Research paper

Measurement of FRFs of coupled geared rotor system and the development of an accurate finite element model

Ankur Saxena\textsuperscript{a}, Manoj Chouksey\textsuperscript{b}, Anand Parey\textsuperscript{a,*}

\textsuperscript{a} Mechanical Engineering Discipline, School of Engineering, Indian Institute of Technology, Indore, India

\textsuperscript{b} Mechanical Engineering Department, SGSITS, Indore, India

**A R T I C L E   I N F O**

Article history:
Received 13 June 2017
Revised 24 October 2017
Accepted 9 January 2018

Keywords:
Geared rotor system
Mesh stiffness
Mesh damping
Frequency response functions

**A B S T R A C T**

An experimental study has been attempted to find out the effect of gear pair contact on the modal behaviour of an actual geared rotor system mounted on rolling element bearings. This is important as gear pair contact considerably affects the dynamic characteristics of the system. Frequency response functions of the uncoupled and coupled geared rotor systems have been measured to find out the effect of gear pair contact on the natural frequencies of the system. Moreover, a finite element model of the geared rotor system has been developed by modelling the shaft continuum, bearing coefficients and gear pair contact (using mesh stiffness and damping). Non-proportional viscous damping has been considered in modelling bearing and mesh damping coefficients. The results obtained after numerical simulation of finite element model are validated with the experimental results to check the accuracy of the developed finite element model.

© 2018 Elsevier Ltd. All rights reserved.

1. Introduction

Geared devices are indispensable components of many of the machines e.g. of wind turbine power plants, automobiles, helicopters etc. Unlike other rotating machines, the dynamics of geared rotor systems is significantly influenced by the mesh stiffness due to the gear pair contact [1,2]. Modal analysis is generally used to find out the dynamic characteristics of structures [3,4]. Measured frequency response functions (FRFs) are generally used to find out the modal characteristics of any system [5]. This work primarily uses measured FRFs of the uncoupled and coupled geared rotor systems to find out the effect of gear pair contact on the natural frequencies of the coupled geared rotor system. The term “coupled” implies that the two gears are meshing with each other under rotation, whereas the term “uncoupled” implies that the gears are not in direct meshing.

In the case of geared rotor systems, it is important to properly calculate the gear mesh stiffness and damping among other parameters. In literature, many studies have been reported for the calculation of the mesh stiffness. Yang and Sun [6] proposed a value of Hertzian energy which is further extended by Yang and Lin [7] to calculate time-varying mesh stiffness (TVMS) of a gear pair using potential energy method by including bending energy and axial compressive energy with Hertzian energy. Effect of crack initiation and propagation and various other faults on TVMS is extensively studied in literature. Wu et al. [8] studied the effect of tooth crack on the vibration responses of single stage gearbox and shown the effect of growth in tooth crack length on total mesh stiffness of spur gear pair. Chen and Shao [9] proposed an analytical model to

\* Corresponding author.
E-mail address: anandp@iiti.ac.in (A. Parey).

https://doi.org/10.1016/j.mechmachtheory.2018.01.010
0094-114X/© 2018 Elsevier Ltd. All rights reserved.
simulate tooth root crack. The model included the effect of crack propagation along tooth width and crack depth and with their analytical formulation mesh stiffness of tooth root crack is compared. Saxena et al. [10] studied the effect of shaft misalignment and friction on total effective mesh stiffness for cracked spur gear pair tooth using potential energy method and showed that the misalignment and friction affect TVMS of the gear pair considerably. Liang et al. [11] extensively reviewed the literature discussing about the use of potential energy method in gearbox dynamic models.

A few studies have been reported in the literature on coupled lateral-torsional vibrations of the geared rotor systems. Iwatsubo [12] studied coupled lateral-torsional vibration of geared rotor using transfer matrix method by assuming tooth rigidity as a linear spring. Li et al. [13] studied coupled lateral – torsional dynamics of the rotor bearing systems of spur bevel gear and concluded that critical speeds of coupled modes are different from those of uncoupled modes. Choy et al. [14] analyzed multi-stage gear transmission system by considering it as an enclosed structure and simulated transient and steady state vibrations due to torque variation, speed change, rotor imbalances and gear box support motion excitations. Pederson et al. [15] reported the mathematical foundations of time-variant modal analysis including the effect of TVMS on spur gear dynamics. Both numerical methods (e.g. finite element method) and analytical methods (giving close form solutions) have been employed to study the dynamics of geared rotor systems.

