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Finite Element Analysis of Cartridge Tapered Roller Bearing of Freight Wagon

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ABSTRACT:

Freight trains run under high service loads during consignment loading and operation so tapered roller bearings are ideally suited to wheel bearing applications. The tapered roller bearings used in the railway industry are of a standard design fixed by the American Association of Railroads regulations. Nowadays rail industry improves the train operating speeds, which means that failure of a bearing will result into a derailment, affecting human lives, network disruption, and damage to the railroad, unplanned maintenance costs, and generating fear in general public about rail transport. So the rail industry has focused on the improvement in maintenance work and improvement in component design. This paper discusses the results of finite element analysis and model analysis of Cartridge Tapered Roller bearing (CTRB). Solid modelling of CTRB has been done using solid works. The CTRB is then discretized using ANSYS software and 3D hexahedral solid elements are used to mesh the components. The effect of vibration modes on the dynamic behaviour and stability of wagon is described. Frequencies up to a range of 100 Hz are considered for mode shapes.

KEYWORDS:

Railways vehicle; Cartridge tapered roller bearing; Natural frequency; Mode shapes; Finite element method

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1. Introduction

Freight trains run under high service loads during consignment loading and operation, and that requires a bearing design that can tolerate these conditions. Tapered roller bearings are ideally suited to wheel bearing applications because of their ability to sustain both radial and thrust loading conditions. In addition, high-stress levels are imposed on the bearing because of the weight of the freight, impact loads during car loading, car humping, cornering, or poor track conditions. The tapered roller bearings used in the rail industry are of a standard design fixed by the American Association of Railroads (AM) regulations. These bearings are continually being improved within the constraints set by the AAR in order to maximize the service life of the bearing. These controls by the AAR are necessary to maintain compatibility of the bearings within the existing railroad industry. Continuous improvements to the bearings within these constraints are needed as service loads are constantly increasing to meet the increased demand.

Due to day by day increasing demand for logistic support provided by the railroad industry, attention has been given to improve speed so that the operation time can be reduced. It is evident that along with the incorporation of higher speed in railways to meet increasing demand for logistics support, the comfort of passengers as well as the safety of goods has to be taken care as well. In the train wagon structure, there exist a large number of excitation sources, resulting in oscillations, vibrations and noise. Every rail component has its significance in these vibrations. Long wavelength geometric irregularities in the track alignment will result in lateral displacement, making discomfort for the passengers while short wavelength irregularities can generate vibrations and noise. The same phenomenon is valid for long as well as short wavelength irregularities of the track level or vertical profile [1]. Irregularities, design defect, faulty installation or damage of the CTRB can also result in vibration and noise. Harak et al [2] determined the structural dynamic response for the virtual freight wagon using mode shapes. Gerdun et al [3] provided case studies of rail bearing failure and explained the bearing damage mechanism. It is suggested by Shukla & Harsha [4] that mode shapes can be an appropriate tool for preliminary analysis body under dynamic loading subjected to external excitation. Harak et al [5] studied the effect of geometric non-linearity on rail draft pad with the help of modal analysis and Guo et al [6] investigated the modal characteristics of human spine model.

In Indian railways, at present, Cartridge Taper Roller Bearings (CTRB) class E (6"x11") is being used on all air brake freight railway vehicles. Cartridge tapered roller bearing is a self-contained, pre-adjusted, pre-assembled, pre-lubricated, and a completely sealed unit and is installed or un-installed to the axle without removing the bearing elements, seals or lubricants to contamination or damage. It is designed for 100-ton wagon capacity i.e. more than 11-ton design load for each bearing. The reason of the vibration generation in rolling element bearings can be wrong fitting or various defects. Abrupt changes occurred due to this faulty installation or defect will result into contact stresses. Structural excitation because of these changes results into excitation and the generation of vibrations [7]. As per the traditional dynamic analysis approach for railway component analysis, generally the components were considered and modelled as rigid bodies which result in over-stiffening of the bearing or only small section of bearing raceway was taken. While in this work, all bearing components are considered as flexible, hence able to present accurate depiction of the stiffness and vibration response. For this kind of analysis a computer model, whose output can be used to predict exact dynamic behaviour of a railway vehicle can be constructed. A generalized input output relationship of simulation process used in most computer tools is shown in Fig. 1. ANSYS 12.1 has been used to build the Indian railway ICF bogie and finite element dynamic analysis has been used to extract the Eigen mode natural frequencies. Solid45 elements were used to mesh the bogie frame, bolsters and wheels. Combin elements have been used to represent the suspensions [9].

