Evaluation and Vibration Analysis of Ball Bearing of Coffee Beans Processing Machinaries using Finite Element Model Simulation

Suresh K. G, Dr. C. T. Jayadeva,

Abstract—Ball and roller bearings, generally called rolling bearings, are among the commonly used components in machineries, since they provide relative positioning and rotational freedom while transmitting a load between two structures, usually a shaft and housing. In various applications, these bearings are considered as critical mechanical component since defect in these components may lead to malfunction, even catastrophic failure. The present work is focused on the development of mathematical models and solution algorithms for the analysis of porous ball bearing in coffee bean processing machinaries. Vibration analysis is one of the most established methods used to evaluate bearings. In this study, finite element model simulation is developed to analyse the vibration of ball bearings and finding initial modes and corresponding natural frequency.

Index Terms— Ball Bearing, Coffee Bean Machinaries, FEM, Natural Frequency, Modal Analysis, 6 Modes, Vibration Analysis.

1 INTRODUCTION

Bearing is a machine element that constrains relative motion and reduces friction between moving parts to only the desired motion. The design of the bearing may, for example, provide for free linear movement of the moving part or for free rotation around a fixed axis; or, it may prevent a motion by controlling the vectors of normal forces that bear on the moving parts. Many bearings also facilitate the desired motion as much as possible, such as by minimizing friction. Bearings are classified broadly according to the type of operation, the motions allowed, or to the directions of the loads (forces) applied to the parts. A bearing being a machine element allows one part to bear (i.e., to support) another. The simplest bearings are bearing surfaces, cut or formed into a part, with varying degrees of control over the form, size, roughness and location of the surface. Other bearings are separate devices installed into a machine or machine part. The most sophisticated bearings for the most demanding applications are very precise devices; their manufacture requires some of the highest standards of current technology. A plain bearing (in railroading sometimes called a solid bearing) is the simplest type of bearing, comprising just a bearing surface and no rolling elements. Therefore the ball (i.e., the part of the shaft in contact with the bearing) slides over the bearing surface. The simplest example of a plain bearing is a shaft rotating in a hole. The present research is carried out on Ball bearings of coffee bean processing machinaries. In the previous experimental work, the vibration is measured using a hand held RIOVIBRO meter. Using a hand held RIOVIBRO vibration meter, overall vibration data has been collected in three directions (vertical, horizontal and axial) of coffee bean processing machinaries for bearing points 1, 2 &3. By using those data's a Finite Element Model Simulation is developed to analyse the vibration of ball bearings and finding intial modes and corresponding natural frequency.

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2 .REVIEW OF LITERATURE

The research efforts and direction related to the present work will be identified through the following literature survey.

2.1 N.S.R. Apandi, work presents a numerical approach on the frequency characteristics of new and defected bearings for the increasing rotational frequency of the shaft. The simulated vibrational response of the bearing with different local faults was used to test the suitability of the envelope-analysis technique and the continuous wavelet transformation was used for the bearing fault identification and classification. A 3D model of bearing system with 0.5 mm artificially defects including outer and inner race was modelled by using CATIA software. The numerical simulation was completed by employing ANSYS WORKBENCH 16.0. The simulation result shows the existence of significant and non-synchronous peaks which represent the new and defected bearing defaults with the frequency characteristics of the system.

2.2 J. S. Tripathi, Dr. J. F. Agrawal considering the radial vibrations of rigid shaft supported ball bearings are studied. In the analytical formulation the contacts between the balls and the inner and outer races are considered as nonlinear springs, whose stiffness are obtained by using the Hertzian elastic contact deformation theory.For perfect bearings, vibrations occur at the ball passage frequency. All results are presented in the form of Fast Fourier Transformations (FFT). The experimental validation of a mathematical ball bearing model with localized defects is presented here. The bearing is considered as a mass – spring – damper system, considering each rolling element as a contact spring – damper pair, based on Hertz equations for contact deformation, moving along the inner and outer raceways. In accordance with the obtained results, in this work a bearing model is validated with a purpose built test bench.