Finite element method is a widely used technique for numerical simulation of statics and dynamics of structures [16]. Modal analysis using finite element method has been employed by many researchers for estimation of natural characteristics of dynamic systems [17,18]. However, such studies are limited to the geared rotor systems. Wilcox and Coleman [19] reported a technique using finite element method for determining stress values at the tooth surface in the root fillet and analyzed tensile fillet stress in the generated tooth shapes incorporating either symmetric/asymmetric profiles. Kahraman et al. [20] developed a finite element (FE) model of a geared rotor system supported on the flexible bearing and showed that some of the natural frequencies reduce due to the gear pair contact in the coupled geared rotor system. Saxena et al. [21] used FE method to study the effect of gear tooth faults on the FRFs, natural frequencies and mode shapes of a geared rotor system. This paper reports an experimental investigation of the effect of gear pair contact on the natural frequencies of the geared rotor systems and to validate the FE model data with the measured vibration data of the system.

2. Details of experimental set-up for modal testing

The experimental setup of the geared rotor system is shown in Fig. 1. The experimental setup consists of two spur gears mounted on the shafts using keyways. The shafts are supported by roller bearings and bearings are mounted in the rigid frame. Uniaxial accelerometer (Model no. – PCB 333B32) with a sensitivity of 10.57 mV/ (m/s²) has been used to measure response during modal testing of the system. An impact hammer (Make – Dytran Dynapulse) with a sensitivity of 2.25 mV/N is employed to excite the geared rotor system. The frequency resolution of 0.625 Hz has been set to measure the FRFs. The measured excitation and response are fed into the data acquisition system (Make - OROS), which computes the FRFs of the structure. Modal parameters can be extracted from the measured FRFs. A review of various methods to extract modal data from the FRFs is given by Avitabile [22]. In this work, NVGate software’s FFT plug-in is used to extract the natural frequencies from the measured FRFs. Fig. 2 shows the schematic diagram of gear and bearing arrangement. Table 1 shows the parameters of the geared rotor system.
This work initially reports the experimental measured FRFs to analyse the effect of coupling. This is followed by the development of an accurate FE model for the geared rotor system. The measurement points considered for the model of the geared rotor system are shown in Fig. 3. It is assumed that each shaft consists of six elements of the same length. The measurement points for the experimental FRFs are also considered as per the node points as shown in Fig. 3. Each rotor-shaft is supported by a pair of roller bearings. Both the gears are mounted at the centre of the shafts. The spur gears are considered to be rigid and are represented by their inertia properties, which are found by generating a solid model of the gear using Solid Works software. The bearing supports and gear pair contact are modelled using springs and dampers.

### 3. Measurement of the FRFs for uncoupled and coupled gear pairs

This section includes the experimental study to find out the effect of gear pair contact on the measured FRFs and the natural frequencies. A loading arrangement has been provided as shown in Fig. 1 to ensure gear pair contacts. Geared rotor systems remain uncoupled in the horizontal and vertical planes of motion for the non-spinning conditions. Therefore
measured FRFs are recorded in both the planes separately. This is followed by measurement of FRFs for the coupled geared rotor system by establishing the gear pair contacts using loading arrangement. Fig. 4(a) shows the measured FRFs ‘H_{y2y3}’ as measured on uncoupled 32 teeth gear shaft, which implies the response at point 2 in vertical direction due to unit harmonic force at point 3 in the vertical direction. Similarly, Fig. 4(b) shows the FRF plot (H_{y9y10}) for the uncoupled 48 teeth gear shaft in the vertical plane. These FRFs have been used to record the natural frequencies of the system in vertical modes of vibrations. Similarly FRFs have been recorded in the horizontal plane to record the horizontal modes of vibration. Fig. 5 shows the measured FRF (H_{y2y3}) for the coupled geared rotor system in the vertical plane. The measured FRFs have been used to extract the natural frequencies of the coupled and uncoupled geared rotor system. These natural frequencies have been listed in the Table 2. It can be noted that the coupling has the effect of reducing the natural frequencies of the geared rotor system. This effect of reduction of natural frequencies is found to be more prominent in the first mode of vibration. The effect of coupling on reduction of natural frequency in first mode of vibration is clearly seen in the coupled system FRF (Fig. 5), as the reduced natural frequency peak comes at 121.9 Hz, which had a previous value of 134.4 Hz in the FRF of uncoupled system (refer to Fig. 4(b)).