Harmonic response peaks at various salient locations of the bogie frame are observed to be well matching with the natural frequencies of the Indian railway bogie model developed using ANSYS 12.1. PSD random response at the front and rear end of the bogie are observed to be within the ISO 2631 curve limits for the standard PSD inputs at the appropriate bogie wheel contact nodes [10]. Eigen mode frequencies were obtained for a 7 degree of freedom passenger car model developed in ANSYS 14 through finite element analysis and also by numerical computations. The predominant modes of the 7 DoF model are observed to be in good agreement [11]. In this work, a dynamic non-linear finite element model of a Cartridge Tapered Roller Bearing is developed. The model was made in SOLIDWORK2016 and ANSYS17.2 and has been used as analysis tool to determine its structural dynamic response. Material detail of components and formulation of meshing and contacts in various geometries are presented in Section 2. In Section 3 results of modal analysis and mode shapes have been discussed and finally concluded in Section 4.

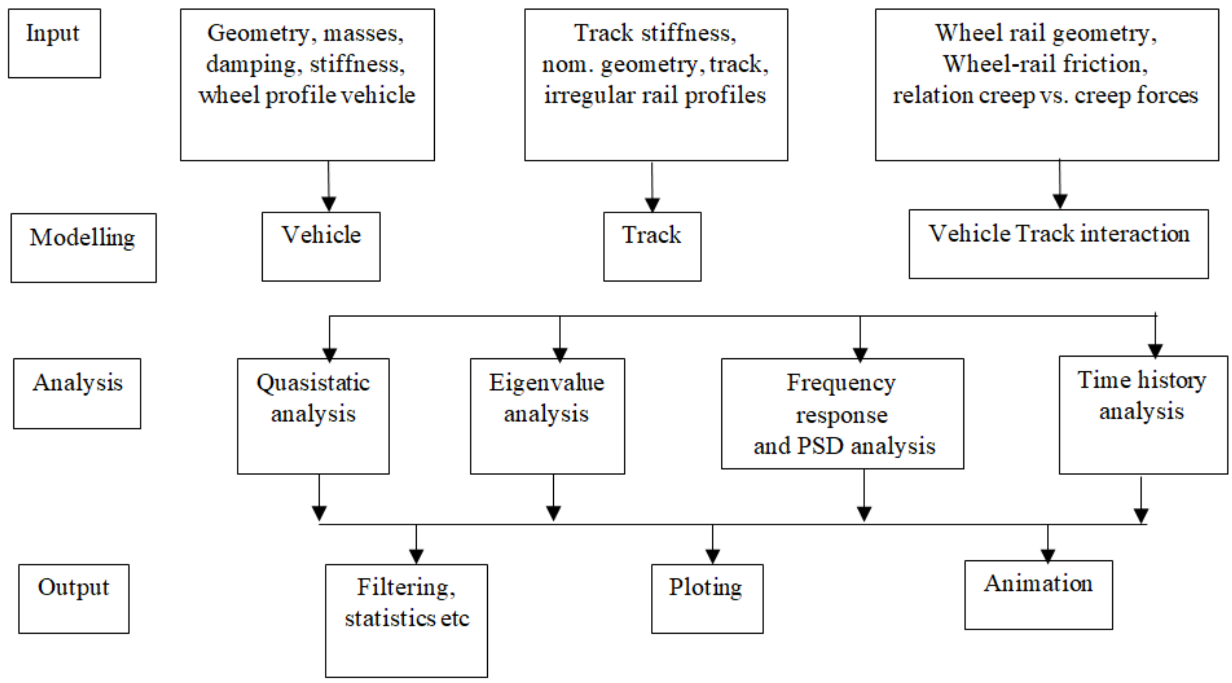

Fig. 1: Generalization of the simulation for the railway system [8]

