Evaluation of frequency excitation of helical suspension spring using finite element analysis

Manoj A. Kumbhalkar*

Department of Mechanical Engineering,
KDK College of Engineering,
Nagpur, Maharashtra, India
Email: manoj.kumbhalkar@rediffmail.com
*Corresponding author

D.V. Bhope

Department of Mechanical Engineering,
Rajiv Gandhi College of Engineering, Research and Technology,
Chandrapur, Maharashtra, India
Email: dvbhope@rediffmail.com

A.V. Vanalkar

Department of Mechanical Engineering,
KDK College of Engineering,
Nagpur, Maharashtra, India
Email: avanalkar@yahoo.co.in

Abstract: Frequency response analysis is a technique used to determine the steady-state response of a linear structure to loads that vary harmonically with time. The aim is to calculate the structure's response at several frequencies and obtain results as response quantity versus frequency or time using finite element analysis. Peak responses are then identified on the graph and stresses reviewed at those peak frequencies. This paper discusses the modal and harmonic or frequency response of helical suspension spring of rail road vehicle using finite element tool ANSYS for different harmonic forces on inner and composite springs. The experimental result of accelerometer has been recorded for maximum speed of 80 km/hr determines excitations for amplitude of acceleration of suspension system versus time. Modal analysis finds the natural frequencies of spring for different modes of vibration for inner and composite assembly of spring and harmonic analysis of undamped suspension system reveals peak amplitude of stress and acceleration for the frequency range of 0 Hz to 50 Hz. The analysis reveals that the maximum amplitudes occurred at frequency of 40 Hz for inner spring and 35 Hz for composite spring which is nearer to theoretical natural frequency of inner spring 38.06 Hz.

Keywords: helical spring; rail road vehicle; modal analysis; natural frequency; harmonic response.

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Biographical notes: Manoj A. Kumbhalkar received his MTech in CAD/CAM in 2011 and Bachelor of Engineering degree in Mechanical Engineering in 2008 from Rashtrasant Tukdoji Maharaj Nagpur University, Maharashtra, India. He is a PhD Research Scholar of KDK College of Engineering, Nagpur registered in July 2012 in Mechanical Engineering in the same university. He worked as a reviewer of *Journal of Computer in Biology and Medicine* (Elsevier), *American Journal of Mechanical Engineering*, and *Journal of Mechanical Design and Vibration*. He is also an editorial board member of Universal Journal of Mechanical Engineering, USA. There are a total of 20 national/international conference/journal papers on his name. His research interests include machine design, computer aided design and finite element analysis. Currently he is working as an Assistant Professor in JSPM Narhe Technical Campus, Rajarshi Shahu School of Engineering and Research, Pune, Maharashtra, India.

D.V. Bhope received his PhD in Mechanical Engineering from Visvesvaraya National Institute of Technology, Nagpur, in 2005, MTech in Mechanical Design Engineering from Walchand College of Engineering, Sangli in 1988 and Bachelor degree of Engineering from Government College of Engineering, Amravati, India in 1986. He was the Chairman of Mechanical Engineering Board of Studies of Rashtrasant Tukdoji Maharaj Nagpur University from 2011–2015 and member of Mechanical Engineering Board of Studies of Gondwana University, Gadchiroli, India since 2012. He is a member of American Society of Mechanical Engineers, USA, Institution of Engineers, India, Indian Society of Technical Education and Tribology Society of India. There are a total of 47 national/international conference/journal papers on his name. His research interest includes machine design, finite element analysis, experimental stress analysis and theory of elasticity. Currently, he is working as a Professor in Rajiv Gandhi College of Engineering, Research and Technology, Chandrapur, India.