![Fig. 4](image1)

**Fig. 4.** Frequency response function of uncoupled geared rotor shaft in vertical plane for (a) 32 teeth gear shaft (H_{y2y3}), (b) 48 teeth gear shaft (H_{y9y10}).

![Fig. 5](image2)

**Fig. 5.** Frequency response function (H_{y2y3}) of coupled geared rotor system.

<table>
<thead>
<tr>
<th>Mode No.</th>
<th>Natural frequency (Hz) (Uncoupled rotor)</th>
<th>Natural frequency (Hz) (Coupled rotor)</th>
<th>Details of the corresponding mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>32 teeth rotor</td>
<td>48 teeth rotor</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>–</td>
<td>134.4</td>
<td>121.9</td>
</tr>
<tr>
<td>2</td>
<td>–</td>
<td>134.8</td>
<td>125.7</td>
</tr>
<tr>
<td>3</td>
<td>156.9</td>
<td>–</td>
<td>156.9</td>
</tr>
<tr>
<td>4</td>
<td>157.3</td>
<td>–</td>
<td>158.6</td>
</tr>
<tr>
<td>5</td>
<td>–</td>
<td>722.5</td>
<td>719.3</td>
</tr>
<tr>
<td>6</td>
<td>–</td>
<td>723.3</td>
<td>719.7</td>
</tr>
<tr>
<td>7</td>
<td>772.5</td>
<td>–</td>
<td>765.6</td>
</tr>
<tr>
<td>8</td>
<td>773.2</td>
<td>–</td>
<td>766.1</td>
</tr>
</tbody>
</table>
4. Finite element modal analysis of geared rotor system

Finite element modal analysis of a geared rotor system requires the modelling of two shafts i.e. driving shaft, driven shaft apart from other elements like bearings supporting the shafts, gear and pinion etc. The rotor-shafts of the geared rotor system are discretized using two noded Timoshenko beam element with six degrees of freedom (three translations along X, Y and Z axis and three rotations along X, Y and Z axis) per node. The equations of motion have been formulated by taking into account the effects of rotary inertia, translatory inertia; shear deformation, shaft flexibility, support flexibility and gear pair parameters. Fig. 6 shows the ith finite element, connecting ith and (i + 1)th nodes, along with the degrees of freedom at the nodes.

Nodal displacement vector of ith element is given in Eq. (1)

$$ q_i = \begin{bmatrix} x_i, y_i, z_i, \theta_{x1}, \theta_{y1}, \theta_{z1}, x_{i+1}, y_{i+1}, z_{i+1}, \theta_{x2+1}, \theta_{y2+1}, \theta_{z2+1} \end{bmatrix}^T $$

where, the superscript $^T$ denotes transpose of a vector.

4.1. Equations of motion

After assembling the governing equations for all the elements and incorporating the boundary conditions, the equations of motion of the geared rotor bearing system with N degrees of freedom are written as:

$$ \mathbf{Mq}\ddot{q}(t) + \mathbf{Cq}(t) + \mathbf{Kq}(t) = \mathbf{f}(t) $$

where, $\mathbf{M}$, $\mathbf{C}$, $\mathbf{K}$ are square $N \times N$ matrices and are usually referred to as the mass or inertia matrix, the damping matrix and the stiffness matrix respectively. The vectors $\mathbf{q}(t)$ and $\mathbf{f}(t)$, each of size ‘$N$’, stand for generalized displacement and generalized force vectors respectively. Time and time derivative are represented by ’$t$’ and ‘. ’ respectively.

For the geared rotor-bearing system, the matrices $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ of the Eq. (2) are defined as follows:

$$ \mathbf{M} = \mathbf{M}_{trs} + \mathbf{M}_{rot}; \quad \mathbf{C} = \begin{bmatrix} \mathbf{C}_{brg} + \mathbf{C}_m \end{bmatrix} \quad \text{and} \quad \mathbf{K} = \begin{bmatrix} \mathbf{K}_{brg} + \mathbf{K}_m \end{bmatrix} $$

In the above, $\mathbf{M}_{trs}$ and $\mathbf{M}_{rot}$ denote the mass matrices due to translation and rotation inertia respectively. $\mathbf{C}_{brg}$ and $\mathbf{C}_m$ stands for the bearing damping matrix and the mesh damping matrix respectively. The symbol $\mathbf{K}$ is the matrix coefficient to the displacement vector and consists of global bearing stiffness matrix ($\mathbf{K}_{brg}$) and gear mesh stiffness matrix ($\mathbf{K}_m$). All the matrices are of size $N \times N$. It may be noted that $N$ represents the total degrees of freedom of the geared rotor system, i.e. including it for both driving and driven rotor shaft.