2. Problem formulation

2.1. CTRB model description

CTRB considered here for this analysis is class E (6"×11"), which is being used on all air brake freight railway vehicles. Major parts in cartridge tapered roller bearings are cone i.e. inner race of bearing, the cup i.e. the outer race, tapered rollers and a cage as roller retainer. The cone is pressed onto the axle and transmits the load from rollers to the axle. An isometric view of the CRTB is shown in Fig. 2. The components of CRTB are shown in Fig. 3 and their description along with the material is listed in Table 1. Solid modelling of CTRB has been done using solidworks as per the dimensions specified by RDSO. This virtual model of CTRB consists of various components as substructures which are connected with each other to form a complete bearing. These all components have been modelled as elastic bodies to maintain the structural dynamics. Contacts as interfaces between these parts have been defined separately.

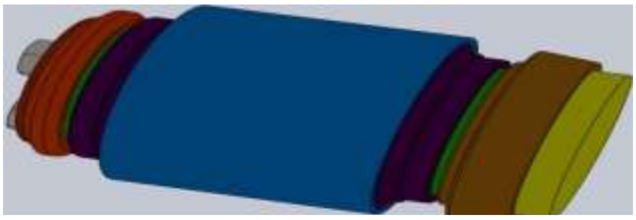

Fig. 2: CTRB

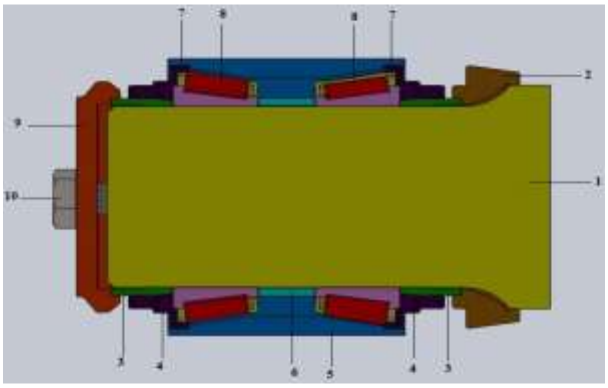

Fig. 3: Schematic cross section

Table 1: Various parts list of CTRB (Serial numbers as per Fig. 3)

S. No.	Part Name	Material
1	Axle	Steel
2	Backing ring	S.G. iron
3	Seal wear ring	Steel
4	Seal	Nitrile Rubber
5	Cup	Steel
6	Cone spacer ring	Steel
7	Cage	H.R. Steel
8	Rollers	Heat treated steel
9	End cap assembly	S.G. iron
10	UNC Cap screws	Steel

2.2. Need for modal analysis

Eigenvalues and Eigenvectors give the direct physical interpretation of mechanical problem system. In structural dynamics problems, system poles generally result in complex conjugate pairs and these pairs represent a structural mode. Resonance frequency of the pole has been described by the imaginary part while damping is shown by the real. As the value of structural damping is generally very low and resonance effects are dominating and directly detectable, resonance effect is directly linked to the structural dynamics problems. Physical coordinates on the structure generate the system's eigenvectors and these eigenvectors are related to the distinguishing structural vibration patterns of the system defined as the system's mode-shapes. Hence model analysis can be described as the derivation and use of this system model, on the basis of resonance frequencies, damping ratios and mode-shapes. These can be calculated analytically for the mechanical system if a lumped mass-spring system is concerned, while for most of the practical cases, the system is of a continuous structure nature, numerical approximation by a Finite Element Model (FEM) is made [12].

2.3. Solution algorithms for a modal analysis

Syntax extract of the system solution information is given as hereunder:

- ANSYS solution routine
- Perform a modal analysis
- This will be a new analysis
- Use QR damp mode extraction method
- Extract 6 modes
- Compute complex mode shapes
- Normalize the mode shapes to the mass matrix

It is evident that QR-damped mode extraction method has been used by solver. This method gives the advantages of a symmetric eigen solution method and complex Hessenberg method.

