

## Study For Frequency Response Of Railway Vehicle Using Modal Analysis

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**Abstract:** Suspension System is an important element of rail road vehicle. It absorbs the shocks and vibrations during tracking, curving and also protect the axle movement. The helical compression type of spring is used to allow axial deformation. In this paper the investigation starts with the stress analysis with respect to the mechanical properties of material by analytical methods of Primary Suspension Springs at various tracking conditions. Further it is continued with the Modal Analysis of the Locomotive for the Vertical excitation.

**Keywords:** Helical Spring, Primary Suspension System, Stress Analysis, Finite Element Analysis, Modal Analysis, Vertical Excitation

### I. INTRODUCTION

The railway vehicle running along a track is one of the most complex dynamical systems in engineering. It has many degrees of freedom and the study of rail vehicle dynamics is a difficult task. On tangent track, at slower speeds of operation, rock and roll problems occur. At higher speeds, a vehicle may hunt and bounce severely. While negotiating on a curved track, wheel tends to climb the rail, excessive lateral forces may develop and rail vehicle may roll over.

Springs are elastic bodies that can be twisted, pulled, or stretched by some force. They can return to their original shape when the force is released. In other words it is also termed as a resilient member.

Based on the shape behavior obtained by some applied force, springs are classified into the following ways:

- Helical springs
- Leaf springs [1]

The gap between the successive coils is larger.

- It is made of round wire and wrapped in cylindrical shape with a constant pitch between the coils.
- By applying the load the spring contracts in action.
- There are mainly four forms of compression springs as shown in Figure1 [1].

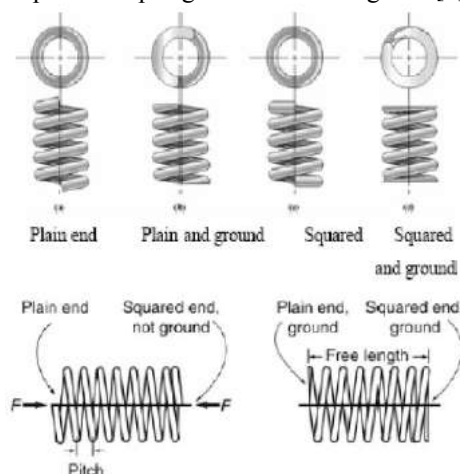


Fig- 1. End connections of helical compression spring [1]

In Indian Railway WAP-4 is one of the commonly used electric locomotive. It has a capability of hauling 26 coaches at a smooth speed of 140 km/h.

This locomotive was developed to avoid the use of some old Locomotives such as WAP-1 which were inadequate to haul 23-24 coaches for high speed trains. It came into existence in 1994 and had a similar body shell like WAP-1 class, with a Hitachi traction motors developing 5000 hp (approx). driving 6 traction motors arranged in Co-Co fashion. This locomotive became a big success. There are over 800 units in service and more being produced. Adapting with new technology the Microprocessor Controlled diagnostics, Static Converter units and roof mounted Dynamic (Rheostat) Brakes have been installed [2].

**Specification Of Helical Compression Spring**

The suspension spring arrangement used in WAP-4 Locomotive consists of two types of springs i.e., primary and secondary suspension spring which are mainly used to absorb shock and vibration coming on the wheels of locomotive. The technical specifications of both springs are given below in table I.

**Dimensional Specifications**

**Table-1**

<u>SNo</u>	<u>PARTICULAR</u>	<u>PRIMARY</u>	<u>SECONDARY</u>
1.	TOTAL REQD	24	16
2.	FREE HEIGHT (mm)	385	478
3.	WKG LOAD (Kg)	3695	4110
4.	NUMBER OF TURNS	8.5	9.5
5.	SPRING RATE (Kg/mm)	56.65	52.21
6.	WKG HEIGHT (mm)	320	399
7.	BAR DIA (mm)	33	38
8.	OUTER DIA (mm)	178	210

**Material Specifications**

**Table-2**

<u>DESCRIPTION</u>	<u>VALUE</u>
MATERIAL	Chrome Molybdenum Vanadium
YOUNGS MODULUS	210 GPa
MODULUS OF RIGIDITY	795 GPa
POISSONS RATIO	0.33

**STATIC STRESS ANALYSIS**

**Case-I: Load analysis for loco moving on straight path [3]**

The overall load of about 112.8 tons on loco is acted over the suspension springs on all wheel set.