2.3 Mr. Shinde S. S, presented the Effectiveness of transient analysis of the finite dement bearing model to simulate the vibration signal emanating from ball bearing with faults is presented in this work. It is difficult to identify the ball bearing defect either in frequency spectrum or time domain when the defect is at incipient stage. Further, it is difficult to experimentally obtain vibration signals from bearing having fault at incipient stage. Thus, need for accurate simulation of ball bearing fault at incipient stage is considered essential. A Computer Aided Design (CAD) model of a ball bearing having a minor crack in outer-race was created using commercially available software. It was shown that identification of ball bearing defect in frequency spectrum is difficult. The results were validated with experimental results

2.4 Matej Tadina, Considering an improved bearing model is developed in order to investigate the vibrations of a ball bearing during run-up. The numerical bearing model was developed with the assumptions that the inner race has only 2DOF and that the outer race is deformable in the radial direction, and is modelled with finite elements. The centrifugal load effect and the radial clearance are taken into account. The contact force for the balls is described by a nonlinear Hertzian contact deformation. Various surface defects due to local deformations are introduced into the developed model. The detailed geometry of the local defects is modelled as an impressed ellipsoid on the races and as a flattened sphere for the rolling balls. With the developed bearing model the transmission path of the bearing housing can be taken into account, since the outer ring can be coupled with the FE model of the housing. The obtained equations of motion were solved numerically with a modified Newark time-integration method for the increasing rotational frequency of the shaft. The simulated vibrational response of the bearing with different local faults was used to test the suitability of the envelope-analysis technique and the continuous wavelet transformation was used for the bearing fault identification and classification.

3 OBJECTIVE OF THE PRESENT WORK

- The present work is focused on the development of mathematical models and solution algorithms for the analysis of porous ball bearing.
- The numerical simulation for analysis of porous ball bearing in coffee bean processing machinaries will be done by using finite element technique.
- The effects of eccentricity, ball speed and porous material properties on static and dynamic characteristics of porous ball bearing are studied.
- Finding initial modes and corresponding natural frequency

4 FINITE ELEMENT FORMULATIONS:

The lubricant flow field has been discretized using four nodded quadrilateral isoparametric elements. Using the Galerkins techniques and orthogonality condition for equation (2) and following the usual assembly procedure for all the elements, the global system equation for entire lubricant flow field is expressed

$$\left[\overline{F}\right]\left\{\overline{P}\right\}=\left\{\overline{Q}\right\}+\left\{\overline{R}_h\right\}+\dot{\overline{X}}_j\left\{\overline{R}_{X_j}\right\}+\dot{\overline{Z}}_j\left\{\overline{R}_{Z_j}\right\} \quad \text{------ } 2$$

For an e[th] element, the elements of the above matrices are defined as follows:

$$\overline{F}^e_{ij}=\iint_{A_e}\left(\frac{\overline{h}^3}{12}+\frac{\overline{k}\overline{h}}{2}\right)\left[\frac{\partial N_i}{\partial\alpha}\frac{\partial N_j}{\partial\alpha}+\frac{\partial N_i}{\partial\beta}\frac{\partial N_j}{\partial\beta}\right]d\alpha d\beta$$

$$\overline{Q}^e{}_i=\int_{\Gamma_e}\left[\left(\frac{\overline{h}^3}{12}+\frac{\overline{k}\overline{h}}{2}\right)\left(\frac{\partial\overline{p}}{\partial\alpha}+\frac{\partial\overline{p}}{\partial\beta}\right)-\frac{\Omega}{2}\overline{h}\right]N_i d\Gamma_e$$

$$\overline{R}^e{}_{Hi}=\frac{\Omega}{2}\iint_{A_e}\overline{h}\frac{\partial N_i}{\partial\alpha}d\alpha d\beta$$

$$\overline{R}^e{}_{Xji}=\Omega\iint_{A_e}N_i\cos\alpha d\alpha d\beta$$

$$\overline{R}^e{}_{Zji}=\Omega\iint_{A_e}N_i\sin\alpha d\alpha d\beta$$

5 FINITE ELEMENT ANALYSES

Finite element analysis is a numerical technique, which provides nearly accurate solution to the complex problems. It divides the complex structure into the finite number of the elements; this process is called as the discretization or meshing. This makes the structure to be analysed in the easier way. Even the complex shaped structure and different engineering domain problem can be solved. Differential equation and integral equation are used as the mathematical models for representing the physical problems mathematically. Complex geometry and boundary condition of particular application makes the solution problem very difficult to solve by analytical method. In this case the finite element methods come to a rescue for obtaining the solution by solving the governing equation.

