iJRASET
International Journal For Research in
Applied Science and Engineering Technology

# INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 7 Issue: V Month of publication: May 2019

DOI: https://doi.org/10.22214/ijraset.2019.5017

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.177
Volume 7 Issue V, May 2019- Available at www.ijraset.com

# Effect of Clearance and L/D Ratio on Rotor Dynamic Stability of Journal Bearing Assembly

Iranna M Biradar[1], Venkatesh M. Kulkarni[2]
[1]Department of Mechanical Engineering, Faculty of Engineering, Baze University, Abuja, FCT, Nigeria.
[2]Dept. of Mechanical Engineering, Visvesvaraya Technological University, Centre for Postgraduate Studies, Kalaburagi-585105, Karnataka, India.

***Abstract:* Rotor dynamic performance for journal bearing is analyzed using finite element approach. Calculations are carried out to find the stiffness and damping coefficients for the considered clearance value and L/D ratio. Standard bearing formulae based on Somerfield number is applied to find the dynamic coefficients. Finite element analysis based on spring and beam elements is used to find the whirling speed for different speeds of the rotor. Here elastic support condition is considered due to the change of stiffness of the bearing for the given applied speed. The results shows higher L/D ratio has better vibrational characters compared to the lesser L/D ratio bearing assembly. But higher clearance is reducing the fundamental frequency of the system which is not desirable for better dynamic stability.***
***Keywords: Journal Bearing, rotor dynamics, stability, whirl.***

## I. INTRODUCTION

A bearing is a machine element that is used to reduce the friction between moving parts by constraining the relative motion to only the desired motion. Hydrodynamic Journal bearings are a very common class of bearings that are used in the rotating machinery. They are mainly used in automotive, pump, hydro/gas/steam turbines for power plants and processing equipments. The stability of the rotor bearing system is very important for proper functioning and life of the machinery. Very early work done by Jeffcott [1] has given the foundation for study of turbo machinery based on journal bearings. The procedure involves complex mathematics and cumbersome calculations based on many design parameters. With the advances in the computational mechanics with the support of finite element method based software, it is easy and faster to find solutions for problems of turbo-machinery compared to the older method of prototype built up and elaborate testing procedures.

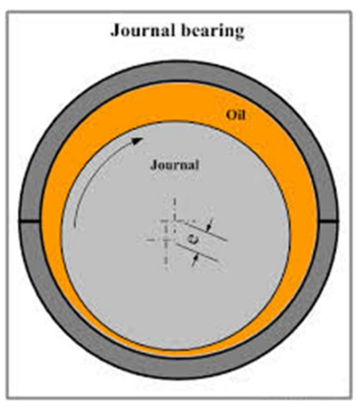



Fig.1: Journal Bearing Assembly

Much work has been on the turbo machinery and fluid structural interaction problems using theoretical, experimental and numerical methods. Always there is quest for better and faster solutions. In the present work, macros are created to find the effect of L/D and clearance ratio on the turbo machinery performance. Some of the literature available on the work is as follows.

Nelson and Mcvaugh [2] have worked on a finite element model comprising rotor bearing systems. It mainly consists of rigid disc and flexible bearing systems. The shaft model has varying cross section to accommodate various mountings on the shaft system. The analysis includes applied axial loads, gyroscopic moments etc.

Lumped massing is considered for analysis.

Khulief and Mohiuddin [3] applied model reduction scheme to solve rotor dynamic problem using rotating frame reference. Both damped and undamped extraction methods are applied to find the solution to analyze the performance of the system. Model reduction scheme which is used for saving the memory and faster execution is applied to find the dynamic response of the system. They concluded that planar model representation has advantage over complex mode representation.

Taplak and Mehmet Parlak [4] also used finite element analysis for rotor dynamic analysis. Analytical approach is very difficult to apply as the model is complex for representation and so finite element analysis is applied to find the dynamic response of the system. He concluded that finite element method saves the time and expenses along with faster execution capabilities with feasible solutions. A program *dynrot*, a finite element code for rotor dynamic analysis, is used to find the rotor dynamic performance of the system which is based on macro programming and design language (DL) codes. He observed that low imbalance in the system provides higher stability for the system.