2.4. Details of the finite element model

The model designed in SOLIDWORKS, has been imported to ANSYS. For defining and storing the material specifications, engineering data manager is used. The contact faces were identified and defined as per the suitable contact definition between the pair. A quality metrics mesh was generated. The CTRB is then discretized in the domain of ANSYS software and 3D hexahedral solid elements are used to mesh the axle, backing ring, cones, spacer, rollers, seal wear ring and cup (Figs. 4, 5). Orthogonal quality and skewness matrices were given special consideration. Most of the elements fall into the 'excellent' to 'good' zone. A few fraction of elements fall into the poor zone which is negligible for this structure. Total numbers of nodes are 476897 and elements are 207609.

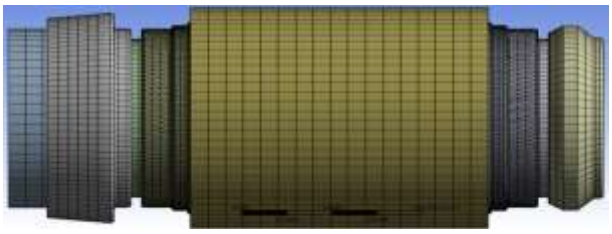

Fig. 4: CTRB mesh including all components

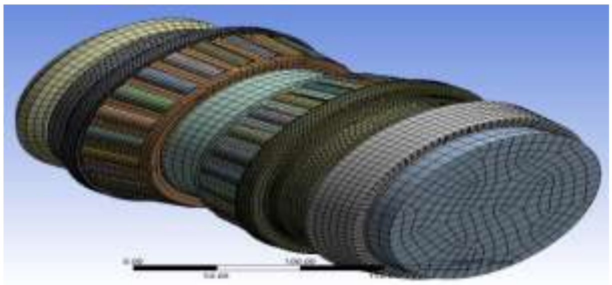

Fig. 5: Meshing of internal parts

3. Result and discussion

The FE model and the system Eigen-modes show the CTRB which is not attached to its actual working environment. The deformation of bearing structure is affected by its cages and rollers. Those modes, that depict local deflections and hence significantly influence the noise and stability, have been considered. The range for mode shapes was taken as frequency of 100 Hz. So by taking frequency limit of 100 Hz, 6 mode shapes of the bearing system are recognized and shown in Table 2.

Table 2: Modal analysis results for CTRB

Mode No	Frequency (Hz)	Explanation of mode shapes
1	1.08×10^{-2}	Rigid motion
2	1.14×10^{-2}	Rigid motion
3	1.234×10^{-2}	Rigid motion
4	52.385	End cap in Longitudinal deflection
5	53.733	End cap in Longitudinal deflection
6	99.885	Cages and Backing under max deformation