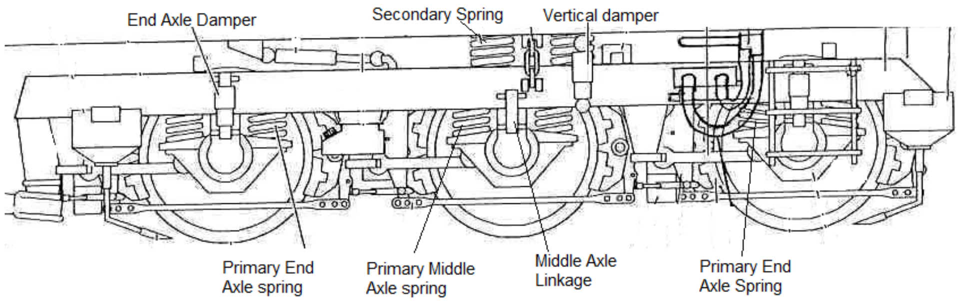
A.V. Vanalkar received his PhD in Mechanical Engineering in 2004 and MTech in Mechanism and Machine Design from Visvesvaraya National Institute of Technology, Nagpur in 1997 and 1990 respectively. He was a member of Mechanical Engineering Board of Studies of Rashtrasant Tukdoji Maharaj Nagpur University from 2011–2015. He is a life member of Indian Society for Technical Education. There are a total of 45 national/international conference/journal papers on his name. His research interest includes machine design, finite element analysis, and experimental stress analysis. Currently, he is working as a Professor in Department of Mechanical Engineering of MTech, Mechanical Engineering Design at KDK College of Engineering, Nagpur, India. He is a coordinator of consultancy cell in the department and he has carried out consultancy projects for Central Power Research Institute, Thermal Power Station, Koradi on 'Evaluation of transient thermal stresses in steam turbine shaft'.

1 Introduction

A mechanical or structural system is said to undergo forced vibration whenever external energy is supplied to the system during vibration. External energy can be supplied through either an applied force or an imposed displacement excitation. The applied force or displacement excitation may be harmonic, non-harmonic but periodic, non-periodic, or random in nature. The response of a system to a harmonic excitation is called harmonic response. The non-periodic excitation may have a long or short duration. The response of a dynamic system to suddenly applied non-periodic excitations is called transient response. The determination of the natural frequencies and modes of vibration of a system is known as modal analysis. A mode of vibration is characterised by vibration at a particular frequency (natural frequency) of all points of the system (Hingane and Sawant, 2013).

This paper discusses the behaviour of suspension system of rail road vehicle in dynamic way. The suspension system contains the primary spring between wheel and frame and secondary spring between frame and bogie. Primary suspension has concentric inner and outer primary spring at the middle wheel of the rail vehicle. The structure of rail vehicle contains the bogie whose whole weight is acting over all the suspensions, the frame which divides the primary and secondary suspension spring and the wheel base over which primary suspensions are mounted. The complete structure is divided into two frames which carry three axles and six axle box on each frame. Every axle box carries two primary suspensions but only the middle axle contains two concentric suspension springs. All the end axles contain the vertical damper between frame and axles which sustains the load acting on the spring but the middle axle does not contain any damper as it negotiates the curvature due sliding up to 16 mm. This paper focuses on the frequency response of concentric spring which carries the maximum load at dynamic condition.

Figure 1 Suspension system of rail road vehicle



2 Forces on primary suspensions

The spring has free play up to its solid height as it can achieve its deformation at static as well as in dynamic way up to this limit. It is observed that the end axle spring may deflected up to 76 mm as it achieve solid height after 76 mm, accordingly other springs deflections are restricted up to its height as end axle spring is 20 mm less in height than middle axle outer spring as shown in Table 1, technical specifications of primary spring. It is considered that springs may deform up to maximum height in dynamic condition at its maximum speed of 80 km/hr during curving, tracking or on uphill. By considering its deformation, the forces on each spring are calculated by considering the rails are smooth and there are no geometrical irregularities.

Table 1 Technical specification of spring set

Particulars	*Unit*	*Middle axle outer spring*	*Middle axle inner spring*	*End axle spring*
Free height	mm	258.6	252.4	238.8
Mean diameter (Dm)	mm	180.5	87.5	185
Coil diameter (d)	mm	31.5	16.5	36
No. of active coil (n)	-	3.5	7.5	3
Total no. of coil	-	5.0	9.0	4.5
Ultimate tensile strength	N/mm^2	1,720	1,720	1,720
Shear modulus	N/mm^2	78,500	78,500	78,500

The effects of deformation are acquired by each primary spring by reducing height of whole suspension system due to variation in free heights of every spring as shown in Table 1. As per the deflection of spring (Michalczyk, 2009), forces can be calculated for all springs by using following formula. The forces are also calculated for actual deformation of spring of rail road vehicle in static and dynamic condition, i.e., for the vehicle moving at straight track, curved track and at uphill. The deformations of spring are also verified in shed when rail vehicle is at rest, i.e., at static condition. The possible deflections and forces on each primary spring for reduced height are shown in Table 2 and forces calculated for actual deformation of springs are shown in Table 3.

$$\delta = \frac{8FD_m^3 n}{Gd^4} \tag{1}$$

Table 2 Deflection and force distribution on primary springs for its reduced height