Miranda and Faria [5] have worked on flexible journal bearings based on Fluid structural interaction along with finite element analysis to find the actual behavior of the systems. Fluid film damping effect is also simulated. Mainly the concentration is given on obtaining Eigen frequencies and the shaft model is based on Timoshenko theory. Governing equations are developed based on Galerkin approach. They have concluded that appropriate configuration for journal bearing is very essential to find the best possible results.

Mutalikdesai *et. al.* [6] has extended the analysis with internal damping. Generally internal damping is considered as 2% of overall damping. Due to internal damping, a damping force is generated which will change the orbit of rotation which is mainly eccentric and elliptical. The force generated is tangential to orbit of rotation of the shaft. Higher speed results to higher force and increase of imbalance of force which destabilize the rotor system. Euler Bernoulli based beam elements are considered for analysis. They observed that reducing damping effect in the forward whirl and increased damping effect during backward whirl.

Wang *et.al.* [7] have applied finite element analysis on rotor compressor. The main elements of the system are piston and bearing. Inertia, contact and gas forces are applied periodically to find the dynamic response. The magnitude of these forces is function of spin speed. As the speed increases, the imbalance in the system is more and vibration observed is more in the system. So a three dimensional finite element analysis is applied to find the nature of vibration. They observed that, all type of vibration (tensional, lateral and axial) exists in the structure. They concluded that lateral and torsional vibrations will increase with the speed increase, but the longitudinal vibration does not effected by the increase of speed.

Rosyid *et. al.* [8] have applied component mode synthesis or sub-structuring technique which is best suitable for large models to find the dynamic response of the systems. The sub-structuring is mainly based on super-element creation technique which reduces the solution time, memory requirements etc. They have made comparison between full model and reduced model with super element creation. They have also observed that there is some error with the results obtained through component mode synthesis compared to the full model.

## II. SOLUTION METHODOLOGY

The geometry of the hydrodynamic bearing selected is shown in Figure 2 and the L/D ratio is varied in finite element method to find the performance of the shaft model. Initially calculations are carried out to find the direct stiffness coefficients and later cross coefficients for the given configuration of design specifications. The shaft model is built using *one dimensional* modeling concept and later elastic support condition of the bearing is provided through the bearing element *combi214*. Campbell analysis is carried out to find instability changes with the support conditions. Further, the modal analysis is carried out to find the natural frequencies of the configuration.

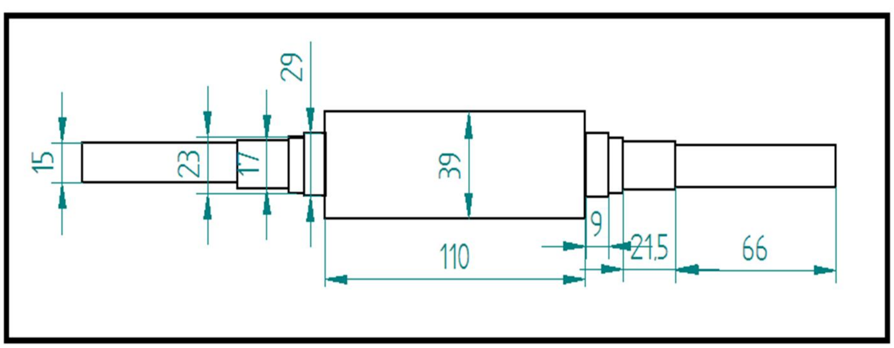



Fig.2: Geometry of the hydrodynamic journal bearing shaft ( dimensions in mm)

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.177
Volume 7 Issue V, May 2019- Available at www.ijraset.com

## III. MATHEMATICAL MODELING

Mathematical modeling is the critical stage of design and analysis of the system. The modeling equations show important parameters of influence on the system.