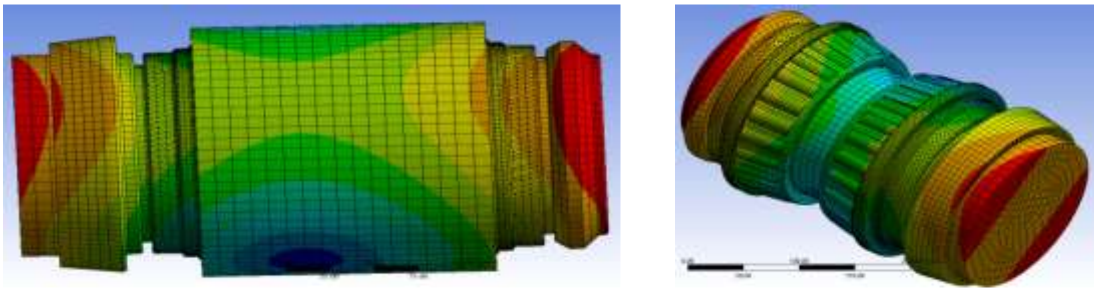

Fig. 6: Mode shapes-4th modes, 52.385 Hz

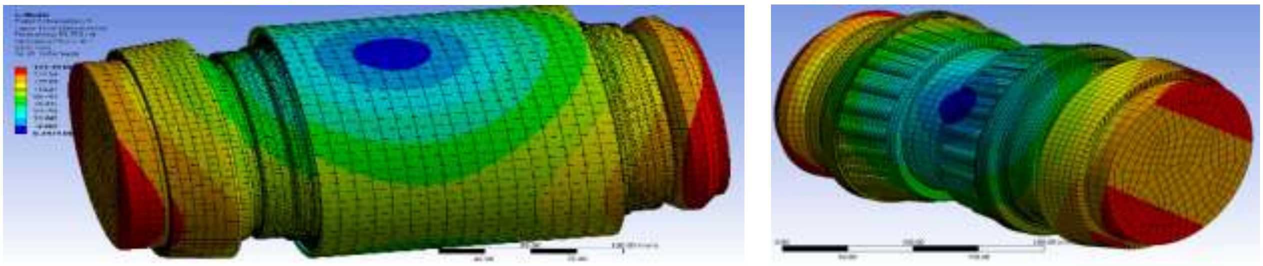

Fig. 7: Mode shapes-5th mode, 53.733 Hz

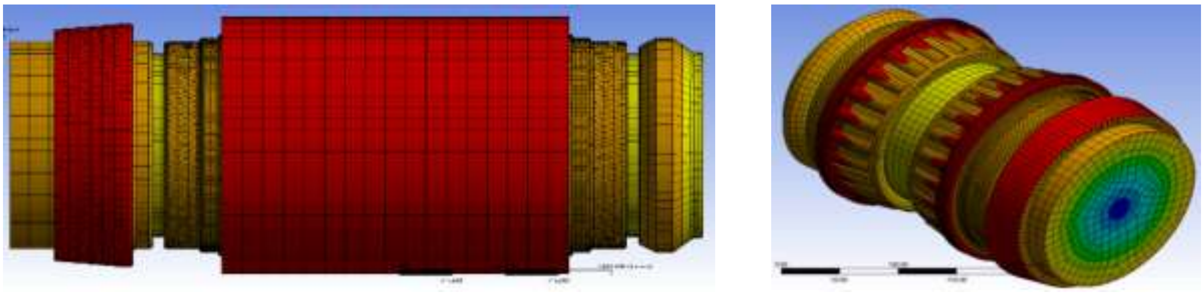

Fig. 8: Mode shapes-6th mode 99.885 Hz

Up to first three mode shapes, structure exhibit almost rigid motion. Frequency obtained is almost zero and no elastic deformation of the structure is visible. Each component of the structure displays the same deformation. The 4th mode shape displays a drastic change in behaviour. Modal frequency at this mode shape is jumped to 52.385 Hz from almost zero of previous mode shape. Fig. 6 depicts the directional deformation. End cap assembly of CTRB is deformed maximum among component. Fifth modal frequency is 53.733 Hz and this quantitative equivalence to previous mode shape is also visible in the Fig. 7 where the same component is shown under longitudinal deflection along with some portion of backing ring. The sixth modal frequency occurs at 99.885 Hz and backing ring, both cages and upper portion of all rollers (in direct contact with cup) are under major deformation. Fig. 8 depicts that the each component of the CTRB at all locations is sensitive to lateral deflection at this mode shape. So excitation frequency near this value has to be taken care.

4. Conclusions

The dynamic behaviour analysis of the CTRB has been determined by finite element tool. First three mode shapes show the rigid behaviour of CTRB. Further mode shapes shows sudden change in deformation, because of geometric non-linearity in the bearing structure. Mode shape having frequency range near 100 Hz is of great interest for this dynamic behaviour. The dynamic behaviour of the CTRB shows if external excitation frequencies due to track profile or carbody excitation match with model frequencies of the CTRB, instability of the structure will drastically increase. This dynamic analysis of rail bearing structure is carried out at no preloading of wagon. Studies show that the bearing preloads has a significant effect on the natural frequencies and hence influences the vibration behaviour of the bearing assembly. Thus the future work should be concentrated around preloading of bearing in empty and loaded wagon case.

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