Sr. no.	*Spring height (mm)*	*Particulars*	*Unit*	*Middle axle*		*End axle spring*
				Outer spring	*Inner spring*	
1	185	Load (W)	N	34,553.99	9,522.46	46,701.38
		Deflection (δ)	mm	73.6	67.4	53.8
2	180	Load (W)	N	36,901.41	10,228.87	51,041.66
		Deflection (δ)	mm	83.6	72.4	63.8
3	175	Load (W)	N	39,248.83	10,935.29	55,381.94
		Deflection (δ)	mm	83.6	77.4	63.8
4	170	Load (W)	N	41,596.24	11,641.71	60,884.95
		Deflection (δ)	mm	88.6	82.4	68.8
5	165	Load (W)	N	43,943.66	12,348.12	65,309.74
		Deflection (δ)	mm	93.6	87.4	73.8

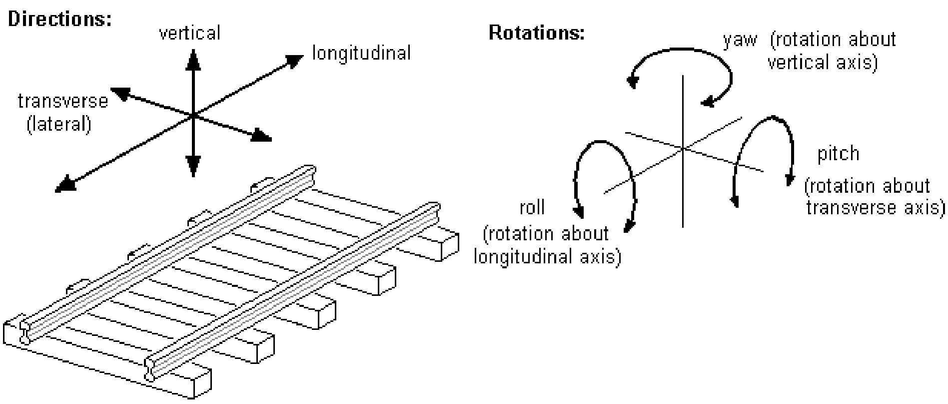
Table 3 Forces on primary spring for vehicle moving at straight track, curved track and at uphill

Sr. no.	*Condition*	*Particulars*	*Unit*	*Middle axle*		*End axle spring*
				Outer spring	*Inner spring*	
1	At straight track	Load (W)	N	31,268.7	9,351.46	40,451.38
		Deflection (δ)	mm	66.6	64.6	46.6
2	At curved track	Load (W)	N	35,250	10,512	47,740
		Deflection (δ)	mm	75	73	55
3	At uphill	Load (W)	N	43,052	12,902.4	62,148.8
		Deflection (δ)	mm	91.6	89.6	71.6

3 Experimentation using accelerometer

An accelerometer is a device that measures proper acceleration which is not necessarily the coordinate acceleration, i.e., rate of change of velocity. Instead, the accelerometer sees the acceleration associated with the phenomenon of weight experienced by any test mass at rest in the frame of reference of the accelerometer device. Single and multi-axis models of accelerometer are available to detect magnitude and direction of the proper acceleration or g-force, as vector quantity, and can be used to sense orientation (because direction of weight changes), coordinate acceleration, vibration, shock, and falling in a resistive medium, a case where the proper acceleration changes, since it starts at zero, then increases.

Figure 2 Direction of excitation of spring



To acquire reading of acceleration for suspension system, accelerometer device is placed on the top surface of primary suspension and check the vibration or excitation of the springs in vertical, longitudinal and lateral directions with respect to time in millisecond. The excitations of springs are recorded during running of rail road vehicle for distance of about 60 km on actual Indian track condition at a maximum speed of 80 km/hr. From the readings, it is observed that the lateral vibrations of spring are comparatively larger than the longitudinal vibration is shown in Figure 3. It means the suspension spring bears a lateral deflection at the curvatures which is limited to 16 mm for middle axle springs because only middle axle are responsible to negotiate the curvature for rail road vehicle. The vertical excitations are extremely high and peak point is maximum between 20 sec to 30 sec as shown in Figure 4, indicates the suspension spring which bears the vertical load have vibrations in running condition which can cause failure of spring.

