at ISRO test facility. The dynamic excitations in the air bearing are induced by varying the excitation frequency and keeping the acceleration amplitude (base acceleration) constant i.e. 0.25g. The whole test set-up (Fig -1) is mounted rigidly on the 8 ton capacity Electrodynamic shaker table as shown in fig -8. The control (input) is applied on the aluminium plate, thereby inducing base excitations. Accelerometers are placed over the bearing pad and balancing mass to record the response.

The initial load on the bearing is 19.105 kg at 8 bar pressure. The sinusoidal frequency (input) is swept from 5-100 Hz at acceleration amplitude of 0.25g. The excitations from the table transmit to the base plate and then to the bearing pad via thin layer of air film. The accelerometers recorded the response of the bearing

under these dynamic conditions. The output (response) is obtained both over the mass and over the bearing.

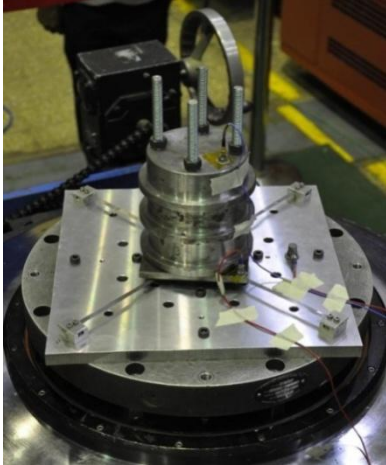


Fig -8: Air bearing system on shaker table

4. RESULTS AND DISCUSSIONS

4.1 FEA Results

The FE model is built considering shim assembly connected at a common (centre) node. The shims are constrained (All DOF) at their end nodes. The air spring is connected at the centre node. All DOFs are constrained at the bottom end of the spring and translation DOF is left free to vibrate at the top end of the spring. The Fig -9 shows Frequency response curve for 1.505kg obtained from Hypermesh.

The input acceleration = 2452.5 mm/sec² (0.25g) at bottom end of spring and the frequency response is obtained at the top end of the spring (centre node).

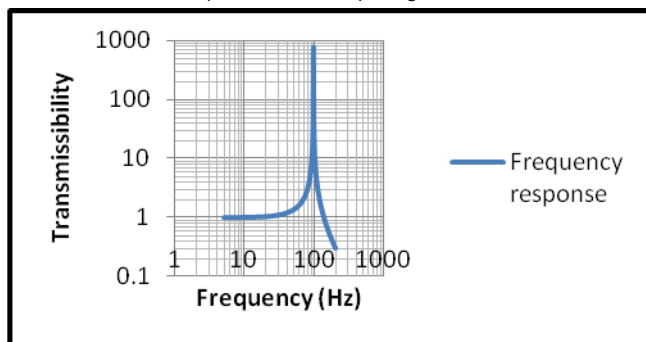


Fig-9: Frequency response curve for 1.505kg

Similarly, frequency response curves are obtained for other loading conditions. It is concluded from the frequency response curves that the resonant frequency occurs in the range of 96 Hz to 160 Hz for masses 1.505kg and 19.105kg respectively.

4.2 Experimental results

For the case of no load conditions (no balancing masses are added to bearing), response is obtained over the bearing only. In this condition the mass of the bearing pad and supporting plate is 1.505 kg. Further, the air bearing is loaded with balancing masses (5.905kg to 19.105kg) and pressure is varied from 2 bar to 8 bar.

Plots of transmissibility versus excitation frequency for the above cases are obtained over the mass and bearing as well. It is seen that both the responses are almost identical. Hence, only response over the bearing is compared with FE results.

For the case of no load condition i.e. Mass=1.505 kg, Supply pressure=2 bar, the control plot is as shown in the fig -10. The response plot over the bearing is obtained as shown in fig -11.

Here, the activity starts from a frequency of 30 Hz and the predominant resonance occurs at 97 Hz. Similarly, for other loading conditions, the activity starts at around 20 Hz and resonance occurs above 90 Hz.

