Failure Investigation of a Freight Locomotive Suspension Spring and Redesign of the spring for Durability and ride index

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Abstract

A good bogie design of freight locomotive ensures ride comfort and lateral stability during high speed cornering. The ride comfort and lateral stability which includes hunting and derailment are sensitive to dynamic load distribution. An improper load distribution and arrangement of primary suspension components may lead to failure of suspension springs and shock absorbers. In the present investigation, it was found that the existing primary suspensions with composite spring assembly could sustain loads in normal operating conditions and maintain the required ride index, however, during cornering and hunting speeds failure of outer spring of primary suspension was observed. In the present work, an attempt has been made to analyze in detail the reason for failure and a single non linear spring has been suggested to improve durability of the primary suspension and in the meantime the required ride index.

Failure of the composite spring assembly was analyzed by applying the forces obtained from dynamic analysis. The dynamic analysis was performed using ADAMS/Rail at four different velocities and three different track conditions. The critical loading condition was achieved at a hunting speed of 132km/h on a curved track. A single spring set was considered in ADAMS/View to perform stress analysis to know durability.

Numerical simulation showed that the stress level in the composite set which has both outer and inner spring was above allowable limit of 412 MPa where as a newly designed single non linear spring replacing the composite spring was found to have stress level of 160 MPa.The stress value obtained from numerical simulations in ADAMS was verified with analytical design calculations for the spring and the ride index was found to 1.78 which was 8% better than the earlier spring. It is concluded that the new spring design can enhance durability and ride index.

Key Words: Freight Locomotive, Hunting, Non Linear Coil Spring, Ride index, ADAMS/Rail

Nomenclature

Dm	Coil Mean Diameter, mm
G	Modulus of Rigidity, N/mm^2
GR	Rail Gauge, mm
Hf	Free Height, mm
hcg	CG Height, mm
K	Stiffness, N/mm
L	Total Length, mm
R	Cornering Radius, mm
V	Velocity, m/s

Abbreviations

ADAMS	Automatic Dynamic Analysis of Mechanical Systems
AAR	Association of American Railroads
BE	Braking Effort
CG	Centre of Gravity
DWM	Dynamic Weight Management
FFT	Fast Fourier Transform
SWM	Static Weight Management
TE	Tractive Effort

1. INTRODUCTION

Freight locomotives are widely used for bulk shipment over a long distance. Many locomotive manufacturing industries are introducing diesel electric locomotives which survive the purpose of fuel economy, cost and most important parameter ride comfort. The ride comfort standards defined by Association of American Railroads (AAR) for the freight locomotives are the focus of the industries and efforts have been put in terms to improve the same by modifying the existing bogie subsystem design. Instead of preferring expensive and time consuming prototype build and testing options, engineers analyze the performance in design phase. The modifications in subsystem components and improvements in dynamic performance have been achieved by Numerical methods and especially with dynamic analysis softwares.The North American freight locomotives are having 20 *T* per axle capacities and overall locomotive weight ranging from 120 *T* to 140 *T* with top speed of 132 *km/h* [1]. The bogie/truck subsystem design of these locomotives ensures the load distribution and ride comfort. In view to enhance these characteristics locomotives bogies are assembled with special weight management mechanisms. This mechanism enhances the load per axle during dynamic motions with effective tractive effort utilization. The bogie is consists of primary and secondary suspensions. The primary suspension connects wheel set to the bogie frame and its main function is to isolate the body from track irregularities, so as to give the car occupants an acceptably comfortable ride and to maintain vehicle track loads within acceptable limits. The secondary suspension function is to connect the bogie frame to the carbody and provide the required stiffness and freedom of movement of carbody in all coordinates [2].

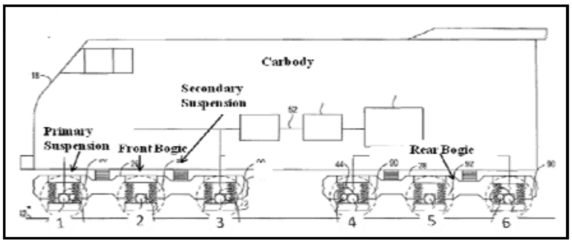

Fig. 16 Locomotive [3]

Figure 1 shows primary and secondary suspension connections with the carbody. These subsystems needs to comply to worthiness assessment which includes dynamic motions as Hunting, Carbody twist and roll, Pitch and bounce, Yaw and sway, Curving and spiral negotiation, Longitudinal train action.