Figure 3 Excitation of spring for acceleration vs. time in longitudinal and lateral direction for speed of 80 km/hr (see online version for colours)

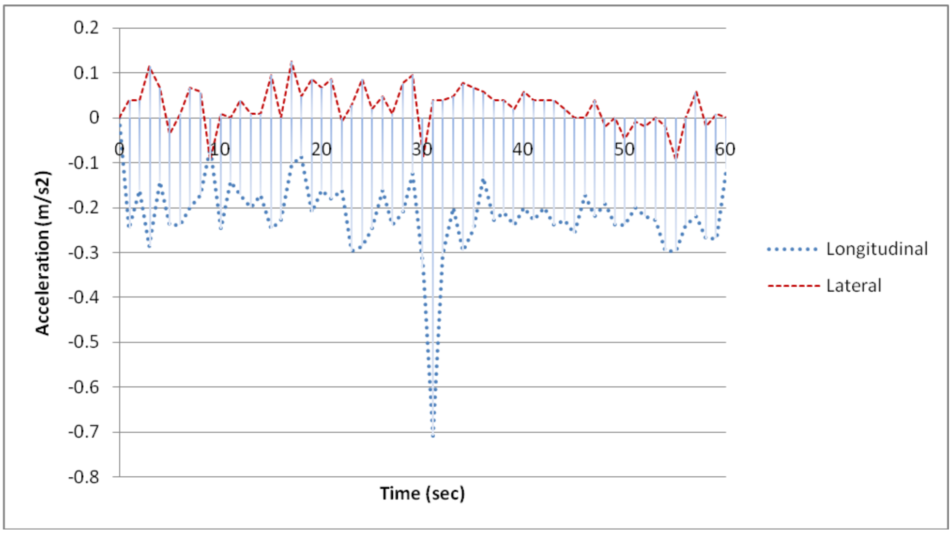
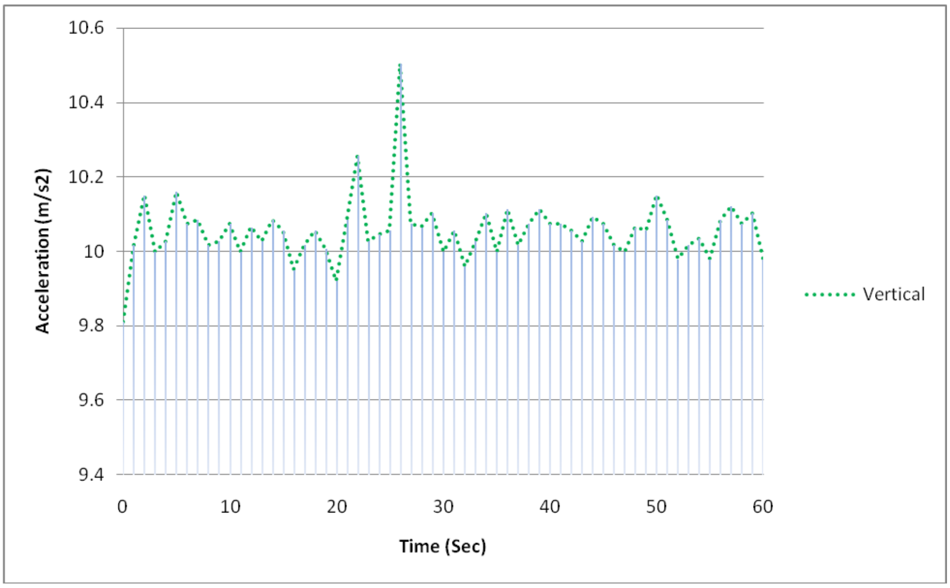


Figure 4 Excitation of spring for acceleration vs. time in vertical direction for speed of 80 km/hr (see online version for colours)



4 Modal analysis

A modal analysis determines the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component. It can also serve as a starting point for another, more detailed, dynamic analysis, such as a transient dynamic analysis, a harmonic response analysis, or a spectrum analysis. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions (Sudhakar Reddy and Madhusudhan Reddy, 2013).

This is probably the most common type of dynamic analysis and is referred to as an 'eigen value analysis' (Bratland and Rølvåg, 2008; Sreetharan, 2008). In addition to the frequencies, the mode shapes of vibration which arise at the natural frequencies are also of interest. These are the undamped free vibration response of the structure caused by an initial disturbance from the static equilibrium position. This solution derives from the general equation by zeroing the damping and applied force terms. Thereafter, it is assumed that each node is subjected to sinusoidal functions of the peak amplitude for that node. If the displacement vector {D} has the form,

$$\{D\} = \{A\}\text{sim}(\omega t) \tag{2}$$

where A is the amplitude of displacement for every node and w is the frequency of vibration.