$$K_{xx} = \frac{F_0}{C} \times \frac{f_{r0}(f^2_{ro} + 1 + 2\varepsilon^2)}{\varepsilon(1-\varepsilon^2)} \qquad C_{xx} = \frac{F_0}{C\omega} \times \frac{2 f_{t0}\{(2+\varepsilon^2) f^2_{ro} + 1 - \varepsilon^2\}}{\varepsilon(1-\varepsilon^2)}$$

$$K_{yy} = \frac{F_0}{C} \times \frac{f_{r0}(f^2_{to} + 1 - \varepsilon^2)}{\varepsilon(1-\varepsilon^2)} \qquad C_{yy} = \frac{F_0}{C\omega} \times \frac{2 f_{t0}\{(2+\varepsilon^2) f^2_{to} - 1 + \varepsilon^2\}}{\varepsilon(1-\varepsilon^2)}$$

$$K_{xy} = \frac{F_0}{C} \times \frac{f_{t0}(f^2_{ro} + 1 + 2\varepsilon^2)}{\varepsilon(1-\varepsilon^2)} \qquad C_{xy} = \frac{F_0}{C\omega} \times \frac{2 f_{r0}\{(2+\varepsilon^2) f^2_{to} - 1 + \varepsilon^2\}}{\varepsilon(1-\varepsilon^2)}$$

$$K_{yx} = \frac{F_0}{C} \times \frac{f_{t0}(f^2_{ro} - 1 + \varepsilon^2)}{\varepsilon(1-\varepsilon^2)} \qquad C_{yx} = \frac{F_0}{C\omega} \times \frac{2 f_{r0}\{(2+\varepsilon^2) f^2_{to} - 1 + \varepsilon^2\}}{\varepsilon(1-\varepsilon^2)}$$

Fig.3: Formulae for Stiffness and damping Coefficients

The figure 3 shows formulae used for calculating the dynamic coefficients. Here 'K' represents stiffness coefficients, 'C' represents damping coefficients. The subscript represent direction of the coefficient. 'F' represents magnitude of force, 'ε' represents eccentricity ratio. '$f_{ro}$' and '$f_{to}$' represents additional dynamic coefficients.

Table 1: Details of hydrodynamic journal bearing

| Details | Magnitude |
|---|---|
| Journal Diameter (m) | 0.04 |
| Length of Bearing (m) | 1 |
| Radial Clearance (m) | 0.0001 |
| Speed ( rpm) | 500 to 11000 |
| Oil | SAE 20 |
| Viscosity (Pa-s) | 0.0981 |
| L/D ratio | 0.5 - 1 |

Table 2: Eccentricity & Sommerfeld Numbers

| Details (for 300 rpm) | | | |
|---|---|---|---|
| Clearance | L/D ratio | Eccentricity Ratio(ε) | Sommerfeld Number (σ) |
| 0.1 | 0.5 | 0.1665 | 1.79 |
| 0.1 | 0.75 | 0.0523 | 6.05 |
| 0.1 | 1 | 0.0222 | 14.35 |
| 0.15 | 0.5 | 0.3147 | 0.8 |
| 0.15 | 0.75 | 0.1148 | 2.69 |
| 0.15 | 1 | 0.0496 | 6.38 |
| 0.2 | 0.5 | 0.4381 | 0.45 |
| 0.2 | 0.75 | 0.1928 | 1.51 |
| 0.2 | 1 | 0.0876 | 3.59 |