- 1) Total weight of loco = 112.8 T
- 2) Unsprung mass per wheel set = 3.13 T
- 3) Total Unsprung mass = 18.8 T
- 4) Net load on all axles = 94 T
- 5) This net load is distributed over each of frontal and rear wheel sets through front and rear wheel frames.

Thus net load distributed on each frame = 47 T

Following assumptions are considered during analysis-

- i) The rails are smooth and there are no geometrical irregularities.
- ii) The loading is static.
- iii) Impact load during operation are neglected.
- iv) All other kind of load than mentioned in calculations is Neglected.

The maximum stress in the wire may be computed by superposition of the direct shear stress and the torsional Shear stress. Hence the modified equation to find shear stress using Wahl's factor is given by,[3]

$$\tau = K * \frac{8WD}{\pi d^3}$$

$$K = \frac{4c-1}{4c-4} + \frac{0.615}{c}$$

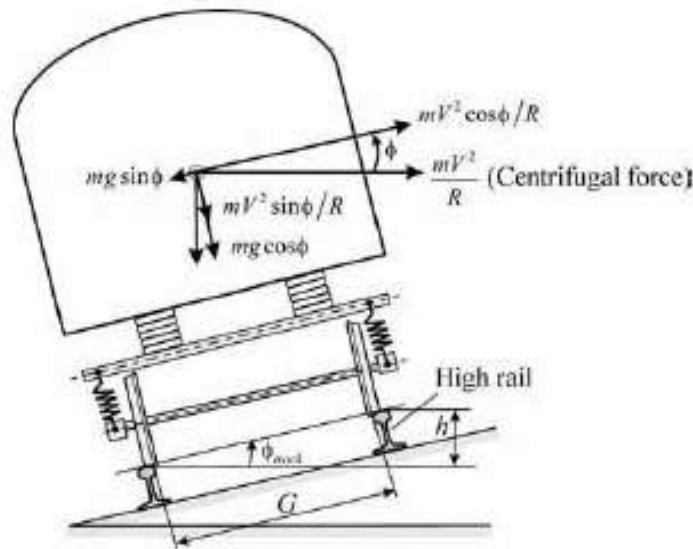
Hence the value of shear stress is found to be-

□ = **897MPa**

**Table 3- Analytical Results Primary Springs For The Loco At Straight Path**

Particulars	Unit	Value
Spring Index (C)	-	4.394
Wahl's Factor (K)	-	1.3609
Deflection ( $\delta$ )	mm	123.86
Load (W)	N	67974
Shear Stress ( $\tau$ )	MPa	950.463
Tensile Strength ( $\sigma_T$ )	Mpa	1200-1500

**CASE-II: Load analysis for loco moving on the curved path**



**Fig-2. Force analysis of Railway vehicle on curved track [4]**

While moving on the curved path the loco is subjected to additional lateral load because of centrifugal force which causes the lateral load on the spring with its bending moment maximum at the top end. The figure8 shows the load when the loco is moving on curved path. The lateral force must be in balance with the centrifugal force due to the track curvature. The centrifugal force tends to push the vehicle out of the curve toward the high rail. Therefore, if the curve has a certain curvature and super-elevation, there is a balance speed at which the component of the centrifugal force is equal to the lateral component of the gravity force [4].