As the stiffness matrix and the damping matrix also include the components due to the mesh stiffness and mesh damping respectively due to engagement of the gear and pinion, their computation is reported in the next section.

4.2. Calculation of gear mesh stiffness

This work uses potential energy method proposed by Chen et al. [23] for calculation of mesh stiffness to study the dynamic mesh performance of spur gear pair. This method has been used by many researchers for the calculation of gear mesh stiffness [10,24,25]. The method includes four components of energy, namely, Hertzian energy, bending energy, axial compressive energy and shear energy and fillet foundation deflection in the formulation. The mathematical basis for inclusion of Hertzian energy, bending energy and axial compressive energy is given in the work by Yang and Lin [7], whereas the inclusion of shear energy is reported by Tian [26] and fillet-foundation deflection reported by Sainsot et al. [27]. The complete method to calculate TVMS of spur gear pair can be found in Ref. [23].
The total effective TVMS represents the variation of mesh stiffness from engagement to disengagement of the spur gear pair for healthy gear tooth. The gear pair couples the bending and torsional degrees freedom at the gear and pinion. The mathematical basis of the additional coupling matrix is reported in Ref. [20] by defining one of the bending degrees of freedom along the pressure line of the gear pair. This results in a $4 \times 4$ coupling matrix for stiffness and damping due to the gear pair contact, which is given as per Eq. (4).

$$\begin{bmatrix} k_m & k_m r_p & -k_m & -k_m r_g \\ k_m r_p & k_m r_p^2 & -k_m r_p & -k_m r_p r_g \\ -k_m & -k_m r_p & k_m & k_m r_g \\ -k_m r_g & -k_m r_p r_g & k_m r_g & k_m r_g \end{bmatrix} = \begin{bmatrix} c_m & c_m r_p & -c_m & -c_m r_g \\ c_m r_p & c_m r_p^2 & -c_m r_p & -c_m r_p r_g \\ -c_m & -c_m r_p & c_m & c_m r_g \\ -c_m r_g & -c_m r_p r_g & c_m r_g & c_m r_g \end{bmatrix}$$ \(\text{(4)}\)

where, $[K_m]$ = mesh stiffness matrix, $[C_m]$ = mesh damping matrix, $k_m$ = mean mesh stiffness value, $c_m$ = mesh damping, $r_p$, $r_g$ = base circle radii of pinion and gear, respectively. Adding the mesh stiffness matrix and mesh damping matrix to the overall stiffness and damping matrix of the uncoupled rotor system gives the total overall stiffness and total overall damping matrix of the system.

4.3. Free vibration analysis of gear rotor system

Free vibration analysis of the system of equations (1) gives eigenvalues $\lambda_r$, right eigenvector $u_r$ and left eigenvector $v_r$ which are connected by equations given in (5) for the $r$th mode as

$$(\lambda_r^2 M + \lambda_r C + K) u_r = 0 \quad \text{and} \quad (\lambda_r^2 M^T + \lambda_r C^T + K^T) v_r = 0, \quad \text{where} \quad r = 1 \text{ to } N$$ \(\text{(5)}\)

The equations of motion for proportionally damped systems are generally represented in state space (first order representation) for mathematical convenience [28]. The second order equations of motion as given by Eq. (2) may be written in state space as follows:

$$A w(t) = B w(t) + f(t)$$ \(\text{(6)}\)

The matrices $A$, $B$ the state vector $w(t)$ and the force vector $f(t)$ are given as:

$$A = \begin{bmatrix} M & 0 \\ C & M \end{bmatrix}_{2N \times 2N}, \quad B = \begin{bmatrix} M & 0 \\ 0 & -K \end{bmatrix}_{2N \times 2N}$$ \(\text{(7)}\)

$$w(t) = \begin{bmatrix} q(t) \\ \dot{q}(t) \end{bmatrix}_{2N \times 1}, \quad \dot{f}(t) = \begin{bmatrix} 0 \\ f(t) \end{bmatrix}_{2N \times 1}$$ \(\text{(8)}\)

In the above, the phase variables have been chosen as the state variables.

The symbol ‘0’ represents a null matrix size $N \times N$ or a null vector of size $N$ abiding by dimensional requirement, e.g. ‘0’ represents a null matrix in Eq. (7) and a null vector in Eq. (8).