While developing and designing a product, governing equations and fundamental formulas are used for analysing basic important components of the structure. To validate the results obtained, experimental testing is conducted for every engineering parameter. For extensive testing of product involves building a prototype which is expensive for a complex model, so use of FEM in analysis reduces considerable amount of time and capital for any design and manufacturer.

The following steps are followed in finite element method.

- The principle used in the FEM is the discretization, where the whole structure is divided into finite sized elements of simple shape for ease of analysis.
- Variation of the displacement of the element is determined by polynomial shape function and nodal displacement.
- Relation between the stress-strain and strain displacement is developed in terms of the unknown nodal displacement.
- The equilibrium equation is assembled (3) in matrix form, which is easier to program and solve in FEA software.
- Appropriate boundary condition is applied.
- The nodal displacements are found by solving the matrix equation.

- Knowing the nodal displacement, the required stress and strain can be calculated.

6, Results and Discussion

The mathematical models developed in the previous chapter have been utilized to compute the results of porous bearing static and dynamic performance characteristics. The static performance characteristics include load carrying capacity, frictional torque on ball surface due to viscous shear, maximum fluid-film pressure, and axial flow while the dynamic performance characteristics include the fluid-film stiffness and damping coefficients and stability threshold speed.

The performance characteristics of porous ball bearing system are computed for the following operating and geometric parameters of the porous bearing.Aspect ratio, λ =1.0

Eccentricity, ε = 0.2, 0.4, 0.6, 0.8, Speed parameter, Ω = 2.9, 5.8, 8.7, 11.6, 14.5, 17.4.

Permeability parameters is 0.005 to 0.05 Power law index for Newtonian lubricant n = 1.0.

A FORTRAN computer program which uses the relaxation iterative method is developed to solve the above governing equations. The Performances of finite porous ball bearing taking into account the eccentricity ratio and speed parameter effects are then determined.

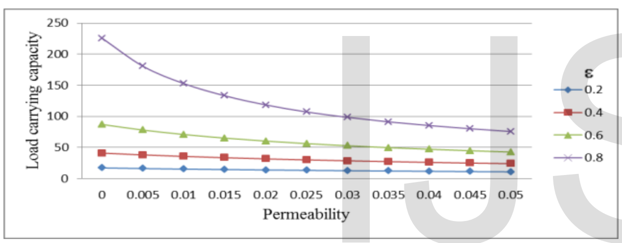


Fig 6.1: Variation of load carrying capacity v/s permeability. Graph shows the influence of permeability of b matrix on load carrying capacity of b ball bearing at different eccentricity ratios. The load carrying capacity of bearing is observed to be decreasing as permeability increases. This is because, the resistance to the axial flow of flow in region increases as the permeability increases and this result in reduction of load carrying capacity. For the range of eccentricity ratio considered, the load carrying capacity of bearing almost decreases exponentially with permeability. The reduction in load carrying capacity with permeability is more pronounced at higher eccentricities..

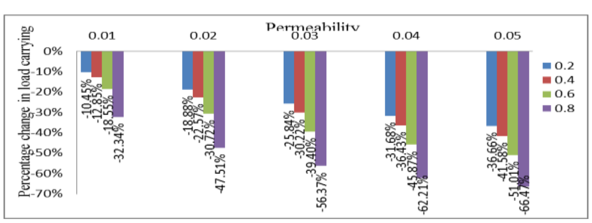


Fig 6.2: Percentage change in load carrying capacity v/s permeability. The percentage reduction in the value of load carrying capacity of bearing is more for the fixed value of permeability. A very significant reduction in the load carrying capacity at eccentricity ratio = 0.8 and it is in the range of 32.32% to 66.47% for this eccentricity ratio as permeability increases from 0.01 to 0.05.

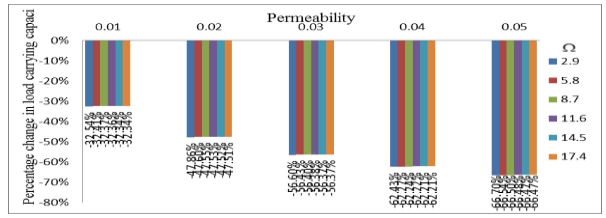


Fig 6.3: Percentage change in load carrying capacity v/s permeability. . Graph shows the variation of load carrying capacity of a b bearing with permeability at different speed parameter. For the range of speed parameter considered, the load carrying capacity of bearing almost decreases exponentially with permeability. The decrease in load carrying capacity is same for all ranges of speeparameter as shown in Fig.4.4