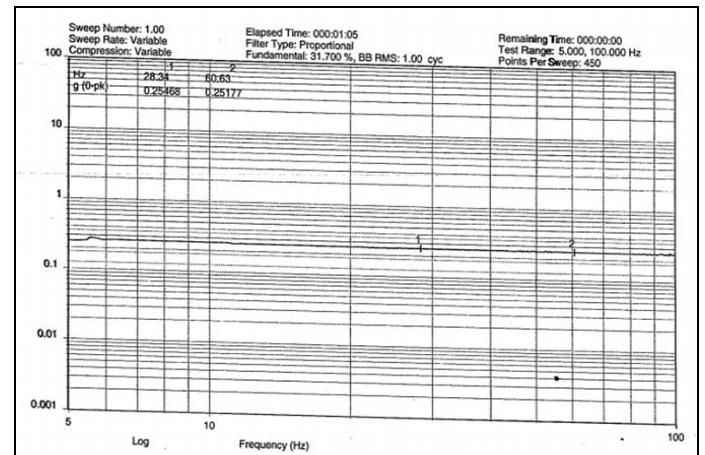


Fig -10: Control plot for 1.505kg & 2 bar supply pressure

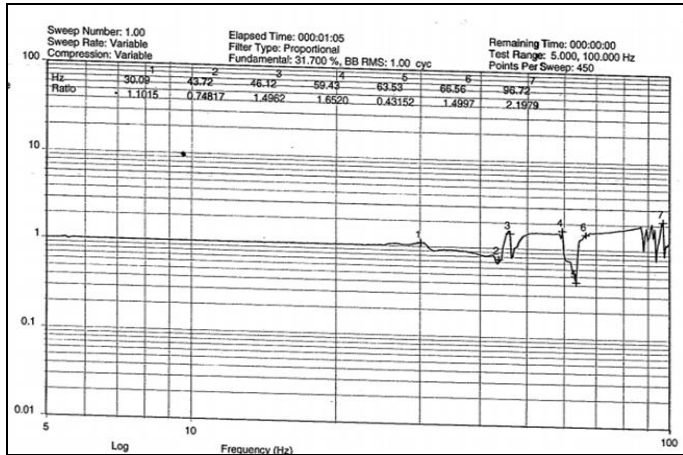


Fig -11: Response plot obtained over the bearing

Similar plots are obtained for other loading conditions (mass varies from 5.905kg-19.105kg) and supply pressure varies from 2-8 bar.

5. CONCLUSIONS

It can be concluded from the above transmissibility plots that the experimental results are in good agreement with the FEA results.

The air bearing behaves as rigid member under dynamic conditions up to 25 Hz and the requirement of the air bearing for its usage under dynamic conditions is met.

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BIOGRAPHIES



Mr. Varun. M is currently a student of M.Tech (Machine Design) in K. S. Institute of Technology, Bangalore, Karnataka, India



Mr. M. M. M. Patnaik is currently an Associate Professor in K. S. Institute of Technology, Bangalore, Karnataka, India. Prior to joining KSIT, he worked in ISRO for 37 years in various capacities. He was General Manager, LEOS, ISRO at the time of his retirement. He was recipient of SAME-ANWEHAK award from Society of Aerospace Manufacturing Engineers (SAME). He has published 16 papers in various journals and conferences.



Mr. Arun Kumar.S is currently a Scientist/Engineer-SE at ISRO Satellite Centre (ISAC), Bangalore, Karnataka, India. He is associated with design and development activities of Ground Support Equipments for Spacecrafts. He is recipient of "YuvaAnweshak" award from Society of Aerospace Manufacturing Engineers (SAME) and has authored 6 Technical Papers.



Mr. A. Sekar is currently a Scientist/Engineer-SG at ISRO Satellite Centre (ISAC), Bangalore, Karnataka, India. He is currently heading the Design and Development Section of SIG Group. He has more than 30 years of experience in Mechanical design area. He is recipient of ASI Gold Medal. He has authored more than 20 technical papers.