Following are the detailsfor calculation of the load

- 1) The Avg. loco speed is 120 km/hr.
- 2) The radius of curvature is 292 m, which is the minimum recommended radius.
- 3) The banking angle is 60.
- 4) Diameter of wheel is 1016 mm
- 5) Angular velocity of wheel is 29.30 rad/s
- 6) Angular velocity of precession is 0.068 rad/s
- 7) Moment of Inertia of wheel is 2249.022 kg-m<sup>2</sup>
- 8) Gyroscopic couple is 4454.82 N-m
- 9) Reaction at gyroscopic couple (calculated)  
 $R_{GB} = 2658N$  ,  $R_{GA} = 2658 N$  “!
- 10) Reaction at outer rail  
 $R_B = 452.58 KN$

Reaction at inner rail

RA = 464.44 KN

Now, Stress correction factor

$$C = \frac{D}{d} = 5.394$$

Wahl's Factor, K = 1.2847

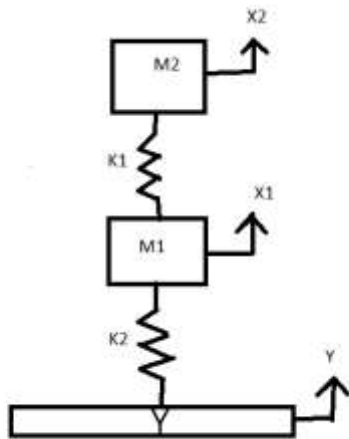
$$\sigma = 889.64 \text{ MPa}$$

**Table 4- Analytical Results Primary Springs For The Loco At Straight Path**

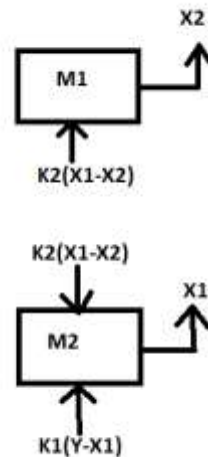
Particulars	Unit	Value
Spring Index (C)	-	4.394
Wahl's Factor (K <sub>w</sub> )	-	1.3609
Deflection (δ)	mm	123.86
Load (W)	N	67974
Shear Stress (τ)	MPa	950.463
Tensile Strength (σ <sub>T</sub> )	Mpa	1200-1500

**FREQUENCY RESPONSE FOR VERTICAL BASE EXCITATION**

For railway vehicle, considering a two-degree-of-freedom system subjected to vertical base excitation, as shown in fig.3. In the following arrangement, the Y element is Exciting element, whereas the masses m<sub>1</sub> & m<sub>2</sub> are the mass of Superstructure and the mass of Frame carrying the Primary & Secondary Suspension Springs respectively. The Free Body Diagram of the system is shown in fig.4 [5].



**Fig. 3- Spring Mass Arrangement**



**Fig. 4- Free Body Diagram**

The equation of motion for mass 2 is,

$$\sum F = m_2 \ddot{x}_2$$

$$m_2 \ddot{x}_2 = k_2(x_1 - x_2)$$

$$m_2 \ddot{x}_2 - k_2 x_1 + k_2 x_2 = 0$$

(1)

The equation of motion for mass 1 is,

$$\sum F = m_1 \ddot{x}_1$$

$$M_1 \ddot{x}_1 = -k_2(x_1 - x_2) + k_1(y - x_1)$$

(2) Assemble

the equations in matrix form,

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{bmatrix} \ddot{z}_1 \\ \ddot{z}_2 \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{bmatrix} z_1 \\ z_2 \end{bmatrix} = \begin{bmatrix} -m_1 \ddot{y} \\ -m_2 \ddot{y} \end{bmatrix}$$

Represent as

$$M \ddot{Z} + k z = F$$

To find natural frequency of any model,

Hence, Frequency equation becomes,

$$\omega^4 m_1 m_2 - \omega^2 \{ -m_2 (k_1 + k_2) - m_1 k_2 \} - k_1 k_2 = 0$$

(3)

Where,

$$M_1 = 37500 \text{ kg}; M_2 = 18750 \text{ kg}$$

$$K_1 = 6668838 \text{ N/m}; K_2 = 4097440.8 \text{ N/m}$$

By substituting the above values in eq. (3)

$$\omega_1 = 8.2325 \text{ rad/sec.} \quad \omega_2 = 23.9458 \text{ rad/sec.}$$

Therefore, Natural frequencies obtained,

$$f_1 = 1.3102 \text{ Hz}$$

$$f_2 = 3.811 \text{ Hz}$$

## II. MODAL ANALYSIS FOR NATURAL FREQUENCY OF VERTICAL SPRING MASS SUSPENSION SYSTEM

MASS21 is a point element having up to six degrees of freedom: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z axes. A different mass and rotary inertia may be assigned to each coordinate direction.