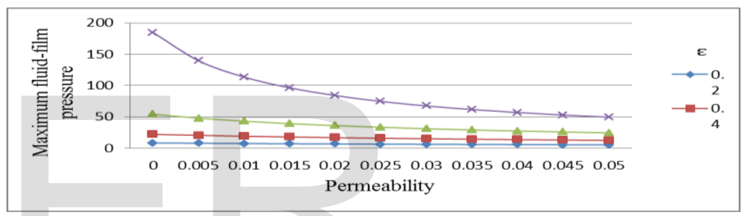


Fig 6.4: Variation of fluid-film pressure v/s permeability. Graph shows the variation of the maximum pressure with permeability at different eccentricity ratios. The increase in permeability reduces the fluid-film pressure due to oil leakage into the b matrix. The observed variation of maximum fluid-film pressure with the permeability and the eccentricity will modify the system load carrying capacity. For the range of eccentricity ratio considered, the fluid-film pressure of bearing almost decreases exponentially with permeability. There is a very significant difference noted in the variation of maximum fluid-film pressure with permeability at higher eccentricity ratios, especially at eccentricity ratio = 0.8

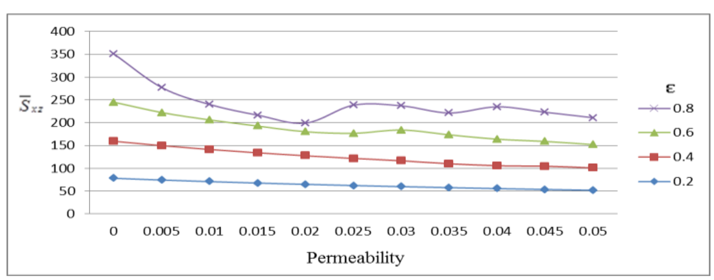


Fig 6.5: Variation of cross coupled stiffness coefficient $\bar{S}_{xz}$ v/s permeability. ε

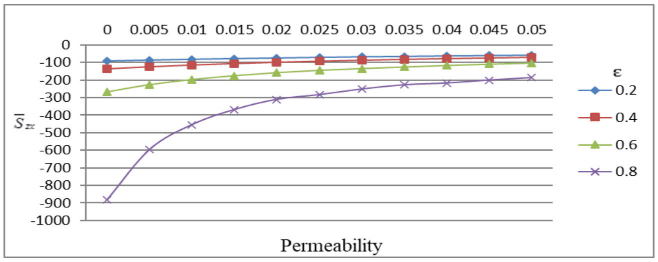


Fig 6.6: Variation of cross coupled stiffness coefficient $\overline{S}_{xz}$ v/s permeabiity.

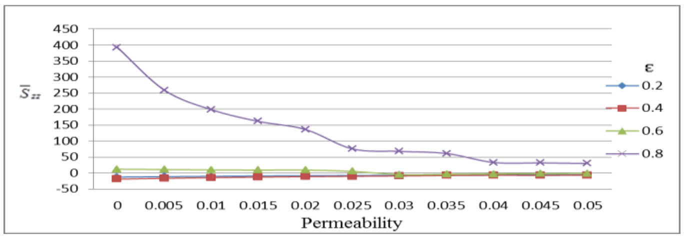


Fig 6.7: Variation of direct stiffness coefficient $\overline{S}_{zz}$ v/s permeability

Fig. 6.5, 6.6 and 6.7, the permeability of b matrix reduce the cross coupled stiffness coefficients (and) and reduction in these coefficients is more significant at higher eccentricity ratios. At lower eccentricity ratios less than 0.4, the variation of these cross stiffness coefficients with permeability of b matrix is almost linear

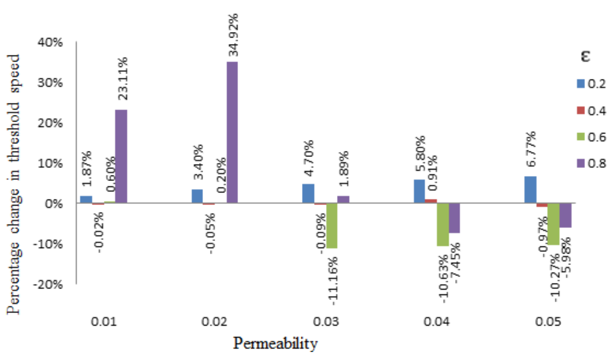
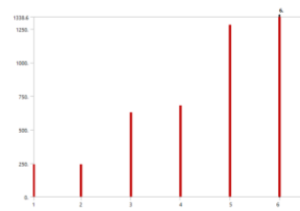


Fig 6.8: Percentage change in threshold speed v/s permeability

As shown in the Graph for the eccentricity ratio 0.8, the maximum enhancement in the threshold speed is 34.92 % for the the permeability value of 0.02.