As primary suspension requirements are focused on axle loading SWM and DWM arrangements fitted on it.

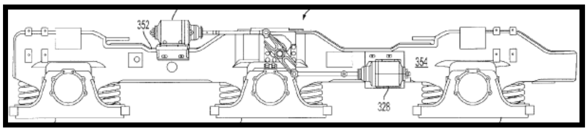

Fig. 17 SWM and DWM arrangements [3]

The weight management mechanism assist to transmit the TE and BE effectively to the locomotive carbody. During locomotive start axle loading occurs to the end axles and middle axles get unloaded because of this DWM mechanism. As soon as the locomotive goes to normal operating condition load get distributed equally. Fig 2 shows the mechanism fitted at the middle axle. The TE and BE are transmitted by end axles only when the mechanism is fitted. In view to sustain with such dynamic loadings, end axle's primary suspension of each bogie is arranged with composite spring assembly. It is shown in Fig 3. The composite spring assembly comprises of concentric outer and inner springs. Outer spring is supported at journal box and inner spring is supported and guided by holding rod. Holding rod is pivoted at the bottom by nut and bolt.

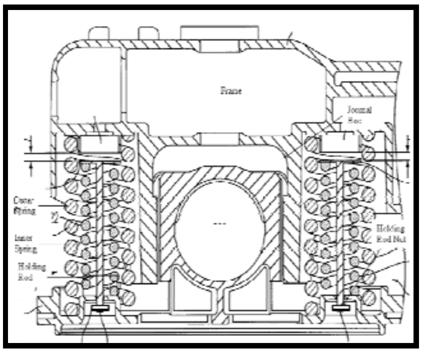

Fig. 18 Primary suspension at end axles [3]

These end axle suspension components required to behave with desired characteristics. The mounting and positioning components of the springs are failing in the field and consequently this is leading to the outer and inner spring failures and impacting on the dynamic behavior of the locomotive. In view to reduce the impact on the dynamic behavior of the locomotive and to assist the SWM and DWM arrangement, detailed investigation of existing spring assembly required to analyze. Also to eliminate this problem, it is desired to design a new coil spring which will show the similar stiffness characteristic as existing primary suspension springs characteristics.

In this investigation, the dynamic analysis at three different track condition and four different velocities performed on locomotive model in ADAMS/Rail. The cornering and hunting speeds observed the most critical loading condition for spring. The stress analysis on single spring set in ADAMS/View carried with critical condition force and stresses observed. The new non linear spring design concept suggested and initial design verified for stress at the cornering and hunting speed force. The prime important vehicle dynamic factor ride index behaviour compared between existing and new spring design.

Literature review performed to identify the dynamic rail analysis conditions and spring loadings. It is observed that numbers of researchers have studied the vehicle dynamic performance by varying suspension characteristics with the help of various simulation softwares.

Kossmann *et al* [4] focuses on the design and calculation of fast running shunting locomotives. The locomotive was a 2 axle electric shunting with a maximum operating speed of 40 km/h for the shunting of the coaches in the main stations. The weight of the vehicle ranges between 40 – 45 tons with an axle distance of 4 to 4.5 m. The maximum speed for transfer vehicle between stations and maintenance site was estimated at 100 km/h. The design stage involved developing a SIMPACK model which comprised of 9 bodies and standard force elements for primary springs, primary dampers, traction rods and bump stops.

Sharma *et al* [5] worked out the impact of communizing springs over the bogie models available in view to standardize bogies with next generation high speed coach. The existing and new spring secondary suspension design modeled keeping all other parameters same and dynamic analysis performed in ADAMS/Rail to study this impact. The analysis results indicated that the lateral ride index was improved whereas vertical ride index was deteriorated marginally. Karim H. A.*et al* [6] studied and analyzed railway ride comfort of car body at critical hunting velocity due to change in spring stiffness of vertical secondary suspension through mathematical model of a carbody which was modeled with 31 degrees of freedom. The study revealed that lateral, yaw and roll dynamic response of the railway carriage carbody was more sensitive to the hunting velocity than the vertical and pitch dynamic response.

Kumbhalkar *et al* [7] investigated the failure of the middle axle primary inner suspension spring of electric locomotive. In this study high rate failure loading conditions were observed on the field. The stresses at each loading conditions, on each spring, in the composite assembly were verified by analytical design and FE analysis. The study revealed that, the stresses on the springs were high in curve path conditions and above the allowable limit. They concluded the study with suggestion of providing damping to the spring in lateral direction to avoid spring failure.