Table 3: Dynamic Coefficeients(K in N/m, C in Ns/m)

| Cases | Kxx | Kyy | Kxy | Kyx | Cxx | Cyy | Cxy | Cyx |
|---|---|---|---|---|---|---|---|---|
| 1 | 94202 | 164935 | 437940 | -369694 | 26920 | 24337 | 5235 | 5235 |
| 2 | 25221 | 49761 | 378039 | -371651 | 23977 | 23735 | 1583 | 1583 |
| 3 | 10565 | 21109 | 373499 | -372349 | 23761 | 23858 | 674 | 674 |
| 4 | 276214 | 359040 | 653992 | -362487 | 37639 | 27519 | 11622 | 11622 |
| 5 | 59266 | 111185 | 402583 | -371098 | 25187 | 24048 | 3540 | 3540 |
| 6 | 23862 | 47120 | 377203 | -371467 | 23932 | 23664 | 1497 | 1497 |
| 7 | 656363 | 605287 | 1057219 | -335595 | 57509 | 31707 | 19685 | 19685 |
| 8 | 115626 | 193804 | 461947 | -368650 | 28093 | 24513 | 6135 | 6135 |
| 9 | 43874 | 84953 | 391657 | -373341 | 24681 | 24528 | 2747 | 2747 |

Table 1, 2, and 3 shows considered journal bearing specifications and calculated Sommerfeld number, eccentricity ratio and dynamic coefficients.

Governing Differential Equation for Rotor Dynamics is given by equation (1) and is as below.

$$[M]\{\ddot{z}\} + ([C] + [Cgyro])\{\dot{z}\} + ([K] + [H])\{z\} = \{f\} \quad \text{------ eq.(1)}$$

Where [M] - Mass Matrix, [C] - Symmetric Damping Matrix , [$C_{gyro}$ ] – Skew Symmetric gyroscopic matrix, [K] - Stiffness Matrix, [H] - gyroscopic matrix of deflection , {Z}- Generalized coordinate vector and {f}- External Force Vector respectively.

## IV. RESULTS

Analysis has been carried out by varying the clearance, L/D ratio and Campbell analysis is carried out to find the whirl speeds. The results are as follows.

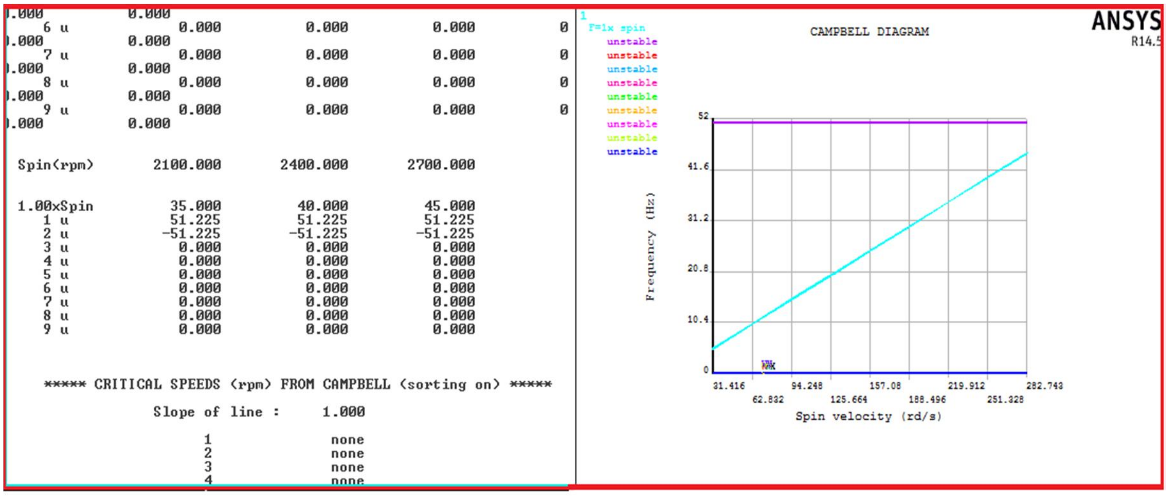



Fig. 4: Response Analysis for Clearance: 0.1 and L/D ratio= 0.5

The figure 4 shows whirl speed of 51.225 rpm for the given journal bearing configuration. Even Campbell diagram shows instability in the plot with 9 speed intervals. Left side block pictures shows unstable speed value and right side shows instability in the system during the spin.