The $r$th eigenvalue problem associated with Eq. (7) may be written in state space after following Lee [29] as:

$$\lambda_r A \psi_r = B \psi_r \quad \text{and} \quad \lambda_r A^T \varphi_r = B^T \varphi_r, \quad \text{where} \quad r = 1 \text{ to } 2N$$ \(\text{(9)}\)

where, $\lambda_r$ is the $r$th eigenvalue, and $\psi_r$ and $\varphi_r$ stand for the corresponding $r$th right and $r$th left eigenvectors, respectively, in state space and are given by

$$\psi_r = \begin{bmatrix} \lambda_r u_r \\ u_r \end{bmatrix}, \quad \varphi_r = \begin{bmatrix} \lambda_r v_r \\ v_r \end{bmatrix}$$ \(\text{(10)}\)

$\psi_r$ and $\varphi_r$ may be biorthonormalized so as to satisfy

$$\varphi_i^T A \psi_r = \delta_{ir}, \quad \varphi_i^T B \psi_r = \lambda_r \delta_{ir};$$ \(\text{(11)}\)

The indices ‘$i$’, ‘$r$’ vary from 1 to 2$N$, and $\delta_{ir}$ is the Kronecker delta (i.e. = 1 when $i = r$ and = 0 when $i \neq r$) For $r = 1$ to 2$N$. All the left eigenvectors, right eigenvectors may be written in matrix form as $\Psi, \Phi$ and $\Lambda$ respectively and are given by:

$$\Psi = \begin{bmatrix} \psi_1 & \psi_2 & \ldots & \psi_{2N} \end{bmatrix}, \quad \Phi = \begin{bmatrix} \varphi_1 & \varphi_2 & \ldots & \varphi_{2N} \end{bmatrix}, \quad \Lambda = \text{diag}[\lambda_1, \lambda_2, \ldots, \lambda_{2N}]$$ \(\text{(12)}\)

Therefore bi-orthogonality condition expressed by Eq. (11) may be written in compact form as

$$\Phi^T A \Psi = \text{I}, \quad \Phi^T B \Psi = \Lambda$$ \(\text{(13)}\)

In Eq. (5), the symbol ‘$T$’ represents identity matrix of size $2N \times 2N$.

Frequency Response Function (FRF) matrix, $\mathbf{H}$, in state space may be written as:

$$\mathbf{H} = \sum_{i=1}^{2N} \frac{\psi_i \varphi_i^T}{i \omega - \lambda_r} = \sum_{i=1}^{2N} \frac{\psi_i \varphi_i^T}{i \omega - \lambda_r} + \mathbf{\tilde{H}} \varphi_i \varphi_i^T$$ \(\text{(14)}\)
where ‘\( i \)' represents complex conjugate of a vector/scalar.

Further, Frequency Response Function matrix \((H)\) relating generalised displacement and forces may be written as:

\[
H = \sum_{r=1}^{2N} \frac{u_r v_r^T}{i\omega - \lambda_r} = \sum_{r=1}^{2N} \frac{u_r v_r^T}{i\omega - \lambda_r} + \frac{\tilde{u}_r \tilde{v}_r^T}{i\omega - \lambda_r}
\]  \(\text{(15)}\)

The FRF matrix may be expanded as:

\[
H = \begin{bmatrix} H_{yy} & H_{yz} \\ H_{sy} & H_{zz} \end{bmatrix}
\]  \(\text{(16)}\)

4.4. Modelling of bearings

In the study, the experimental geared rotor system is supported on rolling element bearings. Proper modelling of bearing coefficients is very important in rotating machinery, which otherwise may add to inaccuracies in the model. The bearings supports are considered to be isotropic. The bearing and damping coefficients are considered as given below:

\[K_{yy} = K_{zz} = 1 \times 10^8 \text{ N/m};\ C_{yy} = C_{zz} = 200\text{N} \cdot \text{s/m};\]

These assumptions and the values of the bearings coefficients have considered after following the work by Stone et al. [30] and Chouksey et al. [18].

4.5. Convergence studies of the geared rotor system

A finite element model of the experimental geared rotor system has also been developed in this work. Convergence check, which ensures selection of proper number of finite elements, is very crucial in any study based on finite element method. The results of natural frequencies obtained by numerical simulation of finite element method have been checked for convergence. It is found that the results of natural frequencies, in all the eight modes, converge for 128 numbers of elements as shown in the Fig. 7.

5. Numerical simulations

This section comprises mainly the results of FRFs as found out using finite element method and their co-relation with the measured FRFs.