Yu Cheng Su [8] studied mathematical dynamic models of helical spring for variable pitch angle, variable wire diameter, and variable coil diameter. These models were optimized by comparing the dynamic responses with physical dynamic experimental responses. FFT tool was utilized to evaluate the severity of resonances. The study concluded that the mathematical models derived to study the non linearity of spring can be optimized for better results by introducing more design variables. Association of American Railroad Chapter XI [12] presents the guideline for testing and analysis to ascertain the interchange service worthiness of freight cars. This chapter defines the structural, statics, impact and dynamic performance of freight cars to be examined. The dynamic analysis, testing instrumentation, testing requirements with acceptable criteria are defined as a standard.

It was concluded from the existing literature that a fundamental dynamic analysis studies have been performed on most of the passenger locomotives considering variables like track conditions and locomotive speeds through mathematical modeling and ADAMS\Rail. The existing literature focus on the study of impact of change in secondary suspension springs on the vertical ride index at hunting speed. The focus on lateral ride index is given in the literatures, intended for yaw and roll motions of the carbody. The literature reveals that hunting curve conditions is the critical condition for locomotive to operate. A spring failure and ride index improvement study has been carried at that condition.

2 ANALYTICAL DESIGN

2.1 Load Calculations

There are concentric outer and inner springs in primary suspension of the end axles. The technical specifications of each spring are shown in Table1. The analytical design methodology is used to calculate the loads at different conditions and stresses in each spring. In view to find the loads coming on each spring overall locomotive weight distribution studied, loads at static and dynamic conditions obtained.

- Total Locomotive Weight 1867893 N
- Carbody Weight 348690 N
- Bogie/Truck Weight 236567 N
- Sprung Mass on each Bogie 457477 N
- Unsprung Mass 212241 N

Locomotive dimensions and particulars used to calculate the loads are mentioned in the Table 2. The sprung weight per bogie is calculated and distributed as 85% and 15% and on outer and inner springs respectively. This is the static load acting on each spring.

Table. 1 Technical Specification of Existing Springs

Parameters	Units	Outer Spring	Inner Spring
Wire Diameter, d	*mm*	44	30
Coil Mean Diameter, D_m	*mm*	191	94
Free Height, H_f	*mm*	705.6	387.46
Working Height, H_w	*mm*	585	373
No. of Active Turns, N_a	-	9	10
Total No. of Turns, N_t	-	11	13
Modulus of Rigidity, G	*N/mm²*	78600	78600
Deflection, S	*mm*	121	15
Stiffness, K	*N/mm*	549	1021

Table. 2 Locomotive Specifications

Particulars	Units	
Total Length, L	*mm*	1534
Rail Gauge, G_R	*mm*	1435
CG Height, h_{cg}	*mm*	860
Cornering Radius, R	*mm*	100000
Velocity, V	*m/s*	36.66
Straight Path Acc, $a_{straight}$	m/s^2	0.05

Longitudinal direction CG point is considered at the centre of the overall length of the locomotive. Load transfer during straight path condition occurs with respect to this point. To negotiate curves on the railway track, only the middle axles are given the free play in a lateral direction perpendicular to the direction of motion. It should be noted that the axle do not turn about the loco but there is a sliding of the middle axle which helps in negotiating the curve.

The loads at each condition calculated and stress at cornering and hunting speed calculated. Table 3 shows the loads and stress on each spring.

2.2 Frequency Calculations

Frequency calculation performed to verify that the fundamental frequency of each coil spring is 15-20% above the design frequency. The Fixed-Fixed end frequencies are calculated and verified with modal analysis frequencies in view to assure that the correct spring geometry and material properties applied to the spring models used for stress analysis. The modal analysis is performed in ANSYS by using Block Lanczos Eigensolver. Table 4 shows the frequencies of outer and inner springs. Modal frequencies observed in line with the calculated frequencies

Table. 3 Loads and Stress in Existing Springs

Parameters	Units	Outer Spring	Inner Spring
Static Load	*N*	32405	8578
Straight Path Load	*N*	64768	14580
Cornering Load	*N*	66262	14976
Stress τ (Cornering & Hunting)	N/mm^2	543	197

.2.3 Ride Index Calculations

ride index is defined as the rate of change of acceleration, and human sensitivity to frequency of oscillation. It is calculated by using Sperling method and is given by