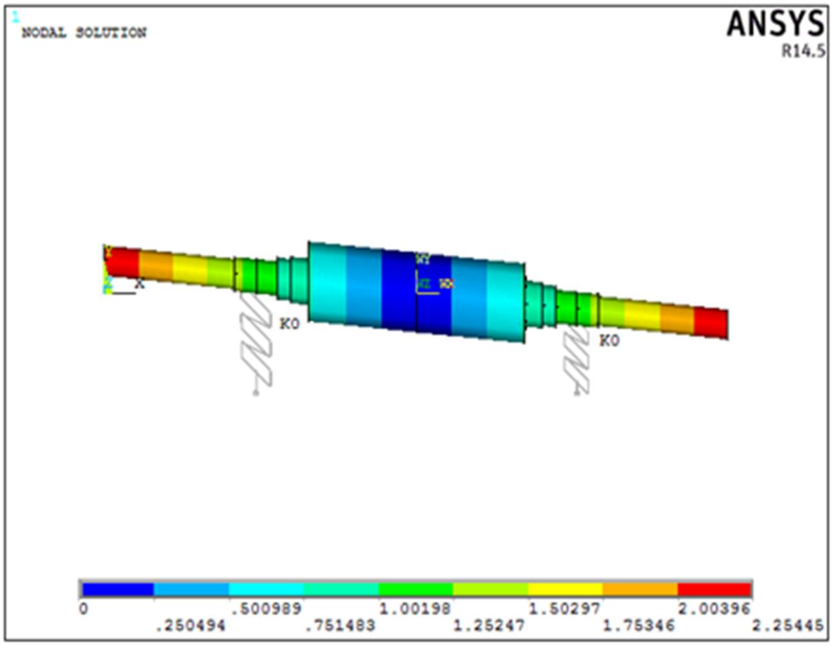



Fig. 5: Response for first fundamental frequency

The figure 5 shows mode shape corresponding to the first natural frequency of 41.232 Hz. The ends have more deformation compared to the center of the shaft. It shows a lateral mode of deformation. The mode shape picture helps in providing the proper constraints for the configuration in the real environment.