Therefore, the velocity vector is,

$$\{\dot{D}\} = \{A\}\omega\cos(\omega t) \tag{3}$$

and the acceleration is,

$$\{\ddot{D}\} = -\{A\}\omega^2\sin(\omega t) \tag{4}$$

Substituting these into the general equation of motion yields the eigenvalue equation,

$$([K]-\lambda[M])\{A\} = \{0\} \tag{5}$$

where the eigenvalue, λ, is equal to ω^2, and {A} is the eigenvector associated with each value of λ. The total number of eigen values or natural frequencies is equal to the total number of degrees of freedom in the model. Each eigen value or frequency has a corresponding eigenvector or mode shape (Kelly and Knight, 1992). Since each of the eigenvectors cannot be null vectors, the equation which must be solved is,

$$([K]-\lambda[M]) = \{0\} \tag{6}$$

4.1 Modal analysis of inner and composite spring using finite element analysis

For the modal analysis of inner and composite primary suspension spring in finite element analysis tool, ANSYS, the imported geometry is fixed at the bottom end in all degree of freedom and allow vertical deformation on the top face and solve for the possible ten modes gives natural frequencies shown in Figure 6. A three-dimensional model of inner and composite spring prepares in modelling software pro-engineer and it is imported in IGES file format in ANSYS for analysis. The imported geometry is mesh with solid 187 ten-node tetrahedral element with fine meshing. For analysis purpose structural steel plates are placed at bottom and top surface of spring of material chromium vanadium. The excited model of inner and composite spring using finite element analysis is shown in Figure 5 and ten modes of natural frequencies in Hz is shown in Figure 6. The theoretical natural frequency of inner spring is 38.06 Hz calculated by using following relation.

$$f_n = \frac{d}{2\pi D^2 n}\sqrt{\frac{6Gg}{\rho}} \tag{7}$$

Figure 5 Finite element model of inner and composite spring showing excitation during modal analysis (see online version for colours)

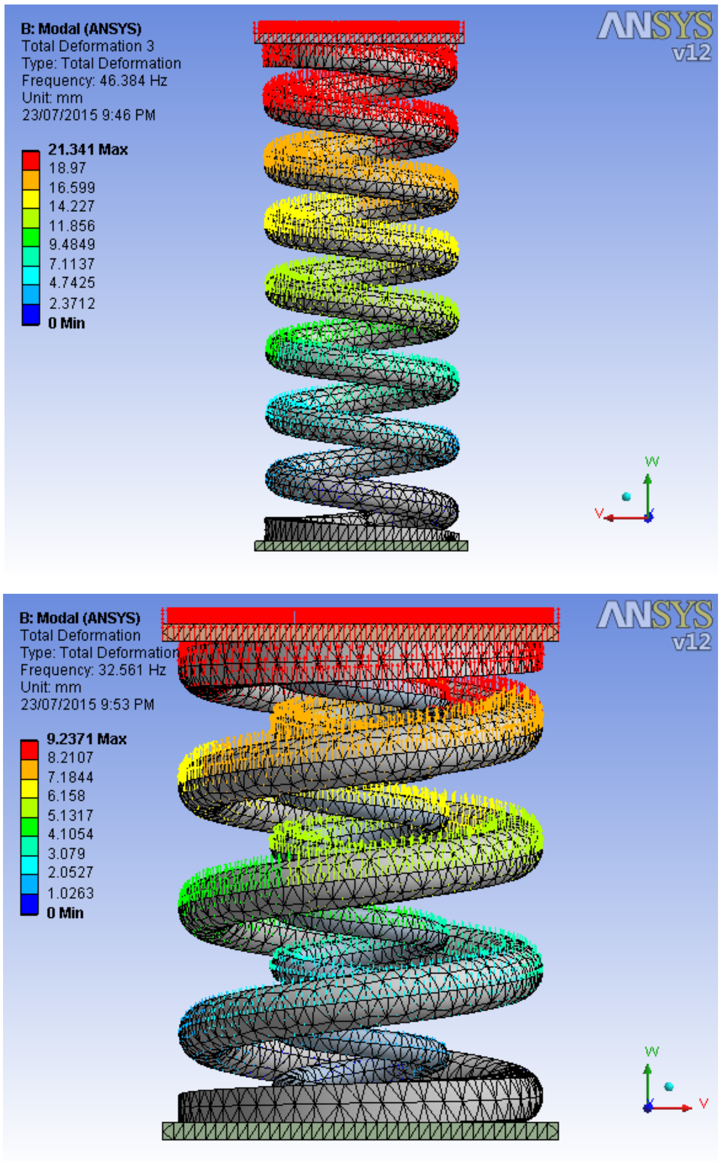
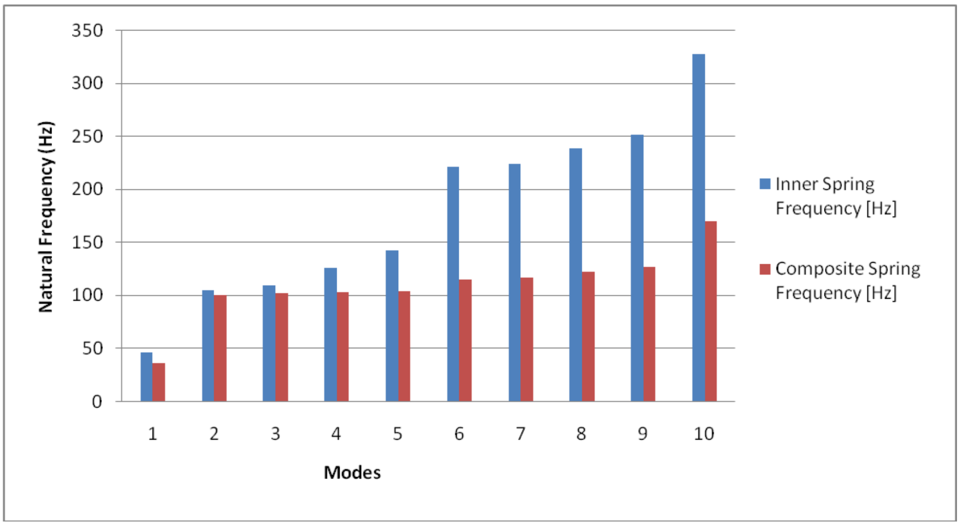