Table. 4 Frequencies of Existing Springs

	Units	Outer Spring	Inner Spring
Natural Frequency spring at Fixed Free End	*Hz*	23.54	61.44
Fundamental Natural Frequency	*Hz*	23.54	61.44
System Natural Frequency	*Hz*	1.43	4.12

$$Ride\ Index = K\left[\sum_{i=1}^{n} A_i^{3}\,\frac{F(f_i)}{f_i}\right]^{\frac{1}{10}}$$ [10]

Where,

A_i =Peak Accelerations in *'g's*

F(f) = A frequency sensitive physiological weighting function

K = Constant =7.1 (with 'a' measured in'g')

$F(f) = 0.325f^2$ ………………….$0 \le f < 5.9$

$= 400/f^2$ ……………………$5.9 \le f < 20$

$= 1$…… ……………………$20 \le f < \infty$

F = Acceleration Frequency in *Hz*

For Vertical Acceleration

$\text{ride index}_{max} \le 3.25$

ride index calculated by using above formula with new spring design is 1.81. The acceleration and frequency values are obtained from the results of new spring acceleration at centre block.

3. NUMERICAL DESIGN

Analytical approach to find out the spring loads assisted to understand the maximum loading occurs on springs during cornering and hunting speed. The numerical analyses also performed in ADAMS to simulated the actual conditions with locomotive model and obtain the maximum force. The locomotive model with carbody, front and rear bogie template created in ADAMS/View and assembly of these three subsystems templates imported to ADAMS/Rail to perform dynamic analysis at various conditions.

ADAMS/Rail V12, used to perform the dynamic analysis and ADAMS/View R12, used to carry out stress analysis on single composite spring set. Figure 4 shows the locomotive model built in ADAMS/View and the same is imported to ADAMS/ Rail.

The dynamic analysis carried at various conditions studied from literatures and test conditions data available in the literature. The straight, curve and irregular track conditions at 30, 60 ,80 and 132 *km/h* considered to perform the dynamic analysis. Track of 1000 *m* used for straight and irregular conditions and 4000 *m* for curve the conditions. Curvature radius of 100 *m* and irregularity of 1.04 *m* incorporated in the track property files.

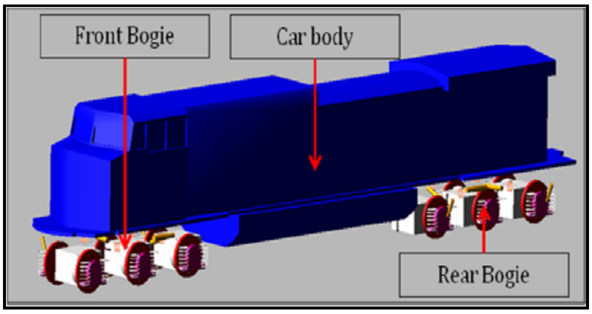

Fig. 19 ADAMS/Rail Locomotive Set Up

The maximum force obtained from above conditions dynamic analysis, is used to carry out the stress analysis on single spring set.

Figure 5 shows the single spring set with concentric outer and inner springs imported to ADAMS/View to perform stress analysis. The boundary conditions applied to the composite spring assembly have also been shown in Figure 5.

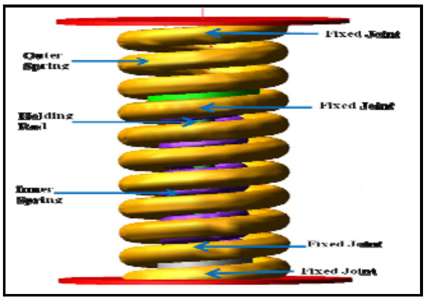

Fig. 20 Stress Analysis Set Up

4. SPRING REDESIGN

In view to avoid failure, different spring arrangement concept generated considering various factors like the packaging requirement, number of components, positioning of the components, manufacturability and suppliers availability. The trade off study assisted to predict to the non linear coil spring concept to be taken ahead as solution to avoid the spring failure. This concept eliminates the inner spring and holding rod arrangement. Hence, number of components gets reduced. The supplier's availability and manufacturability are also the important factors to proceed ahead for non linear spring design.

5. NEW SPRING DESIGN

The final concept of non linear coil spring studied for various types of non linear springs and trade off study performed to decide the type non linear coil spring detail design to be carried ahead. The trade off study assisted to finalize the variable pitch non linear coil spring details design. The analytical approach used to carry out the detail design of the variable pitch two stage non linear coil spring.