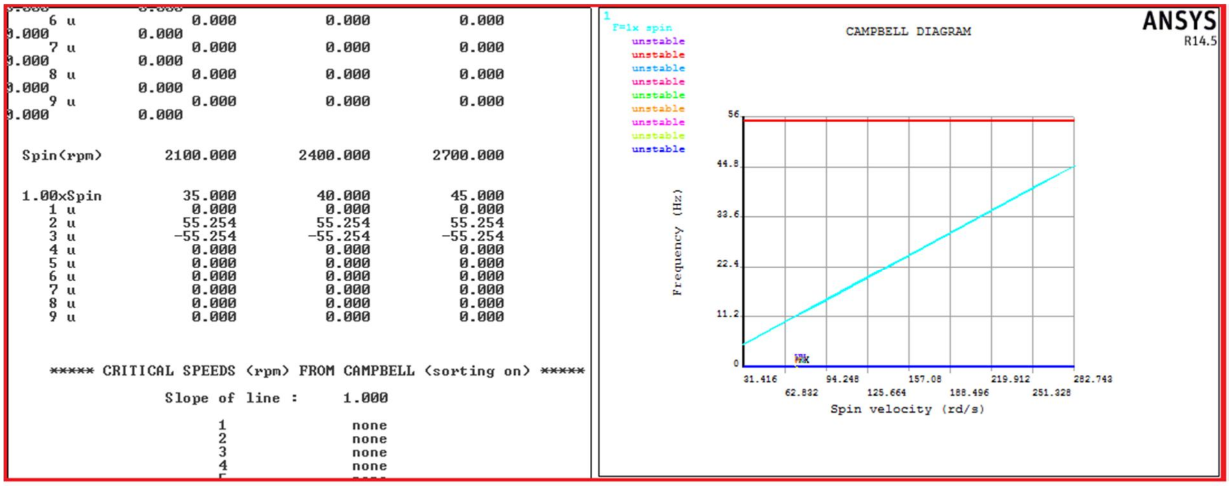



Fig. 6: Whirl Speed for Clearance 0.15 and L/D ratio=0.5

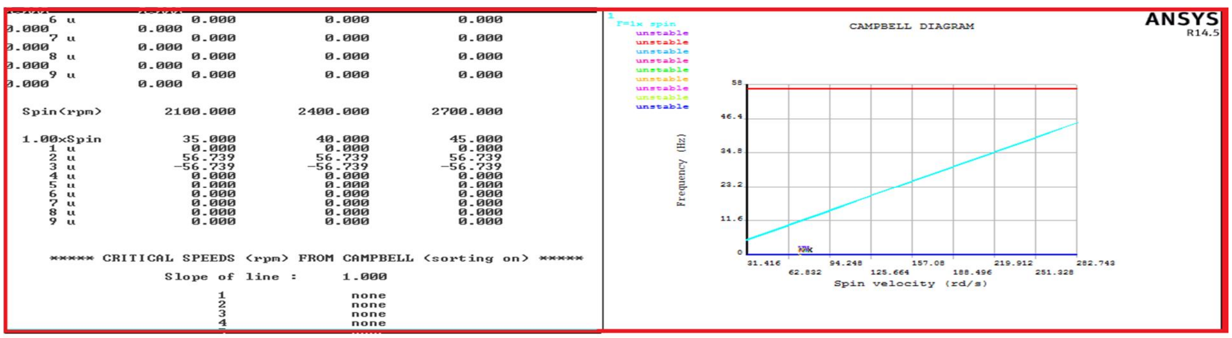



Fig.7: Whirl Speed for Clearance 0.2 and L/D ratio=0.5

The figure7 shows whirl speed for the given clearance and L/D ratio of the journal bearing. The whirl values are increasing with the increase in the L/D ratio. So height length of the bearing is desirable for stable configuration of the journal bearing.

| Clearance (mm) | L/D ratio | Critical speed (rpm) |
|---|---|---|
| 0.1 | 0.5 | 51.225 |
| | 0.75 | 55.254 |
| | 1 | 56.739 |
| 0.15 | 0.5 | 48.517 |
| | 0.75 | 52.8 |
| | 1 | 55.363 |
| 0.2 | 0.5 | 45.969 |
| | 0.75 | 50.563 |
| | 1 | 53.911 |

Table 4: Clearance, L/D Ratio and Critical Speeds

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.177
Volume 7 Issue V, May 2019- Available at www.ijraset.com

The table 4 shows increasing critical speed values with L/D ratio but decreasing critical speed with increase in the clearance. So increase in the length of bearing is a desirable feature along with the reduction of clearance. Further modal analysis is carried out to find the first three natural frequencies for change in clearance and L/D ratios.

| Clearance (mm) | L/D ratio | Natural Frequency 1 | Natural Frequency 2 | Natural Frequency 3 |
|---|---|---|---|---|
| 0.1 | 0.5 | 41.232 | 54.522 | 105.45 |
| | 0.75 | 21.335 | 29.962 | 57.962 |
| | 1 | 13.809 | 19.517 | 37.758 |
| 0.15 | 0.5 | 70.601 | 80.379 | 155.38 |
| | 0.75 | 32.705 | 44.775 | 86.607 |
| | 1 | 20.752 | 29.156 | 56.404 |
| 0.2 | 0.5 | 104.26 | 108.82 | 201.43 |
| | 0.75 | 45.68 | 59.095 | 114.28 |
| | 1 | 28.139 | 39.143 | 75.717 |

Table 5: Comparative first Natural frequencies (Hz) for various Clearance and L/D ratios

The table 5 shows increasing order of the natural frequencies for all the configurations. It is observed that natural frequencies reduce with the increase in the L/D ratio. Also with the increase in the clearance the natural frequencies are increasing which is also a desirable requirement of the journal bearing.