Figure 6 Natural frequency of inner and composite spring in different modes (see online version for colours)



5 Harmonic response analysis

In a structural system, any sustained cyclic load will produce a sustained cyclic or harmonic response. Harmonic analysis results are used to determine the steady-state response of a linear structure to loads that vary harmonically with time, thus enabling to verify whether or not the designs will successfully overcome resonance, fatigue, and other harmful effects of forced vibrations (Triana and Fajardo, 2013).

This analysis technique calculates only the steady-state, forced vibrations of a structure. The transient vibrations, which occur at the beginning of the excitation, are not accounted for in a harmonic response analysis.

In this analysis, all loads as well as the structure's response vary harmonically at the same frequency. A typical harmonic analysis will calculate the response of the structure to cyclic loads over a frequency range (a sine sweep) and obtain a graph of stress and acceleration response quantity versus frequency. 'Peak' responses are then identified from graphs of response versus frequency are then reviewed at those peak frequencies. For the harmonic response, the mode superposition method is used which sums factored mode shapes (obtained from a modal analysis) to calculate the harmonic response (Yıldırım, 2009).

This type of analysis is of interest when the steady state response of a structure to a harmonic force input at a given frequency is required. The response may be needed for a range of frequencies. In a frequency response analysis, the frequency of the response to a harmonic input is also harmonic and occurs at the same frequency. The forcing function for undamped system can be defined as:

$$\{F\} = \{F_o\} e^{i\omega t} \tag{8}$$

where F_o is the peak force amplitude and ω is the harmonic frequency.

The nodal displacement therefore has the form

$$\{D\} = \{D_o\} e^{i\omega t} \tag{9}$$

with velocity

$$\{\dot{D}\} = \{D_o\} i\omega e^{i\omega t} \tag{10}$$

and acceleration

$$\{\ddot{D}\} = -\{Do\}\omega^2 e^{i\omega t} \tag{11}$$

The general equation of motion results in

$$\left(-\omega^2[M] + i\omega[C] + [K]\right)\{D_o\} = \{F_o\} \tag{12}$$

which shows that the displacement, $\{D_o\}$ is clearly a function of frequency, damping and force amplitudes.