System inputs in terms of load Vs deflection characteristics plotted. The loads at each stage derived from the characteristics and observed that working load of 66262 *N* in stage one and solid load of 76710 *N* in second stage acts on the spring. These loads considered as the input for the detail design of non linear coil spring.

Table. 5 Non Linear Input Parameters

Parameters	Units	Stage1	Stage 2
Wire Diameter, d	*mm*	44	44
Coil Mean Diameter, D_m	*mm*	183	183
Loads, P1 & P2	*N*	66262	76710
Pitch, p	*mm*	60	47
Deflection, S	*mm*	53	43
Spring index, C	-	4	
Wahl Factor, K_w	-	1.39	
Deflection, S1 & S2	*mm*	53	43
Modulus of Rigidity, G	N/mm^2	78600	

Based on the packaging dimensions availability geometry input parameters and initial design parameters are assumed as listed in Table 5.Detailed calculations for stiffness and no. of active turns performed and stress at each stage obtained for the spring.

$$Stiffness, K = \frac{F}{S}\text{[10]} \qquad \text{.........(1)}$$

$$No.\, of\ Active\ Turns\ N_a = \frac{Gd^4}{8D_m{}^3K}\text{[11]} \qquad \text{.......(2)}$$

Free Height,

$$H_f = (p_1 \times N_{a1}) + (p_2 \times N_{a2}) + n_s + N_{inactive} + d \quad \text{[11]} \qquad \text{.........(3)}$$

$$Working\ Height, H_w = H_f - S1\text{[11]} \qquad \text{.........(4)}$$

$$Solid\ Height, H_s = N_t \times d\text{[11]} \qquad \text{..........(5)}$$

Equivalent Stiffness,

$$K_e = \frac{K1 \times K2}{K1 + K2}\text{[11]} \qquad \text{..........(6)}$$

$$Stress\ \tau = \frac{8PD_m}{\pi d^4}K_w\text{[11]} \qquad \text{.........(7)}$$

The above equations used and output parameters obtained as mentioned in Table 6.

Table. 6 Non Linear Coil Spring Output Parameters

Parameters	Units	Stage1	Stage 2
No.of Active Turns, N_a	-	5	4
Total No.of Turns, N_t	-	11	13
Free Height, H_f	*mm*	625	
Working Height, H_w	*mm*	594	
Solid Height, H_s	*mm*	499	
Stiffness, K	*N/mm*	549	1021
Equivalent stiffness, K_e	*N/mm*	742	
Stress, τ	N/mm^2	351	406

The working height of the spring is calculated by adding the total deflection form the solid height of the spring. Free height to working height deflection is nothing but the deflection due to working load acting on the spring. Iterations performed to achieve working height by varying pitch at each stage. Hence, the variable pitch non linear coil spring design taken ahead for the detail analysis.

The geometry of the new variable pitch non linear coil spring modeled with input and output parameters in CAD.

The modal analysis is performed in ANSYS with the same methodology used for existing springs to obtain fundamental frequencies. The fundamental frequency of 24 *Hz* observed both in analytical and numerical approach. It is observed 15 to 20 % above the range of system natural frequency. The stress analysis on this model carried in ADAMS/View and ride index verified by performing dynamic analysis in ADAMS/Rail.

6. RESULTS AND DISCUSSION

6.1 Dynamic Analysis

The dynamic analysis performed with existing spring design at various conditions mentioned in section 3 assisted to obtain the maximum force at each track condition and at maximum speed of 36.66 *m/s* (132 *km/h*). Table 7 shows the forces on outer and inner spring at hunting speed of 132 *km/h* at different track conditions.

Table. 7. Existing springs forces

	Units	Straight Track	Curve Track	Irregular Track
Outer Spring	*N*	52970	64460	62960
Inner Spring	*N*	12970	15370	12950

From the analyzed conditions the forces on each spring at hunting and curve tack condition obtained. Figure 6 and Figure 7 shows the force coming on outer and inner springs during hunting and curve track condition respectively.