5.1. Simulation of TVMS of the gear pair

As has been discussed in the previous literature, the mesh stiffness of the gear pair considerably affects the dynamic characteristics of the geared rotor system. Mesh stiffness continuously changes from engagement to disengagement of gear pair and this variation is divided into three regions for low contact ratio gears (contact ratio < 2). These three regions correspond to, (i) a double pair contact, (ii) a single pair contact, and again (iii) a double pair contact.

The correct estimation of the mesh stiffness is very important in getting the overall model of the geared rotor systems. The variation of gear mesh stiffness for one complete gear tooth cycle for healthy spur gear pair can be calculated as mentioned in Section 4.2. It can be seen that in case of healthy gears, the variation in the mesh stiffness in the first and third
region remains same. Reduction of mesh stiffness is expected to influence the natural frequencies and FRFs. Three values of the mesh stiffness have been selected for theoretical modal analysis i.e. (i) the maximum value of the mesh stiffness for the double pair contact region, (ii) average value of the mesh stiffness for the complete gear cycle, and (iii) the minimum value for the single pair contact region. This covers maximum possible variation of the stiffness values for one complete tooth contact cycle. Fig. 8 gives the variation of the mesh stiffness within one shaft period. It is observed that the mesh stiffness varies periodically.

Table 3 shows the variation in modal parameters due to change in mesh stiffness value throughout one complete tooth contact cycle, which is not found to be prominent. However, maximum value of the mesh stiffness ($K_m = 3.90 \times 10^8$ N/m), depicting the double pair contact, has been considered in the numerical simulation for the further results. Therefore, the measured results are also found out for the double pair contact from the experimental set-up for the corresponding correlation with the numerical results. The value of mesh damping coefficient $C_m$ has been taken as $1.8 \times 10^3$ N-s/m after following the work by [31].

5.2. Comparison of measured vibration data and FE model results

Finite element (FE) model used for structural analysis should be precise and reliable. The accuracy of the FE model is often judged by the degree with which its results match with the experimental results. Any deviation in this is attributed to the modelling inaccuracies. The plots of measured FRFs and extracted natural frequencies have been correlated with the corresponding results as computed using FE model for case of uncoupled as well as coupled geared rotor system.

Fig. 9(a) and (b) show the overlay of the FE model FRF ($H_{y2y3}$) with the corresponding measured FRF for the uncoupled 32 teeth and 48 teeth gear rotor system respectively. Whereas the overlays of FE model FRF and the measured FRF for the
coupled geared rotor system is shown in Fig. 10. The plots clearly show that the natural frequencies (shown by location of peaks) predicted by the finite element model have a close match with the measured natural frequencies. These natural frequencies are further listed in Table 4 along with the percentage error in them with reference to the measured natural frequencies. Whereas Table 5 shows the comparison of the natural frequencies as found out from the FE model of the coupled geared rotor system and the measurement. These results indicate close agreement between the predicted natural frequencies using the FE model and the measured natural frequencies. The results also show reduction in the first mode natural frequency due to gear pair coupling.

Damping is a complex phenomenon and there may be different types of damping mechanisms in any actual system. This work, however, considers non-proportional viscous damping to model bearing and mesh damping of the system. It is seen from Figs. 9 and 10 that the peak amplitudes as predicted by FE model FRF have some deviation when compared with corresponding measured FRF peak. This may be attributed to the fact that damping is complex phenomenon and a single damping model may not represent the damping of the actual system.

6. Conclusions

An experimental study has been performed to find out the effect of gear pair contact on the natural frequencies and the frequency response functions of the geared rotor systems. The study reveals that the gear pair contact causes reduction in the natural frequencies of the geared rotor system and this effect is more prominent in the first bending mode. Finite element models of the uncoupled as well as coupled geared rotor systems have been developed. The gear pair contact is represented by mesh stiffness and mesh damping in the finite element model of the coupled geared rotor shaft system. The results of the finite element modal analysis also depict the reduction in the natural frequencies of the geared rotor system. However, the finite element model of any system may possess modelling inaccuracies. Proper modelling of bearings
coefficients along with mesh stiffness and damping coefficients is crucial in getting the accurate finite element model of the geared rotor system. The overlay of measured and finite element model Frequency Response Function show the accuracy of the model in the case of uncoupled as well as coupled geared rotor systems.

Supplementary materials

Supplementary material associated with this article can be found, in the online version, at doi:10.1016/j.mechmachtheory.2018.01.010.

References