Fig. 8: Critical Speed (rpm) Vs L/D ratio (Clearance: 0.1mm)

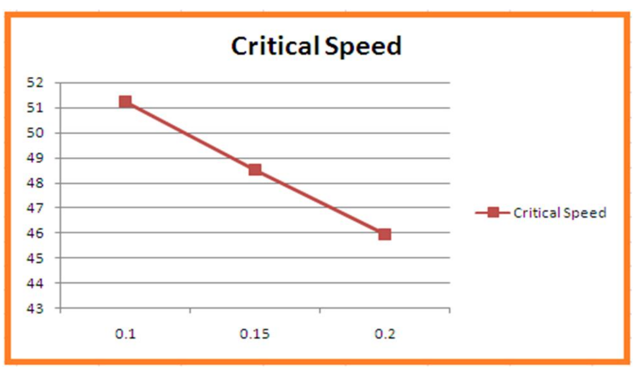



Fig. 9: Critical Speed (rpm) Vs Clearance ( L/D ratio = 0.5)

Figure 8 shows proportionate relation of L/D ratio and critical speed which is an important observation for stable design of bearings. Similarly inverse relation of clearance with critical speeds can be observed in Figure 9. So optimum clearance is very important parameter for stable design of bearings.

## V. CONCLUSIONS

Sommerfeld number and eccentricity ratio are calculated for the given journal bearing specification. Based on these two parameters other dynamic coefficients are calculated from the standard formulas. A shaft system is built using one dimensional beam elements and elastic support of the bearing is provided with combin214 element. Totally 8 dynamic coefficients are calculated (4 stiffness and 4 damping coefficients) and spring models are built based on these parameters. The analysis is carried out between 300 rpm to 10000 rpm. The analysis results are represented with required number of suitable pictures. The results show lesser clearance is desirable for increased whirl speed. Also higher L/D ratio is desirable for better dynamic stability of the system. Even though natural frequencies are increasing with higher clearance, resonance can be avoided by not allowing the system to rotate at these particular speeds. So finite element analysis can be applied to find the optimum configuration for better dynamic stability of the system.