COMBIN14 has longitudinal or torsional capability in one, two, or three dimensional applications. The longitudinal spring-damper option is a uniaxial tension-compression element with up to three degrees of freedom at each node: translations in the nodal x, y, and z directions. No bending or torsion is considered. The torsional spring-damper option is a purely rotational element with three degrees of freedom at each node: rotations about the nodal x, y, and z axes. No bending or axial loads are considered.

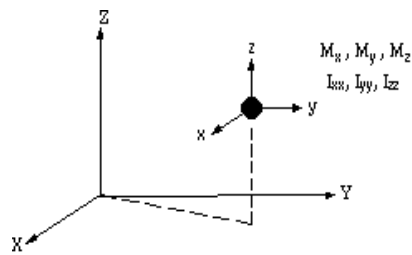
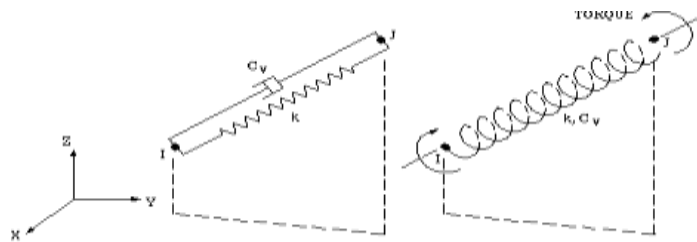


Fig-5. Mass 21 Element



Note—Two-dimensional elements must lie in the X-Y plane

Fig-6. Combin 14 Element

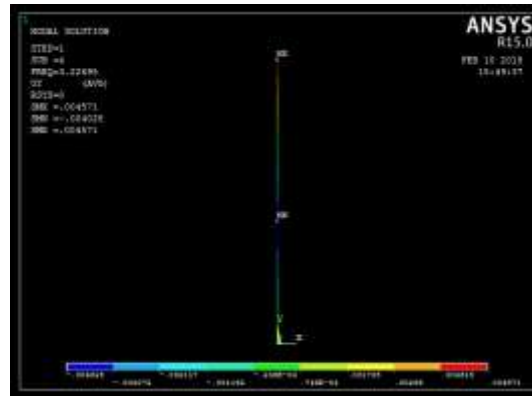
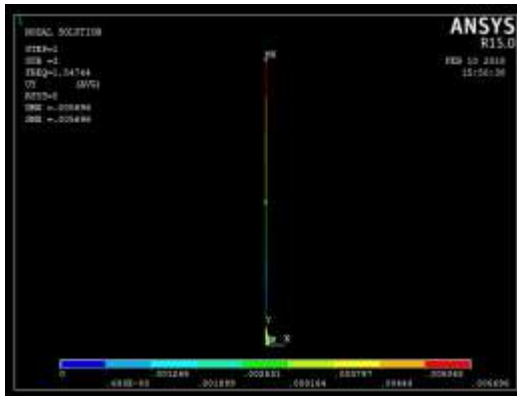


Fig-7. First and second mode for natural frequency of railway vehicle system using ANSYS

Table 5- Comparative frequency result table

PARTICULARS	UNIT	ANALYTICAL RESULTS	MODAL ANALYSIS RESULTS
F <sub>1</sub>	HZ	1.3102	1.5474
F <sub>2</sub>	HZ	3.811	3.2269

## III. CONCLUSIONS

In Static Stress Analysis, the shear stress acting on the Primary Suspension Spring is calculated in various tracking conditions. It has been observed that due to high stiffness of the spring the shear stress induced is less than the Tensile Strength of the spring, which means the spring is safe. But the suspension system of railway vehicle acquires so many excitation in running condition due track irregularities or it s flexibility.

Therefore on the other hand, the natural frequencies by considering two spring mass base excited system has been calculated by the analytical analysis and is compared with the natural frequencies found out by modal analysis using ANSYS. It is observed that the frequencies obtained are nearly same and is verified which has to be compared with excitation frequency of railway vehicle to check the condition of resonance. Hence there is scope to check the frequency response of railway vehicle in lateral and longitudinal direction also.

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