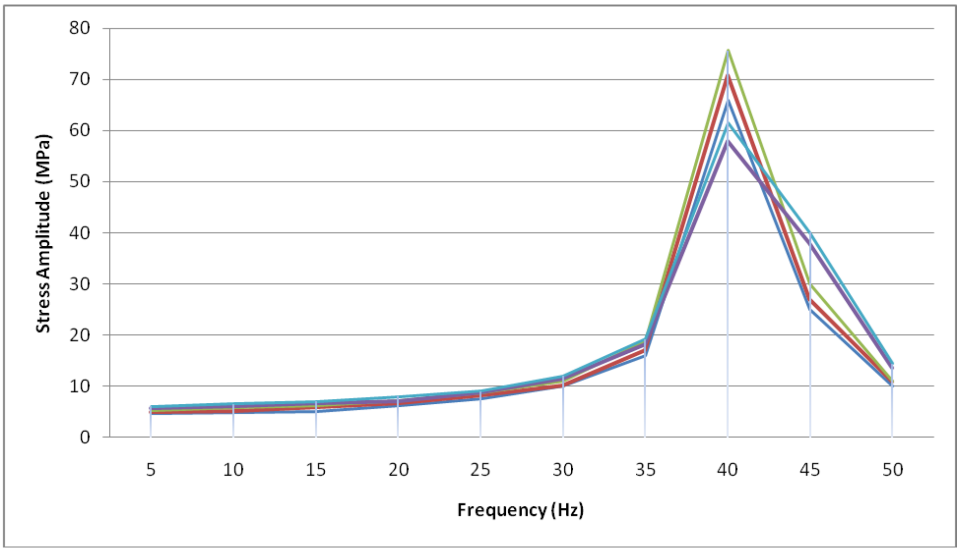
5.1 Harmonic response of inner suspension spring using finite element analysis

For harmonic response of inner spring the load calculated above for specific deformation are considered as harmonic forces and are applied on the top face of spring in ANSYS for the frequency range of 0 Hz to 50 Hz. The analysis has been solved for ten solution interval and five different harmonic forces using mode superposition method for frequency response of stress and acceleration. From the harmonic analysis, it is observed that the inner spring has maximum amplitude at frequency of 40 Hz for stress and acceleration is shown in Figure 7 which is nearer to theoretical natural frequency.

5.2 Harmonic response of composite suspension spring

For harmonic response of composite spring, the combine load of inner and outer spring calculated above are applied on the top face of spring in ANSYS for the frequency range of 0 Hz to 50 Hz and again solve for ten solution interval using mode superposition method. The results obtained in the form of frequency versus stress and acceleration amplitude states the maximum amplitude of stress and acceleration changes at 35 Hz for composite spring. The response of maximum amplitude is shown in Figure 8.

Figure 7 Harmonic response of inner spring for frequency versus stress and acceleration amplitude for different five harmonic forces (see online version for colours)



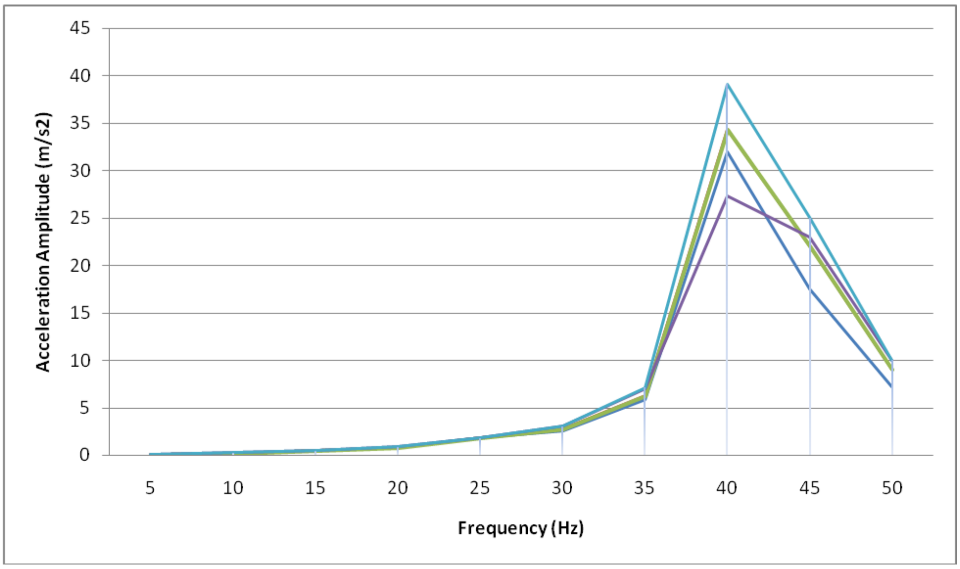
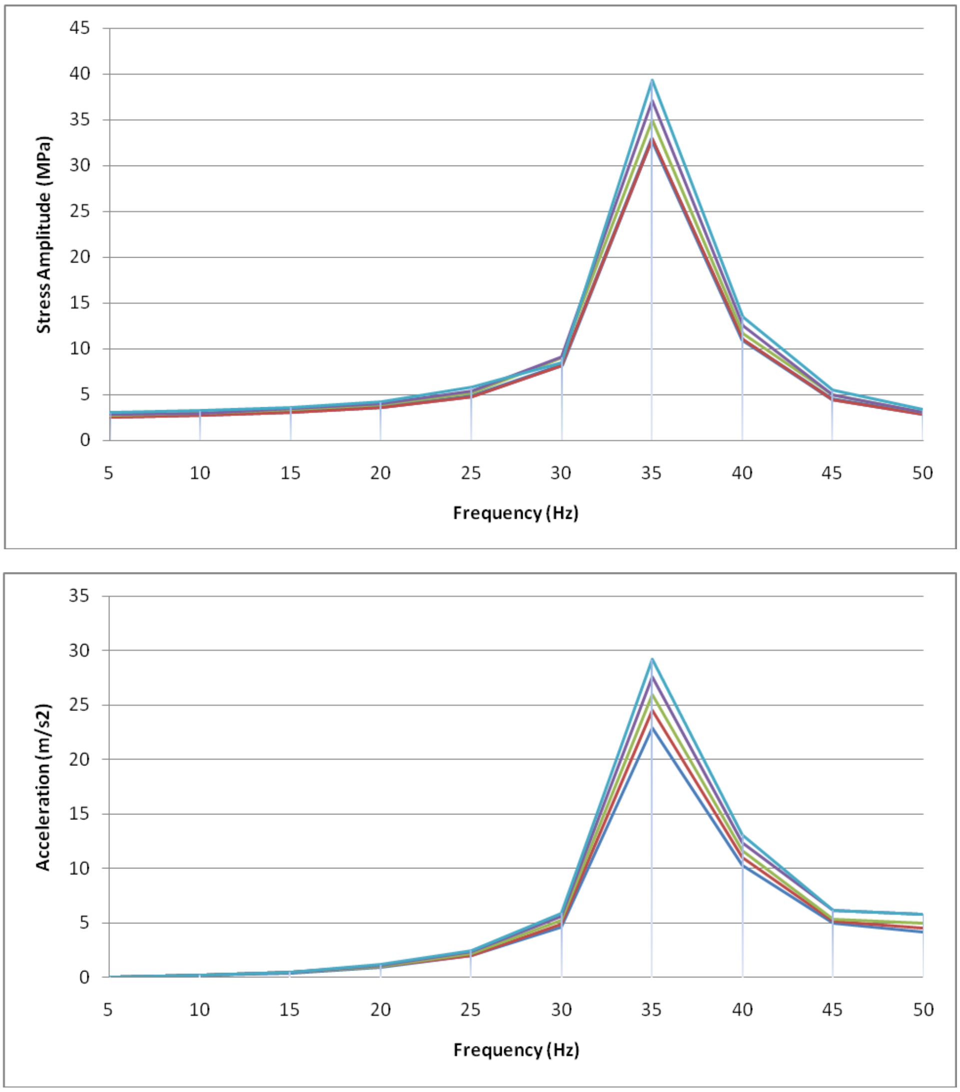


Figure 8 Harmonic response of composite spring for frequency versus stress and acceleration amplitude for different five harmonic forces (see online version for colours)



7 Conclusions

The structure of suspension system of rail road vehicle is studied for frequency response. The outer primary spring is deformed first as its height is 2 mm more than inner spring and 20 mm more than end axle spring. The middle axle has undamped composite assembly of suspension system having much excitation and may cause failure of inner spring. The forces caused on springs are calculated for input as harmonic forces. The experimental result shows the peak amplitude of acceleration is occurred between times of 20 sec to 30 sec for vertical excitation of suspension system.

Finite element analysis is the technique to find frequency response for stress and acceleration for various harmonic forces. At first, the natural frequencies are found out for inner and composite spring for ten modes which will be the input for harmonic analysis. The first natural frequency of inner spring is 46 Hz and of composite spring is 36 Hz. The harmonic analysis states that, the maximum amplitude of stress and acceleration is occurred at 40 Hz for inner spring and 35 Hz for composite spring analysed for the frequency range of 0 Hz to 50 Hz using mode superposition method. This paper deals with the response of suspension system subjected to harmonic excitations. From the response, it is observed that the inner spring has more excitation in dynamic condition and may cause failure due to repeated loading.

8 Future scope

The suspension spring may analyse for free and forced vibration of spring mass damper system for the excitation of displacement, velocity and acceleration theoretically and also by using mathematical tool MATLAB. FFT analyser is the device to get the excitations with respect to frequencies can also be used for analysis purpose which again by analysed by using MATLAB. For the analysis of complete suspension system, the damping could be considered at the system as damped vibration system which is not consider in this paper as middle axle has no damping effect. The complete system will be analysed for two degree of freedom spring mass damper system.

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