7. MODAL ANALYSIS

Modal analysis is performed to determine the vibration characteristics of the structure i.e. natural frequency and modal shapes. It is one of the important aspects of the design. Determining the natural frequency and can predict the frequency at which the structure will fail. It evaluates free vibration modes shape to characterize displacement patterns. Mode shapes gives information about which structure naturally displaces. Lateral displacement is given more importance. Lower order mode shape are greatest contributors of the structural response as the order increases, contribution will be less. Mode shape depends on degrees of freedom. N degrees of freedom will have N modes shape. A mode is a state of a deformed shape in which the structure will exchange kinetic-energy and stain-energy continuously, and the natural frequency at which the mode shape occurs. In un-deformed state - kinetic energy is at its peak and strain energy is equal to zero. In deformed shape strain energy is maximum and kinetic energy is zero.



Tabular Data

	Mode	Frequency [Hz]
1	1.	240.56
2	2.	241.28
3	3.	628.2
4	4.	680.78
5	5.	1278.2
6	6.	1338.6

Figure 7.1, Natural frequency and corresponding modes in ball bearing

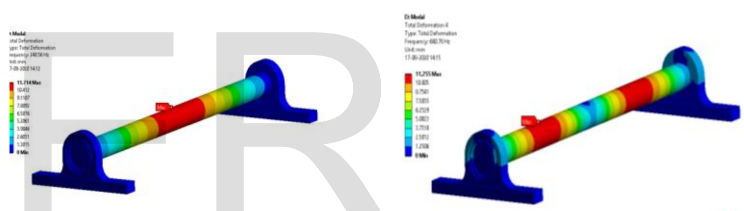


Figure 7.2, 1and 2 mode with corresponding natural frequency

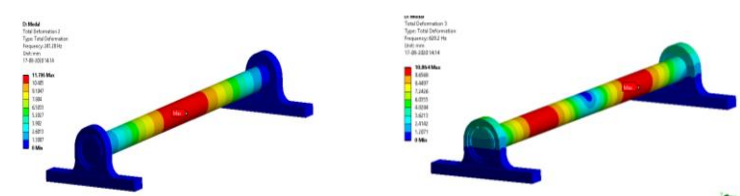


Figure 7.3, 3 and 4 mode with corresponding natural frequency

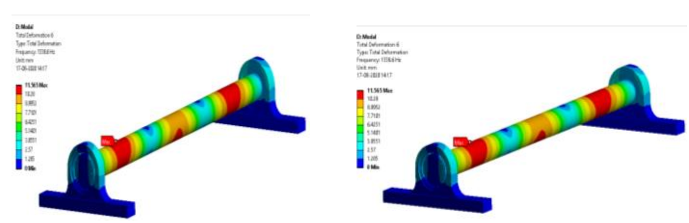


Figure 7.3, 5 and 6 mode with corresponding natural frequency

8. Conclusions

Mathematical models and solution algorithms for the analysis of porous ball bearing has been developed. The numerical simulation for analysis of porous ball bearing has been done by using finite element technique. The model analysis has been done to find natural frequency and corresponding modes. From the results present in the previous section, the following conclusions are drawn:

- The performance parameters of finite porous ball bearings are significantly influenced by the permeability of porous matrix, especially at higher eccentricity ratio.
- The permeability of porous matrix reduces the load carrying capacity, maximum fluid-film pressure, and axial flow of lubricant and frictional torque of the ball bearing. The reduction in the values of above parameters due to permeability effect of porous matrix is more pronounced in a bearing operating at higher eccentricity ratio.
- The fluid-film stiffness and damping coefficients are also reduces due to permeability of porous matrix and this consequence is more pronounced at higher operating eccentricity of bearing.
- Six initial modes and corresponding natural frequency.

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