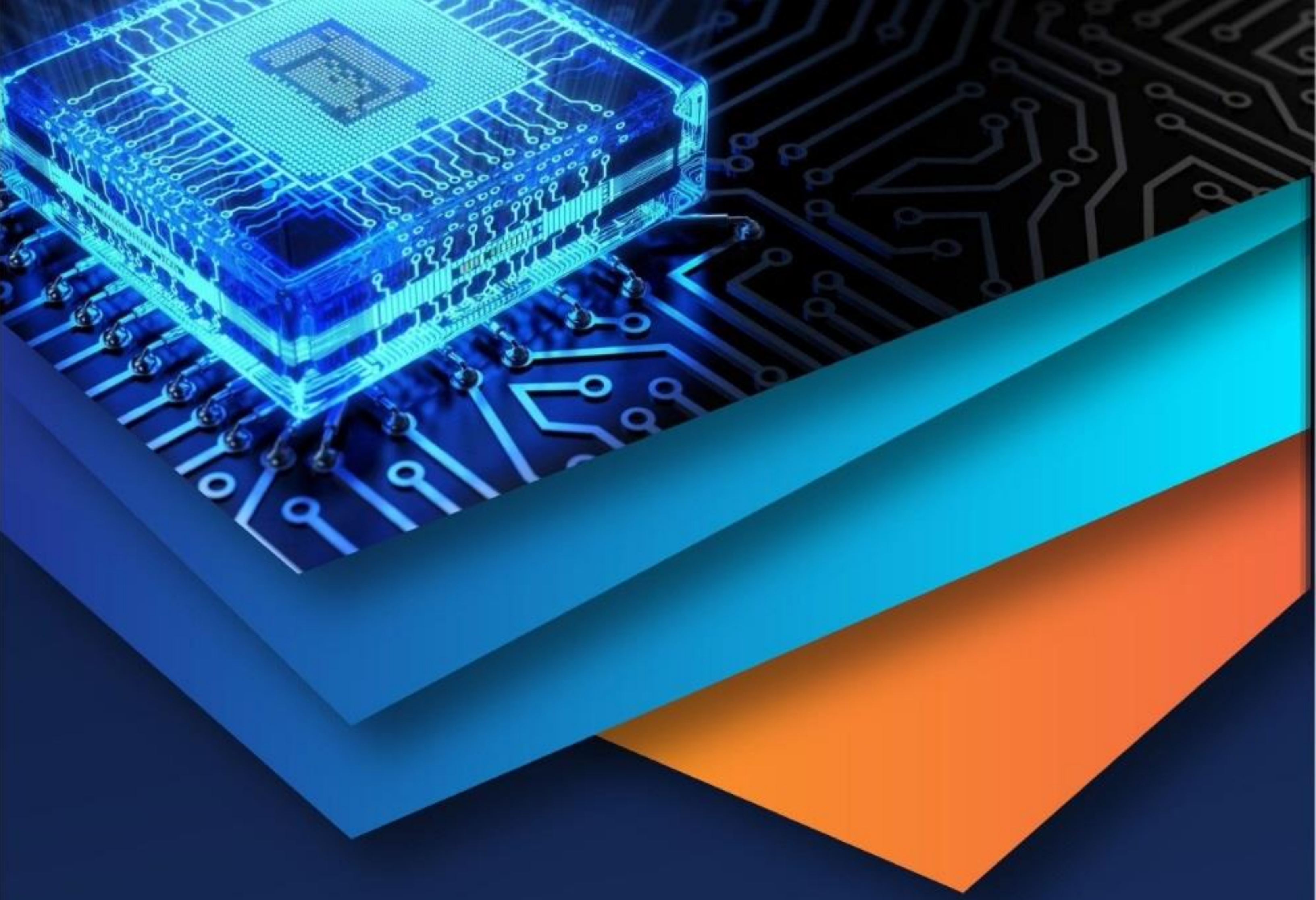
## VI. ACKNOWLEDGEMENTS

This research did not receive any specific grant from funding agencies in the public, commercial, or not-for-profit sectors.

## REFERENCES

[1] H.H.Jeffcott,XXVII, The lateral vibration of loaded shafts in the neighbourhood of awhirling speed—the effect of want of balance, Philos.Mag.Ser.637(219)(1919)304–314, http://dx.doi.org/10.1080/14786440308635889.
[2] H.D. Nelson & J.M. Mcvaugh," Dynamics of Rotor Bearing System using Finite Element", "Journal of Engineering for Industry, pp 593-600,1976.
[3] Y.A. Khulief & M.A. Mohiuddin," Dynamic Analysis of Rotors using Modal reduction", Finite element in analysis and Design, pp41-55,1997.
[4] Hamdi Taplak, Mehmet Parlak, " Evaluation of Gas Turbine Rotor Dyanmic Analysis using Finte element Method", Measurement 45, pp.1089-1097,2012.
[5] W.M. Miranda & M.C. Faria, " Finite Element Method Applied ot the Eigen Value Analysis of Flexible Rotors Supported by Journal Bearings", Scientific Research Engg. , pp 127-137, 2014.
[6] P. Mutalikdesai, S.Candraker and H.Roy, " Modal Analysis of Damped Rotor using Finite Element Method International Journal of Engineering Science Invention (IJESI), pp.98-103.
[7] Wang Z., Yu F Liu, Feng and Q Tan, "Dynamic Analyis for the Rotor Journal Bearing system of a Variable Speed Rotor Compressor", International Journal of Refrigeration, pp 1938-1950, April 2013.
[8] Rosyid A, Madany M.E., Alata M. " Reduction of Rotor Bearing Support Finite Element Model through Substructing", International Scholarly and Scientific Research and Innovation, Vol7, pp 1361-1368, 2013




10.22214/IJRASET

45.98

IMPACT FACTOR: 7.129

IMPACT FACTOR: 7.429


# INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY