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TECHNICAL ARTICLE—PEER-REVIEWED

Failure Analysis of Primary Suspension Spring of Rail Road Vehicle

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Abstract WAG-9-type electric rail vehicle, of Indian Railways' fleet used for goods train hauling and maintained at Ajani, Nagpur Electric Loco Shed of central railway, has a history of frequent failure of middle axle primary inner suspension spring. The study of failures revealed that this specific component fails at a very high rate. The failure investigation starts the experimental spectroscopy analysis to find chemical composition for different failed specimens of springs, and it is observed that all parameters are within the recommended range. Also the stiffness of primary middle axle and end axle suspension springs has been checked on spring testing machine to measure deflection of spring, and it is as per recommended values. Further static stress analysis is carried out using analytical and finite element analysis for various phases of operation of rail vehicle like straight track, curved track and also for tractive effort. The static stress analysis has not revealed the cause of failure, and hence the dynamic analysis is performed. For dynamic analysis, dynamic model of suspension system is considered and analyzed using analytical method, finite element method and using MATLAB Simulink model. The vibration response of actual suspension system is also measured using FFT analyzer. It has been seen that the frequency of excitation and the natural frequency of the system are very close to each other which has resulted into suspension vibration amplitude of 6–8 mm. Fatigue analysis is carried out using finite element method to investigate the effect of dynamic loading on the failures of suspension spring. This analysis revealed that the middle axle inner suspension spring has finite life and due to which spring failure occurs earlier.

Keywords Indian Railway · Primary suspension spring · Vibration · Fatigue analysis · Spectroscopy analysis

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Introduction

A rail road vehicle includes bogies, frame, suspension elements, bushings, bearings, as well as other components. The bogies, such as the one shown in Fig. 1, also represent complex systems that include frames and wheel sets that can have independent motions. The wheel sets, which can rotate freely about their own axes, are connected to the frame using primary suspensions, while the frame is connected to the car body using the secondary suspensions, safety chain and secondary dampers as shown in Fig. 1. The forces of dynamic interaction between the wheels and the rails significantly influence the dynamics and stability of railroad vehicles due to friction between the rotating wheels and the rails during motion. Clearly, a railroad vehicle system consists of a large number of interconnected

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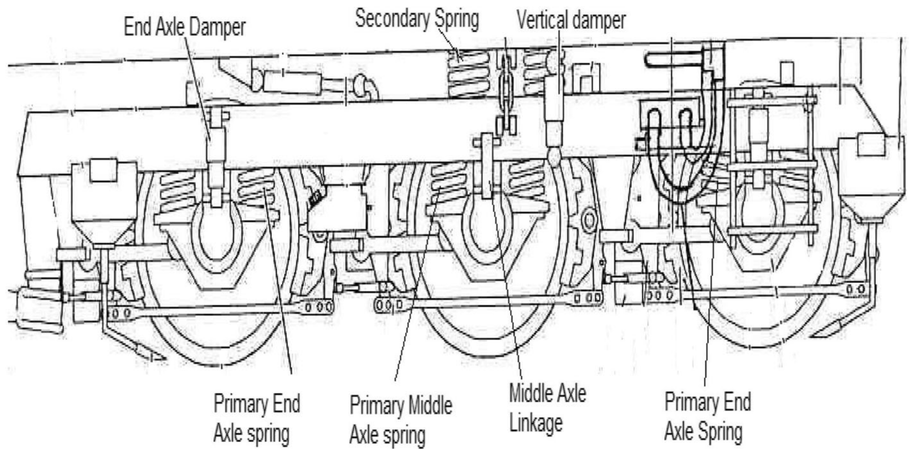


Fig. 1 Primary and secondary spring set assembly (Courtesy: Indian Railway)

Table 1 Technical specifications of spring set (Courtesy: Indian Railway)

		Primary suspension		
Particulars	Unit	Middle axle outer spring	Middle axle inner spring	End axle spring
Free length (l_f)	mm	258.6	252.4	238.8
Outer diameter (D_o)	mm	212	104	221
Inner diameter (D_i)	mm	149	71	149
Mean diameter (D_m)	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
Pitch	mm	64.65	31.5	68.22
Helix angle	Degree	8.15	9.10	8.062
Modulus of rigidity (G)	N/mm^2	78,500	78,500	78,500
Stiffness (k)	N/mm	470	144	868
Maximum tensile strength (σ_u)	N/mm^2	1720	1720	1720
Yield shear strength (σ_y)	N/mm^2	878	878	878

components that experiences dependent/independent motion. These components are connected by force elements such as springs, dampers, and bushings as well as joints that impose restrictions on the motion of the system [1].

The axle box is attached to each of the axle on which all the primary springs are mounted. Each axle box has the same space to mount the primary inner and outer spring, since middle axle contains both the spring assemblies. The outer diameter of the end axle spring is larger, and the height is lower than the middle axle outer spring in uncompressed state. The inner spring is mounted only in the middle axle wheel set which has the composite assembly of inner and outer spring. The free height of middle axle outer spring is 20 mm larger than the end axle spring and also having difference of 9 mm in their respective outer diameters. The free height of middle axle primary inner spring is 2 mm lesser than the free height of outer spring. It means that the total load of loco first acts on middle axle outer spring and then it acts over middle axle inner spring and after that it is distributed over the end axle springs. The technical specifications of suspension spring sets are given in Table 1.

The problem identified in the suspension spring of WAG-9 locomotive is discussed as follows:

- The WAG-9 locomotive undergoes the minor maintenance with interval of 45 days, intermediate maintenance with interval of 90 days and major maintenance with the interval of 18 months or approximately 1,50,000 km whichever occurs earlier. Normally overall of complete suspension system comes under major maintenance.
- It is reported by Electric Loco Shed, Ajni, Nagpur that due to the failure of middle axle primary inner suspension springs, WAG-9 locomotive is called back

to the loco shed within 90 days and it undergoes the unscheduled maintenance requiring the span of minimum 3 days, though only the middle axle primary inner suspension spring has failed. This has increased the down time of WAG-9 locomotive.

- Hence based on the problem identified, the objective of this research work is to investigate the cause of frequent failures of the middle axle primary inner suspension spring and to suggest the necessary modifications to avoid their failures. It is also proposed to receive the feedback from Electric Loco Shed after implementation of suggested modifications.

Experimental Spectroscopic Analysis of Spring Material

In the initial phase of investigation, it is felt necessary to ensure the chemical composition of the failed springs of GBD make in accordance with the recommended chemical composition of spring material [2]. As per the data provided by loco shed, the spring material is chromium vanadium (50CrV4) having the chemical composition as given in Table 2. An effort is made to determine the chemical composition of failed spring using spectrometer of make WAS, model Foundrymen to ensure that the failed springs are having recommended chemical composition or not.

Total five failed spring specimens of different WAG-9 locomotive are used for experimentation which has been failed in the gap of some instances. The surface of specimens is polished using emery paper of grade 80 and placed on flat base of spectrometer. The emitted ray from spectrometer sparks on spring specimen, and it has provided the results in the form of chemical composition in percentage as shown in Table 2. From the chemical composition of all specimens revealed by the spectrometer, it is seen that the chemical composition of failed springs is well within the recommended range. This investigation revealed that the springs are not failed due to improper material composition. Hence, the cause of failure is observed to be different than this and hence it is felt necessary to extend the failure analysis using other techniques.

Table 2 spectrometer readings for failed middle axle inner suspension spring

Loco no.	31097	31087	31068	31072	31105
C (0.47–0.55)	0.47	0.475	0.46	0.52	0.45
Si (0.15–0.40)	0.24	0.15	0.20	0.35	0.18
Mn (0.7–1.10)	0.90	0.74	0.81	0.93	0.82
P (0.035 max.)	0.024	0.023	0.025	0.034	0.034
S (0.019–0.035)	0.019	0.020	0.023	0.035	0.031
Cr (0.9–1.2)	1.07	1.09	1.03	0.91	0.97
V (0.1–0.2)	0.103	0.12	0.14	0.101	0.13

Experimental Determination of Stiffness of Suspension Spring

The recommended stiffness of suspension springs, i.e., middle axle outer spring, middle axle inner spring and end axle is given in Table 1 as 470, 144 and 860 N/mm, respectively. It is felt necessary to determine the stiffness of all the suspension spring experimentally [3]. A spring testing machine of ENKAY make is used to determine the stiffness of GBD make springs. The load is varied from 100 kgf (981 N) to 600 kgf (5886 N), and the deflection of spring is determined. This analysis is carried out on each of the five middle axle inner springs, middle axle outer spring and end axle spring. The average deformations and average stiffness as per loading are given in Table 2. The experimental setup is shown in Fig. 2. The average stiffness of middle axle inner spring, middle axle outer spring and end axle spring is found out to be 145, 475, 870 N/mm which is in close agreement with the recommended stiffnesses of respective springs. This analysis has ensured that the springs used in the suspension system are having the recommended stiffnesses. This investigation ensured that the

Fig. 2 Testing deflection of spring on spring testing machine

design and manufacturing of spring is as per the given specifications.

Analytical Stress Analysis of Suspension Spring

Primary suspension springs are mounted on axle housing on each wheel of WAG-9 rail vehicle to absorb shocks and vibrations. A three axle rail vehicle is divided in two parts, viz. end axle and middle axle. The suspension springs mounted on middle axle housing have composite assembly of inner and outer spring having different coil and mean diameter as compared with end axle springs.

The major part of rail vehicle is three axles, three motor Co–Co bogie assemblies. As per WAG-9 manual, an entire weight of 123 tonne (1206.63 KN) of locomotive is supported by two frame assemblies and provides a means for transmission of the tractive effort to the rails. A rail vehicle is designed to withstand the stresses and vibrations resulting from normal rolling stock applications. An important function of the frame is to absorb and isolate shock caused by variations in the trackbed. The suspension systems minimize the transmission of these shocks to the locomotive under frame.

The maximum stress in the wire may be computed by superposition of the direct shear stress and the torsional shear stress. The maximum torsional shear stress 'τ' occurs at the inside fiber of the spring and total deflection of spring 'δ' [4] as,

$$\tau = \frac{8FD}{\pi d^3} + \frac{4F}{\pi d^2}; \quad \delta = \frac{\partial U}{\partial F} = \frac{8FD^3N}{d^4G}$$

To determine the stresses in all suspension springs, it is necessary to determine the forces acting on the springs for various running conditions and for various phases of operation of rail road vehicle. The forces acting on the primary suspension springs are investigated for following cases.

1. *CASE-I* For Rail Vehicle Moving on Straight Track
2. *CASE-II* For Rail Vehicle Moving on Curved Track
3. *CASE-III* Analysis for Tractive Effort

The total weight of loco is 123 tonne (1206.63 KN). The unsprung weight per wheel set is 3.984 tonne (39.08 KN). Hence, total unsprung weight for 6 wheel sets is 23.904 tonne (234.49 KN). Hence, the net weight distributed on all axles is 99.096 tonne (972.13 KN). This net weight is distributed over each of frontal and rear wheel sets through front and rear wheel frames, and hence the net weight distributed on each frame is 49.548 tonne (486.06 KN). The force corresponding to the weight of 49.548 tonne (486.06 KN) is acting over each middle axle primary outer spring as the free height of this spring is maximum than all other springs, i.e., 2 mm more than middle axle inner spring and 20 mm more than end axle springs. Later the force acts on middle axle inner spring and after 18 mm deflection of middle axle spring (inner and outer); the force is distributed over end axle springs. As the free lengths of all primary springs are not equal, the forces on each spring are determined according to their individual deflections. Total force on each spring is obtained by considering forces acting due to deflection from free height to maximum deflection at static condition.

The vehicle passing over the curve continuously changes its direction over a curve. Due to inertia, the vehicle tends to continue moving in the straight line, but the forced change in direction of the movement by track gives rise to lateral acceleration acting outwards which is felt by the vehicle. Due to this centripetal force, the vehicle will experience lateral forces when it travels on the curved track. To negotiate the loco on curves on railway track, only the middle axles are given the free play of about 16 mm in a lateral direction perpendicular to the direction of motion of vehicle. It should be noted that the axle does not turn about the vehicle, but there is a sliding of the middle axle which helps in negotiating the curve, while moving on the curved track the vehicle is subjected to additional lateral force because of centripetal force which causes the lateral force on the spring with its bending moment maximum at the top end. The shear stresses are evaluated for various radii of curvatures, i.e., from 175 to 875 m for corresponding safe speeds of loco, i.e., from 45 to 100 km/h. Also apart from safe speed, the stresses are also determined at higher speeds for each curve radius to check its design limitations, but they are observed to be safe. The forces, deflections and shear stresses in middle axle outer spring, middle axle inner spring and end axle spring are given in Table 3, and it indicates that the shear stress has maximum magnitude of 587.91 and 586.76 N/mm^2 for radius of curvatures of 219 and 350 m, respectively. Figure 3 shows the forces when the rail vehicle moving on the curved track. The gyroscopic effect of wheel with gear and traction motor with pinion has been also considered.

The force which a rail vehicle can exert when pulling a train is called as its tractive effort and it depends upon various factors. As per WAG-9 rail vehicle manual, TE/BE meters show the readings of tractive and braking effort at various speeds which has been recorded from the driver cab as shown in Fig. 4. From the reading of tractive and braking effort, it is observed that the tractive effort of 460 KN is maximum during starting of vehicle and constant while running. As per observation on locomotive during starting from rest position, the front portion of bogie frame gets upward and rear portion gets lowered, hence the bogie gets tilted about lateral axis. The tractive effort is the

Table 3 Comparison of analytical and FE analysis results for middle axle primary inner and composite spring at straight track, curved track and for tractive effort

Spring	Phase of operation	Force (N)	Analytical results: Deflection (mm)	Analytical results: Shear stress (MPa)	FEA results: Deflection (mm)	FEA results: Shear stress in plane (MPa)	FEA results: Max. shear stress (MPa)	FEA results: Maximum principal stress (MPa)
Middle axle inner spring	At straight track	9351.46	64.6	598.36	66.06	592.25	656.28	745
Composite spring	Max axial load at curved track (due to superelevation)	39,631	63.77	587.91	57.92	577	690.64	892.75
Composite spring	(1) Axial load	39,631	63.77	587.91	60.205	582	737.5	966.53
	(2) Lateral load at curvature (due to superelevation)	6981.52	…	…	12.92			
Middle axle inner spring	Tractive effort	11,030	76.6	705.79	77.9	698.55	774.08	907

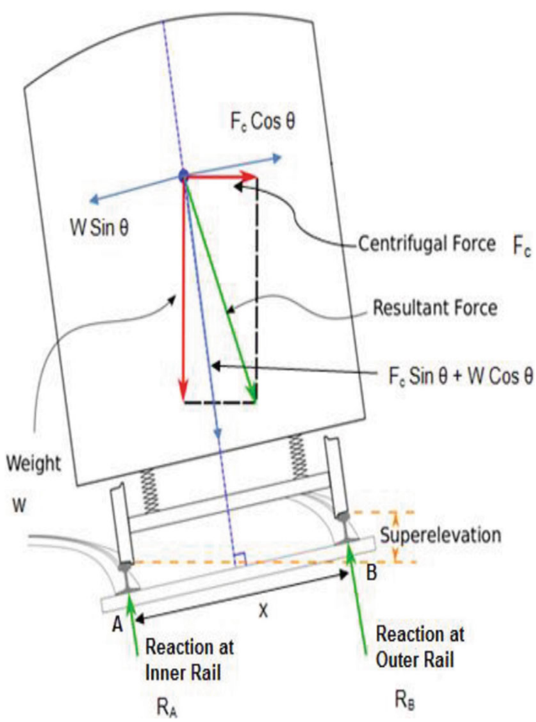


Fig. 3 Rail vehicle on curved track

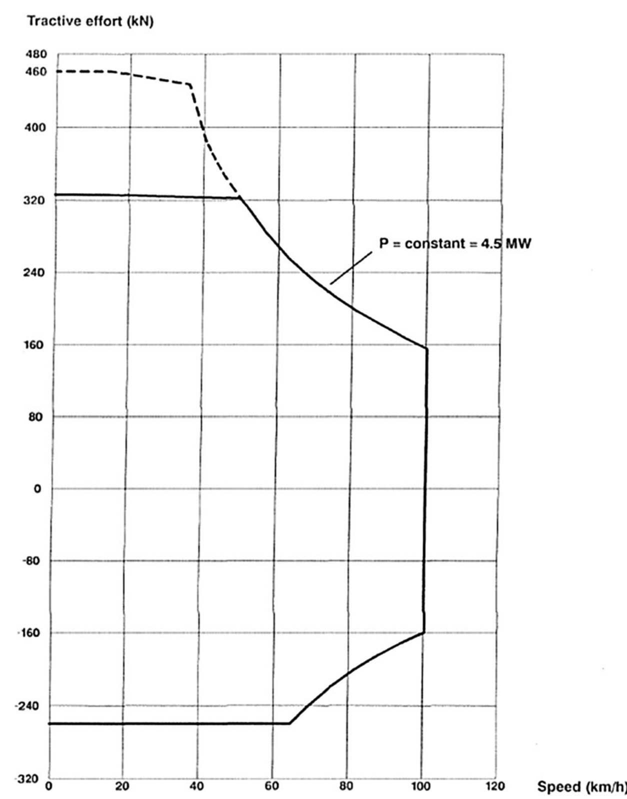


Fig. 4 Tractive and braking effort characteristics for various loco speeds (Courtesy: Indian Railway)

longitudinal suddenly acting force which is hauling capacity of locomotive, and it depends on the coefficient of friction and weight on the driving wheels. The coefficient of friction varies from 0.1 at high speed to 0.3 at low speed, but in some cases it is reaches to 0.5 due to the effect of rail adhesion [5].

Therefore, this longitudinal force of 460 KN corresponding to tractive effort is acting on the bogie frame during starting of vehicle and about 300 KN force is acting during running condition which will cause the additional deflection of suspension springs and it leads to increase in stress magnitudes of suspension spring.

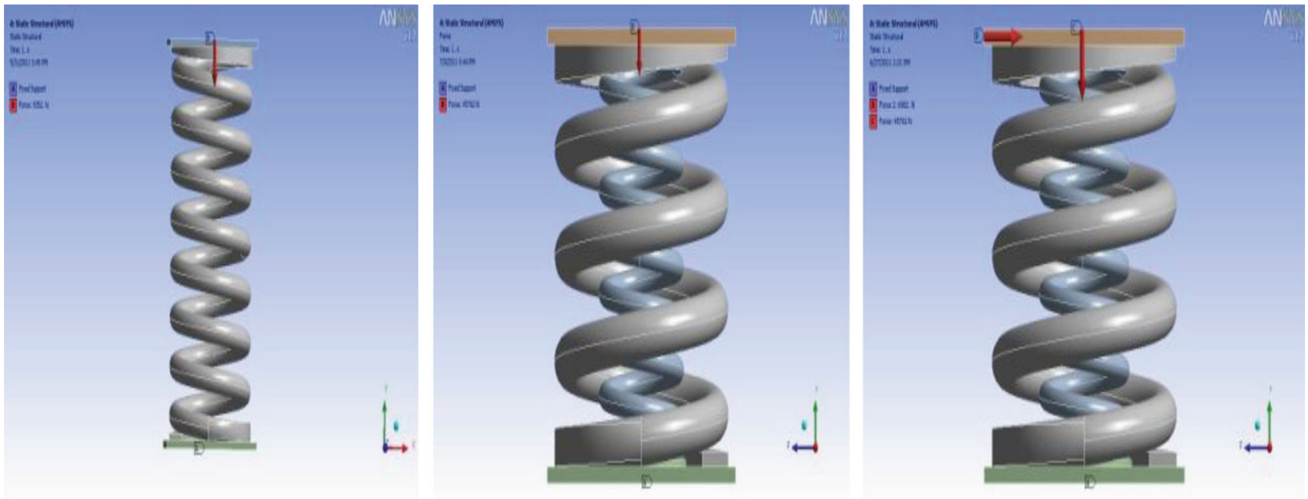


Fig. 5 Boundary condition for middle axle inner and composite suspension spring for rail vehicle moving on straight track, curved track and tractive effort

FE Modeling and Analysis of Middle Axle Primary Suspension Spring

Using the technical specifications of spring, FE analysis of inner suspension spring and concentric assembly of inner and outer suspension spring have been carried out in ANSYS 12.0. A higher-order 3-D, 10-node SOLID 187 element having three degrees of freedom at each node, i.e., translations in the nodal x, y, and z directions, is used for FE analysis. The static structural analysis has been carried out for the earlier mentioned loading cases I, II and III. Initially the inner spring is analyzed by considering the loading cases I, II and III as discussed in static force analysis. The boundary condition and FE results for inner and also for composite spring considering forces on straight track and curved track are shown in Fig. 5.

Further, finite element analysis is carried out for axial and lateral forces acting on the spring. Initially, inner spring is analyzed by considering only vertical axial forces experienced on straight track and considering tractive effort. The investigation further calls for the FE analysis of composite spring for lateral loading, as composite structure of spring is subjected to axial and lateral load on track curvature.

Due to centripetal force induced due to track curvature, the rail vehicle and the suspension spring will experience lateral force given as,

Centripetal force, Fc = mV^2/R = 167,556.54 N
Lateral force on each frame = 83,778.27 N
Lateral force on primary suspension spring = 6981.52 N

The result of shear stress for inner suspension spring and for composite suspension spring is shown as contour plots in Fig. 6. The maximum shear stresses and maximum principal stresses for inner and composite suspension spring for the case of straight track, curved track and tractive effort are given in Table 3.

The stresses in the inner suspension spring are evaluated for the three cases using analytical and FE analysis as discussed earlier for static load condition. Thus, the induced shear stresses and principal stresses are well below the yield shear stress and yield tensile stress. Hence, the static analysis does not reveal the cause of failure and therefore it is found essential to focus on other factors causing the failure. Hence, it is decided to carry out further types of analysis on the suspension spring like, dynamic and fatigue analysis.

Dynamic Analysis of Suspension System

Determination of the natural frequencies and modes of vibration of a system is known as modal analysis. A mode of vibration is characterized by vibration at a particular frequency (natural frequency) of all points of the system [6]. An objective of study of railway vehicle dynamics is to develop a mathematical model describing time response for its amplitude. A six axle locomotive with primary and secondary suspension system is shown in Fig. 7.

Primary and secondary suspension is necessary for vibration isolation and absorption of shock loads. Vertical dampers restrict rate of the vertical rebound of the locomotive bogie, and rebound limit chains restrict the amount of vertical rebound of the locomotive bogie. Pitch rate of bogie is controlled by yaw (longitudinal) dampers.

Railway vehicles are dynamically complex multibody systems. Each mass within the system has six dynamic degrees of freedom corresponding to three displacements (longitudinal, lateral and vertical) and three rotations (roll,

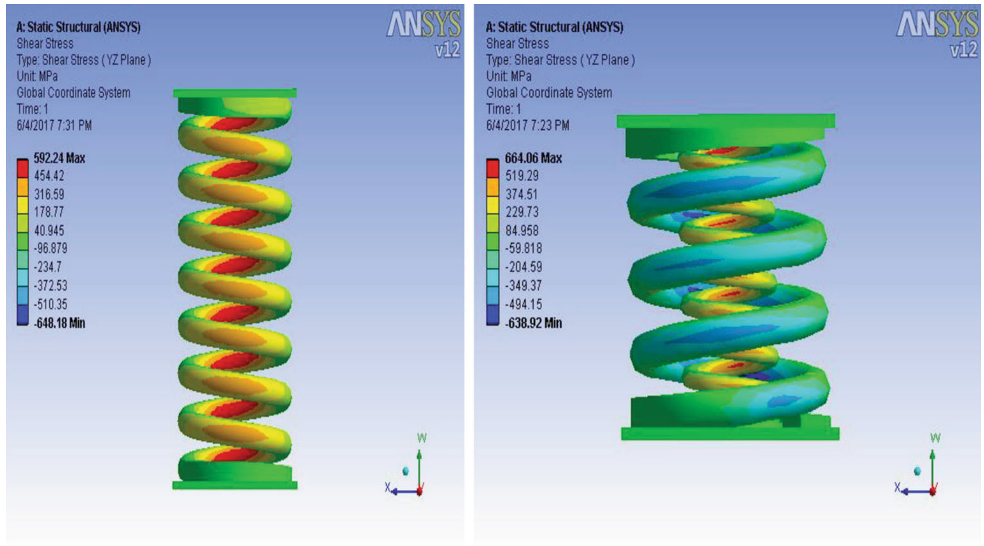


Fig. 6 FE analysis results of middle axle inner and composite suspension spring for rail vehicle on straight track and curved track for axial and lateral loading

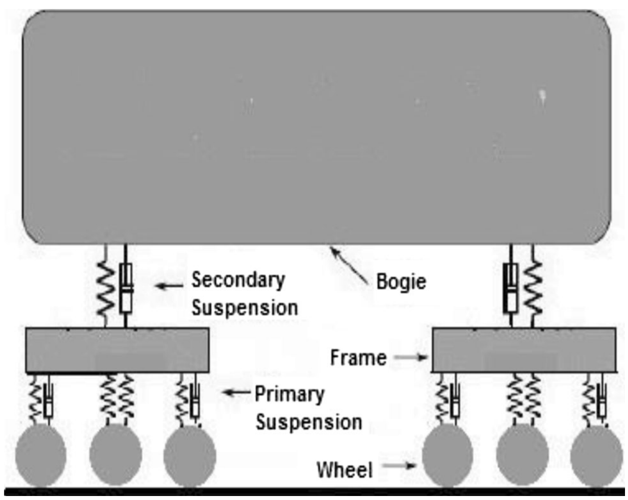


Fig. 7 Six axle locomotive with primary and secondary suspension system

pitch and yaw). Dumitriu [7] in 2015 developed the dynamic model involving longitudinal and lateral dampers between frame and bogie. In WAG-9 locomotive, the primary suspension springs undergoes mainly the vertical displacement. Hence, it is felt necessary to develop the dynamic model only for vertical displacement.

Model for Vertical Vibrations of Rail Vehicle Suspension

Damping control in the primary suspension is applied to the vertical axle-box dampers to suppress the vertical vibrations [8, 9]. The present research has the focus on the oscillation of suspension system in vertical direction which is responsible for the displacement of rigid frame and vehicle body. In dynamic condition, the displacement of rigid frame and bogie occurs by base excitation due the irregularities in rail and relative displacements between the wheels and the rails which are small in the sub-critical range of velocities and, hence, the influence of the contact nonlinearities can be neglected and, consequently, a linear model is recommended for studying the vertical vibrations of the rail vehicle [7, 10]. Hence to examine relative displacement between primary and secondary suspension, a model is formulated for two degree of freedom system with base excitation.

Equations of motion for dynamic response of vehicles can be derived which encompass the elastic and damping forces generated by the suspension. For this analysis, the stiffness and damping coefficients of primary and secondary suspension system are given in Table 4.

For the suspension system, a 1/4th model is used to simplify the problem which involves multiple springs and dampers. Figure 8 illustrates 1/4th model of vehicle body which shows suspension with primary and secondary springs and dampers attached with them and also illustrates the free body diagram for spring forces and viscous forces.

The equation of motion for mass 1 is given as,

$$m_1\ddot{x}_1 = -c_2(\dot{x}_1 - \dot{x}_2) + c_1(\dot{y} - \dot{x}_1) - k_2(x_1 - x_2) + k_1(y - x_1) \qquad \text{(Eq 1)}$$

Substituting relative displacement terms in Eq 1

$$m_1\ddot{u} + (c_1 + c_2)\dot{u}_1 - c_2\dot{u}_2 + (k_1 + k_2)u_1 - k_2u_2 = -m_1\ddot{y} \qquad \text{(Eq 2)}$$

The equation of motion for mass 2 is given as,

$$m_2\ddot{x}_2 = c_2(\dot{x}_1 - \dot{x}_2) + k_2(x_1 - x_2) \qquad \text{(Eq 3)}$$

Substituting relative displacement terms in Eq 3

Table 4 Stiffness and damping coefficients of primary and secondary suspension system

Spring	Stiffness (N/mm)	Damper	Damping coefficient (N s/m)
Primary middle axle outer spring	470	Yaw damper	30,000
Primary middle axle inner spring	144	Horizontal damper	70,000
Primary end axle spring	868	Inclined axle damper	50,000
Secondary suspension spring	612	Vertical damper	1,10,000

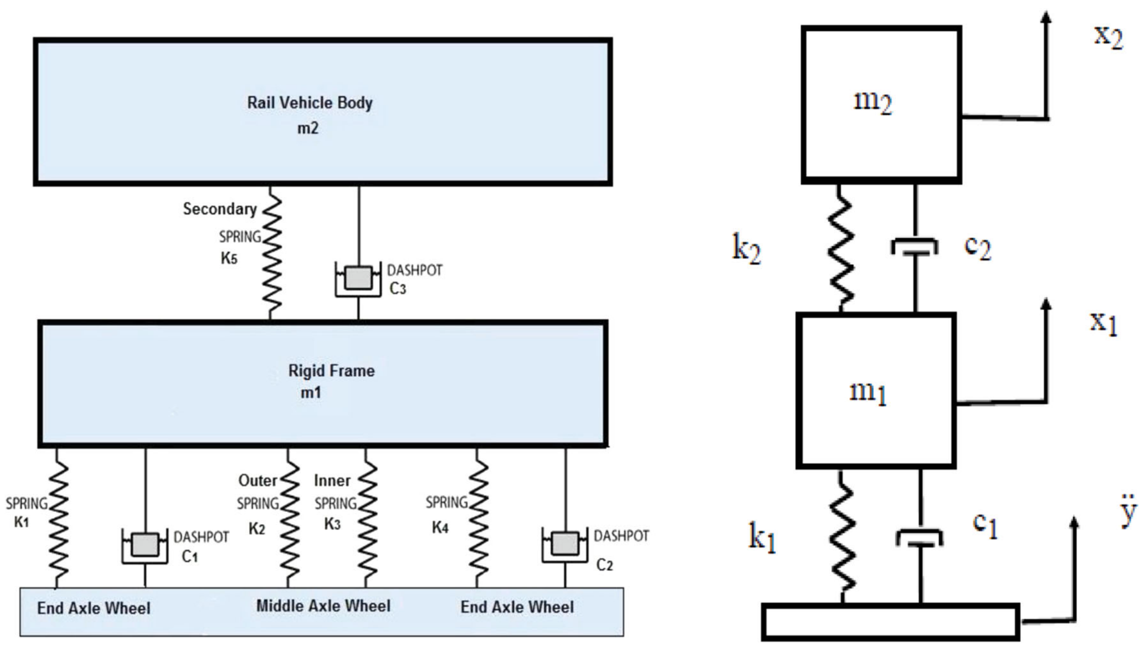


Fig. 8 1/4th model of rail vehicle with its free body diagram

$$m_2\ddot{u}_2 + c_2\dot{u}_2 - c_2\dot{u}_1 + k_2u_2 - k_2u_1 = -m_2\ddot{y} \quad \text{(Eq 4)}$$

The equations of motion for mass 1 and mass 2 are expressed in matrix form as follows.

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{Bmatrix} \ddot{u}_1 \\ \ddot{u}_2 \end{Bmatrix} + \begin{bmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix}\begin{Bmatrix} \dot{u}_1 \\ \dot{u}_2 \end{Bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}\begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = \begin{Bmatrix} -m_1\ddot{y}_1 \\ -m_2\ddot{y}_2 \end{Bmatrix}$$

The natural frequency of system is determined as follows. Homogeneous form of equation for natural frequency determination is given by Eq 6.

$$M\ddot{u} + Ku = F \quad \text{(Eq 6)}$$

Its solution is expressed in terms of generalized coordinate vector 'q' as follows.

$$\text{Displacement,} \quad u = qe^{(i\omega t)} \quad \text{(Eq 7)}$$

$$\text{Velocity,} \quad \dot{u} = i\omega qe^{(i\omega t)} \quad \text{(Eq 8)}$$

$$\text{Acceleration,} \quad \ddot{u} = -\omega^2 qe^{(i\omega t)} \quad \text{(Eq 9)}$$

Substituting Eqs 7, 8 and 9 in Eq 6.

$$(-\omega^2 M + K)qe^{(i\omega t)} = 0$$

The eigenvalues are determined as follows.

$$\det\{K - \omega^2 M\} = 0$$

$$\det\left\{\begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} - \omega^2\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\right\} = 0$$

$$-\omega^4 m_1m_2 + \omega^2[-m_2(k_1 + k_2) - m_1k_2] + k_1k_2 = 0 \quad \text{(Eq 10)}$$

$$\omega_1^2 = \frac{-b - \sqrt{b^2 - 4ac}}{2a} \quad \text{and} \quad \omega_2^2 = \frac{-b + \sqrt{b^2 - 4ac}}{2a} \quad \text{(Eq 11)}$$

where

$a = m_1m_2$

$b = [-m_2(k_1 + k_2) - m_1k_2]$

$c = k_1k_2$

The natural frequencies of the suspensions system are obtained by Eq 11 as,

$$\omega_{n1} = 7.92 \text{ rad/s} = 1.26 \text{ Hz}$$

$$\omega_{n2} = 24.72 \text{ rad/s} = 3.934 \text{ Hz}$$

The natural frequency for 1/4th model of rail vehicle as shown in Fig. 8 is also determined by finite element analysis using ANSYS. The modal analysis is carried out with COMBIN14 element for stiffness and damping coefficient of suspension system and MASS21 element for mass of body. The ANSYS model for 2-DOF spring–mass–damper showing natural frequencies is shown in Fig. 9. The modal analysis using COMBINE14 and MASS21 element gives the natural frequencies in first mode as 1.261 Hz and in second mode as 3.935 Hz which are in close agreement with analytically calculated natural frequencies.

From Fig. 9, the first natural frequency of the system is observed to be about 1.261 Hz which agrees well with the natural frequency determined analytically. The comparison of natural frequencies determined using analytical and FEA is given in Table 5, and they are found to be in close agreement.

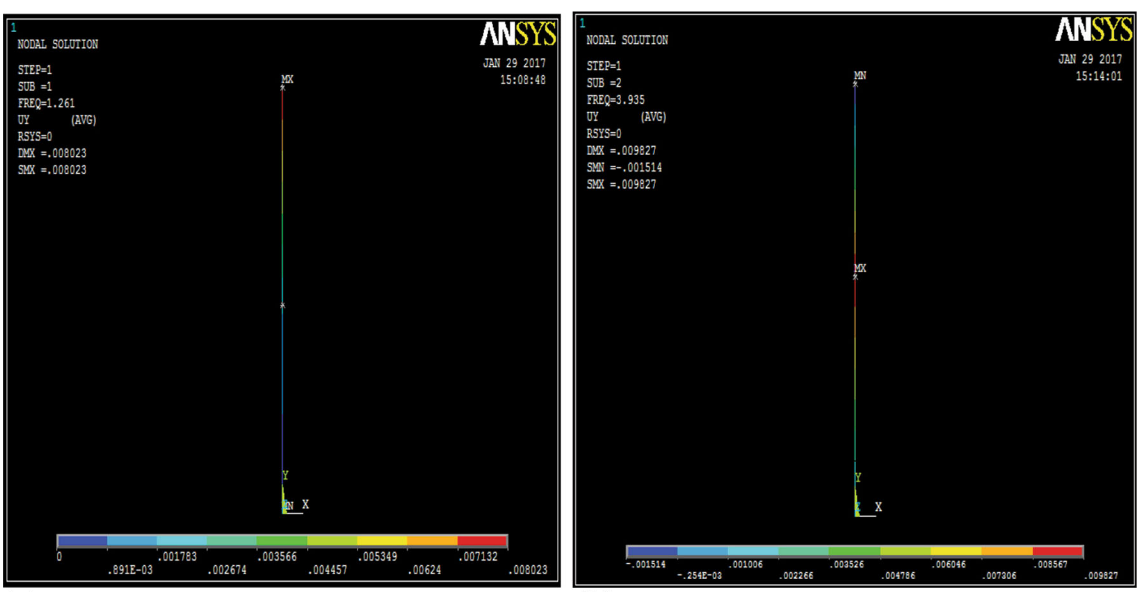
The natural frequencies for suspension determined earlier are 1.26 and 3.94 Hz, i.e., approximately 1.3 and 3.6 Hz, while the excitation frequency is found from the experimentation as 1.3–1.6 Hz. This clearly indicates that the natural frequency of the system is very close with the excitation frequency which is in the band of 1.3–1.6 Hz. The polished damper marks show the amplitude of 3–4 mm corresponding to the damper displacement band of 6–8 mm. Thus, it can be said that the actual amplitude of system is 3–4 mm. The amplitude ratio is given by Eq 12.

$$\frac{X}{Y} = \frac{\sqrt{1 + (2\xi\omega/\omega_n)^2}}{\sqrt{(1 - \omega^2/\omega_n^2)^2 + (2\xi\omega/\omega_n)^2}} \quad \text{(Eq 12)}$$

By considering ω/ω_n ratio approximately equal to 1 and the damping factor of 0.2306, the amplitude ratio is determined as 2.387. From this the amplitude of excitation is determined as 1.3–1.7 mm which indicates that the amplitude of vibration of the primary suspension system is more than the amplitude of excitation. This clearly indicates the rise in the amplitude of primary suspension due to acute closeness of excitation frequency with systems natural frequency. The amplitude of excitation is in the range of 1.3–1.7 mm, and it may further increase due to irregularities over the rail surface, misalignment between two connecting rails, due to improper leveling, etc., as shown in Fig. 10. It may also occur due to ovality, surface defect, flat surface, pitting marks, etc., on rail wheel.

Experimental Investigation of Vibration Response of WAG-9 Locomotive Suspension

A rail vehicle is the dynamic multibody system which has the repeated oscillation of its suspension system due to unevenness of track. The dynamic behavior of a railroad vehicle also depends on the vehicle mass and the mechanical systems, such as springs and dampers, which interact with the wheels, the vehicle body and bogies.



(a) **(b)**

Fig. 9 Natural frequency of 1/4th model of rail vehicle using ANSYS. (**a**) First mode (**b**) Second mode

FFT Vibration analyzer of make SVANTEK, model SVAN958 is used to determine vibration response in vertical, lateral and in longitudinal direction in the form of acceleration with respect to time. The analyzer is used to find peak value which determines the dynamic behavior of rail vehicle on running track. The readings are taken by attaching an accelerometer on the frame of WAG-9 rail vehicle in running condition at an average speed of about 80 km/h.

During acquisition of a peak value of acceleration, RMS vibration and PSD resonance in all directions have been recorded for different tracks which are observed to be maximum in vertical and lateral direction during running condition at an average speed of 80 km/h. The responses have been recorded for rail vehicle moving on straight tracks, curve tracks and for various tracks. The response of rail vehicle suspension in the form of acceleration with respect to time and frequency has been recorded, and its vibration response and power spectrum density for different tracks are captured in vertical, lateral and longitudinal direction. The peak values of acceleration are 4.81 m/s^2 in vertical direction, 4.20 m/s^2 in lateral direction and 0.72 m/s^2 in longitudinal direction. Also the RMS vibrations are 0.79 m/s^2 in vertical direction, 0.62 m/s^2 in lateral direction and 0.15 m/s^2 in longitudinal direction. But as discussed earlier, the analysis has more interest toward the vertical vibration of suspension system and hence vertical responses are presented in Fig. 11. Also the power spectrum density (PSD) value is maximum for the frequencies of about of 1.3 and 12 Hz in vertical direction.

The vibration response indicates the peak at frequencies of 1.3 and 12 Hz for various sets of reading, but the peak frequency zone from 1.3 to 1.6 Hz is observed to be common in all the sets of readings. As the suspension undergoes the forced vibration due to base excitation, it is considered that the excitation frequency lies between 1.3 and 1.6 Hz for most of the selected tracks. The amplitude of excitation determined experimentally is from 1.3 to 1.7 mm. The railway authorities regularly carry out the track recording, analysis and monitoring using track recording cars having appropriate sensors and measuring devices. As per their record, the rail track unevenness values are in the range of 1.1–1.8 mm normally. The amplitude of excitation of 1.3–1.7 mm closely matches with the actual track unevenness readings ranging from 1.1 to 1.8 mm. This also validates the experimentation and dynamic model considered for the analysis.

Table 5 Comparison of natural frequencies determined using analytical and FEA of suspension system

Natural frequency	Analytical natural frequency	Natural frequency using FEA	Mean natural frequency
First	1.26	1.261	1.26
Second	3.934	3.935	3.94

Experimental Determination of Vibration Response of Rail Track

It is also felt necessary to determine the vibration response of the rail track experimentally when the train passes over it. A railway track is an infinitely long uniform beam on an elastic foundation, and it is subjected to a distributed transverse load traveling at a certain speed along the beam. The vibration response of a railroad track under the moving weight of the rail vehicle is determined using FFT analyzer, and different sets of reading for various random tracks are obtained. The vibration response provides the track frequencies for its vertical excitation due to fast moving rail vehicle. A vibration and power spectrum response of railway track has been measured by attaching

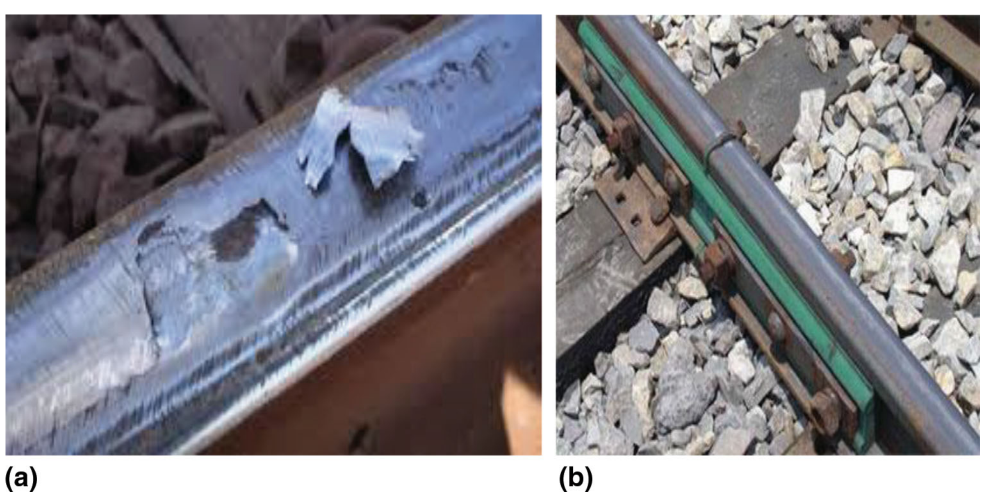


Fig. 10 (**a**) Track defect, (**b**) track joint

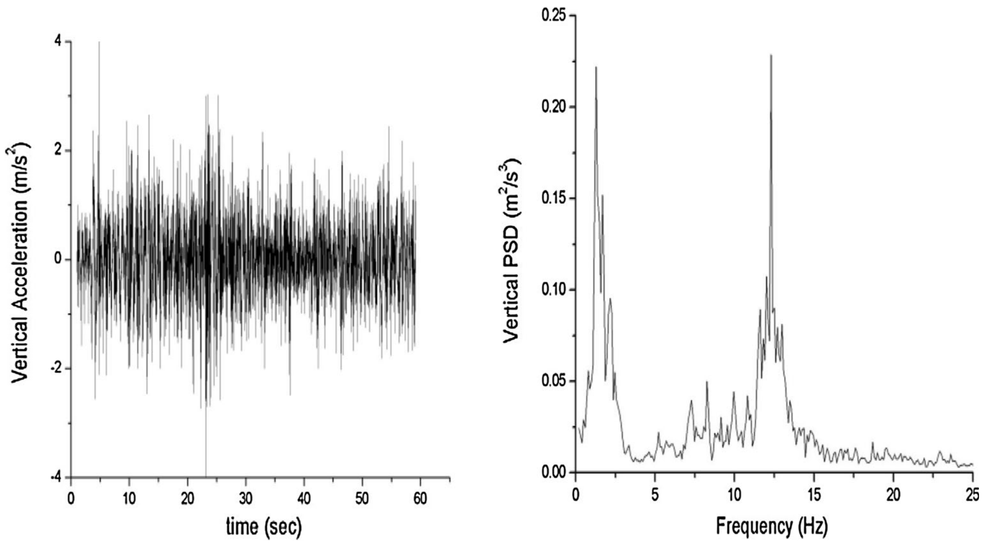


Fig. 11 Vibration and power spectrum density (PSD) of rail vehicle suspension in vertical direction during dynamic condition

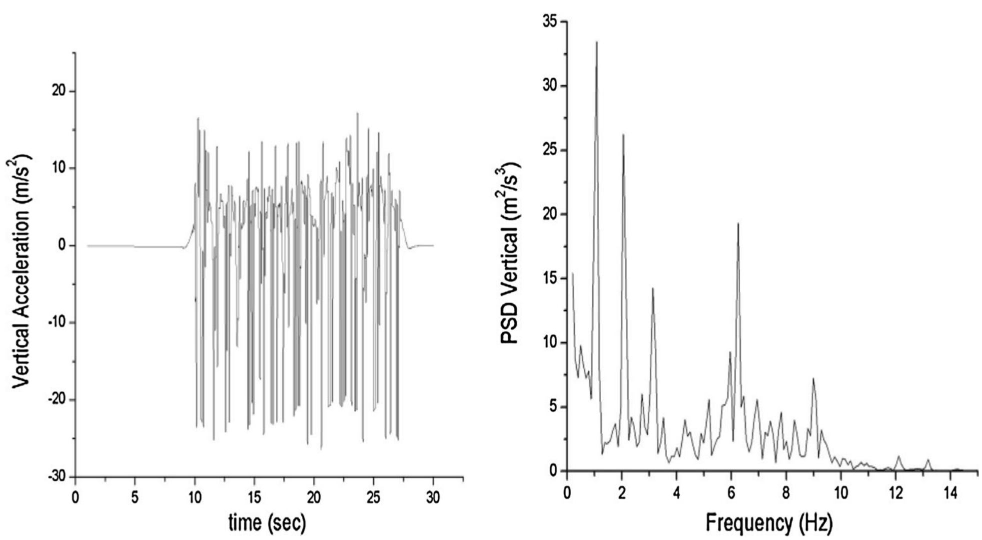


Fig. 12 Vibration and power spectrum density of rail track in vertical direction when railway vehicle passes over the track

accelerometer pick up on web of track. The vibration spectrum is obtained in vertical direction with maximum power spectrum density for frequencies in the band of 1–7 Hz as shown in Fig. 12.

From Fig. 12, it is seen that the frequency of vibration of rails is in the range of 1–7 Hz. This band of frequencies also encompasses the natural frequencies of the suspension system, and this finding justifies the investigation presented in "Model for Vertical Vibrations of Rail Vehicle Suspension" section.

As seen from the earlier investigations that due to variation in amplitude in a cyclic manner, the spring is subjected to variable force. Hence, it is felt further necessary to investigate the failure of middle axle inner suspension spring under variable load by considering its fatigue analysis.

Fatigue Analysis of Suspension Spring

As evident from earlier section, the middle axle primary inner suspension spring is subjected to variable loads and hence the fatigue analysis approach is used to investigate the failure of the spring. Finite element method is used to carry out the fatigue analysis. The fatigue life of a component can be expressed as the sum of two segments of life: (a) the number of loading cycles required to initiate a crack and (b) the number of cycles it takes that crack to propagate to failure [11]. A computational model for fatigue analysis of suspension spring has been presented. A maximum and minimum load is often used for simulation of the cyclic loading in fatigue analyses on helical suspension spring [12].

This section discusses the finite element analysis of middle axle primary inner suspension spring of WAG-9 rail

Fig. 13 Photographs of failed primary inner suspension spring and polished rubbing marks over damper

Table 6 Fatigue life and factor of safety for inner suspension spring

Particulars	Symbol and unit	Inner suspension spring For maximum 6 mm variable displacement	For maximum 7 mm variable displacement	For maximum 8 mm variable displacement
Minimum deflection	δ_{min} (mm)	64.6	64.6	64.6
Minimum load	F_{min} (N)	9351.46	9351.46	9351.46
Maximum deflection	δ_{max} (mm)	70.6	71.6	72.6
Maximum load	F_{max} (N)	10,219.62	10,364.37	10,509.13
Mean load	F_m (N)	9784.54	9857.92	9930.29
Alternating load	F_a (N)	434.08	506.46	578.83
Wahl's factor	K	1.29	1.29	1.29
Mean shear stress	τ_m (N/mm^2)	626.46	630.77	634.72
Variable shear stress	τ_a (N/mm^2)	27.79	32.42	37.06
Stress ratio	R	0.92	0.90	0.89
Equivalent shear stress	τ_{eq} (N/mm^2)	688.07	702.66	717.89
Factor of safety	F.S.	1.17	1.13	1.09
Fatigue life	Cycles	1.89×10^4	1.62×10^4	1.40×10^4

vehicle using a FE tool ANSYS to find its fatigue life. Analytically, the forces acting on suspension springs and shear stresses induced are determined for static condition and continued for fatigue analysis for displacement amplitude variation of maximum 6 mm to 8 mm as per the amplitude of vibration of primary suspension and also as per the observations of polished rubbing marks over dampers. Fatigue analysis has been carried out in ANSYS considering the load ratio, ultimate and endurance shear limit for chrome vanadium (CrV) material. A failed middle axle inner suspension spring and polished rubbing marks over damper are shown in Fig. 13.

It is observed that the spring undergoes variable displacement as observed from rubbing marks over the end axle dampers as shown in Fig. 13 and also as per dynamic analysis presented. Because of this, the variable force acts over the primary inner suspension spring. But the band of polished surface on the damper indicates that there is the displacement of the spring in the range of 6–8 mm which may be the cause of fatigue failure of spring. The maximum and minimum load corresponding to the variable deflection of 6 mm to 8 mm is given Table 6.

A stress ratio is given as input for fatigue analysis in ANSYS and parameters considered in fatigue analysis. For Stress–Life (S–N) curve with low-cycle and high-cycle fatigue life, S–N curve has been plotted for ultimate shear strength of 1152.4 N/mm^2 and endurance shear strength of 395.6 N/mm^2. Considering fatigue strength factor as unity, equivalent alternating shear stress is determined and it is given in Table 6.

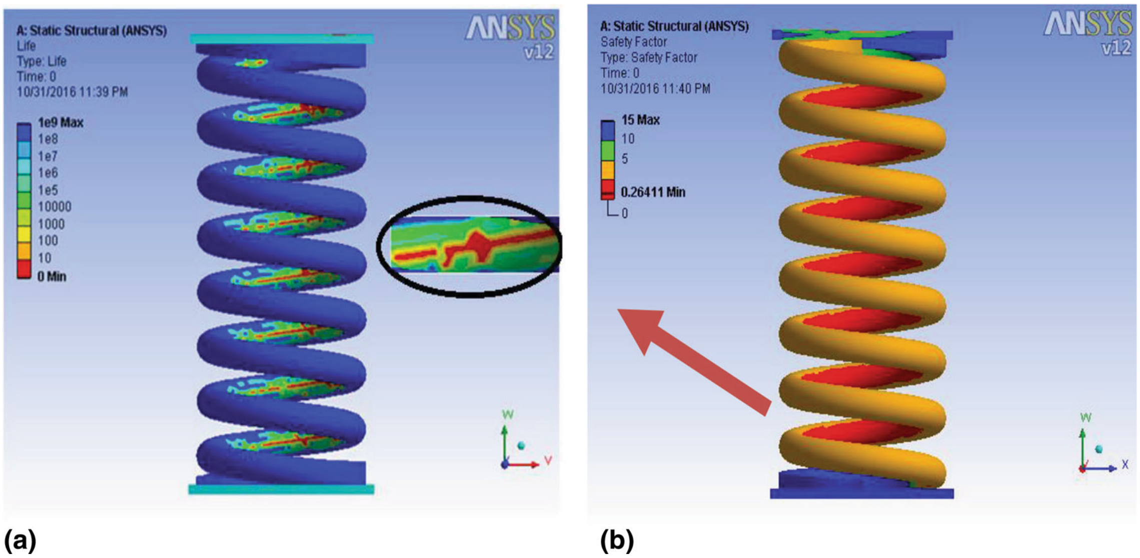


Fig. 14 FE analysis results of inner suspension spring for its (a) fatigue life and (b) safety factor

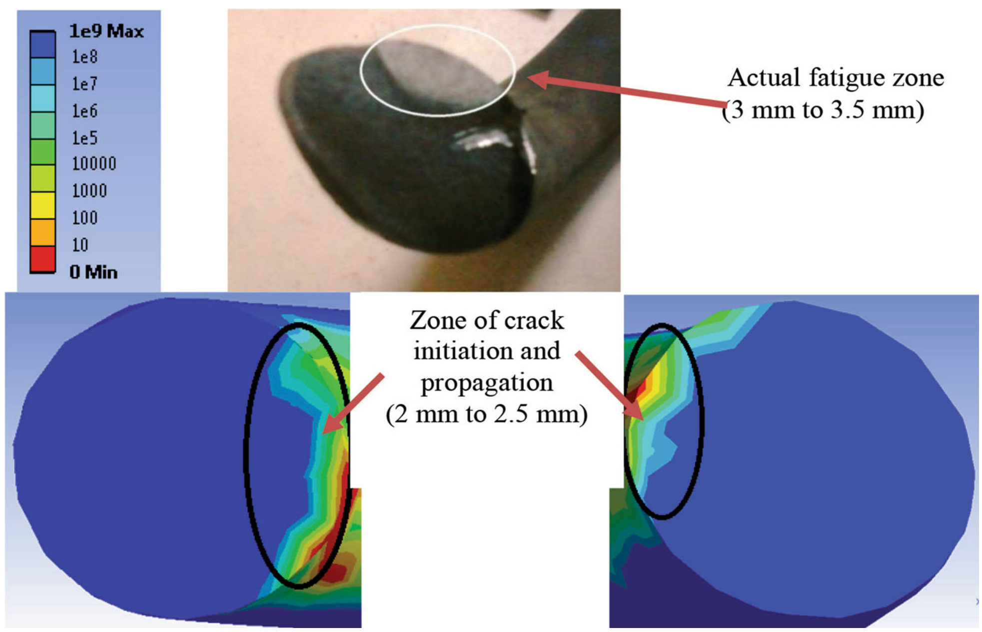


Fig. 15 FE results for fatigue zone of middle axle inner suspension spring showing crack initiation and propagation of 2–2.5 mm

From Fig. 14 and from Table 6, it is observed that the spring has maximum finite life of 1.89×10^4 cycles. While examining the failed specimen as shown in Fig. 13, it has been also observed that the cross section of failed spring resembles to that of fatigue failure.

Figure 15 shows the fatigue life for various regions of the cross section. The fatigue life varies from 10 cycles to 10^5 cycles for the zone nearer to the inner side of the coil. This region has finite life with minimum life at the inner side of the coil. The life progressively increases for the cross section slightly away from the inside diameter, but still this region is having the finite life, while the rest of the cross section has the life more than 10^6 cycles and which corresponds to infinite life. Thus, during the service span of the spring, the crack initiates at the inner side of the spring after very few cycles of operation and it propagates through the cross section.

Up to 10^5 cycles the length of crack growth is estimated to be maximum 2–2.5 mm. After observation of cross section of failed spring, it is seen that the polished surface at the failure zone has the band of approximately 3–3.5 mm which is more than FE fatigue analysis results. The reasoning for this difference is given as follows.

When the crack initiates at the inner surface, the stress magnitude on rest of the cross section increases due to effect of stress concentration and decreases in crosssectional area which further increases the induced stresses and hence decreases the fatigue life. So, as the crack proceeds there is possibility that the actual crack growth will be more as compared to theoretical FE fatigue analysis results due to progressive increase in stress magnitude. This analysis reveals that the spring fails due to fatigue failure as it is having finite life.

Conclusions

From the detail static, dynamic and fatigue analysis of primary suspension spring, the conclusions of research work is as follows.

- The experimental spectroscopy analysis ensured the chemical composition of spring material as per recommended ranges and experimental spring stiffness measurement using spring testing machine for all primary suspension springs yielded the recommended stiffnesses, and hence the cause of failure is not attributed to material composition and design specifications of suspension spring.
- The static stress analysis carried out for the three cases, i.e., straight track, curved track and tractive effort has resulted the induced stress values well below the yield stress values, and hence it could not reveal the cause of failure.
- The dynamic analysis performed on dynamic model of suspension system revealed the natural frequency, excitation frequency and amplitudes. Due to closeness of natural frequency with excitation frequency, the amplitude of vibration is observed to be more than excitation amplitude which resulted in variable displacement and spring force. This has been also verified from the polished damper marks observed on the damper.
- The fatigue analysis reveals that the middle axle inner suspension spring has a finite life of 1.89×10^4 cycles which clearly indicates that the spring fails because of fatigue failure with crack initiation at inside diameter. This has been confirmed from the observation of cross section of the actual failed spring.

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