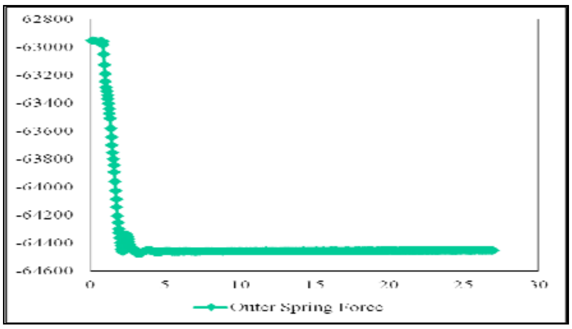

Fig. 21 Outer spring force

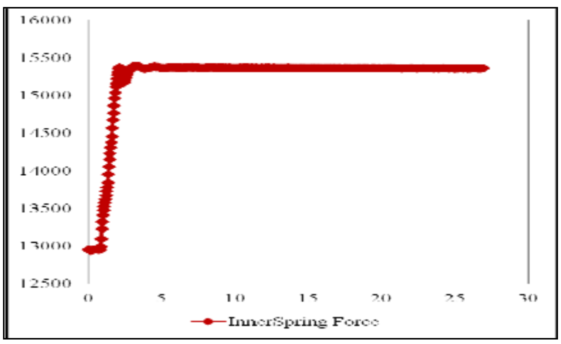

Fig. 22 Inner spring force

6.2 Stress Analysis

The composite spring assembly fitted at first axle at the front of journal box taken to ADAMS/View for stress analysis under the above obtained force.The stress value of 742 N/mm^2 (*MPa*) observed on outer spring when the analysis performed for time period of 30, with the force obtained at hunting and curve track condition. Figure 8 shows stress coming on the outer spring.

This stress values is above the allowable limit of 412 N/mm^2 (*MPa*). It can be predicted that the stress on outer spring was on higher side.

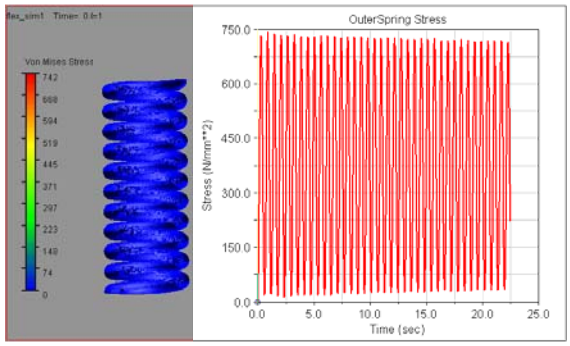

Fig. 23 Outer spring stress

Figure 9 shows the stress on inner spring analyzed under the similar condition as outer spring with the maximum force. The stress value of 192 N/mm^2 (*MPa*) observed which is below the allowable limit of 412 N/mm^2 (*MPa*).

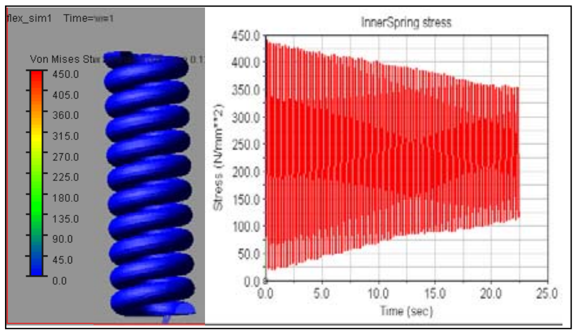

Fig. 24 Inner spring force

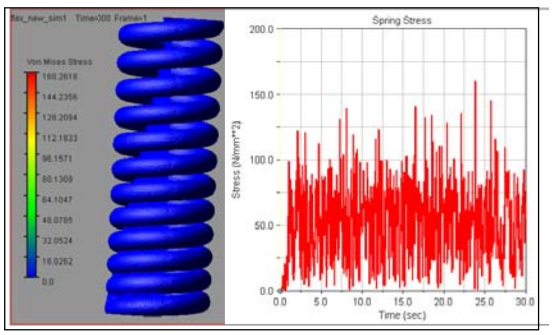

Fig. 25 Non linear spring stress

Newly designed non linear coil spring also analyzed under the same condition as existing outer spring analyzed for stress. Figure 10 shows the stress coming on new non linear spring.

The stress value of 162 N/mm^2 (*MPa*) observed on new spring which is well below the allowable stress limit of 412 N/mm^2 (*MPa*).

6.3 Ride Index Comparison

Ride index is the rate of change of acceleration and generally accelerations are measured at operators' seat.

The newly designed spring stiffness characteristics updated in the suspension element property file and dynamic analysis performed at hunting curve track condition. In this study, ride index is measured at the centre block of the first bogie. Though, the standard method to measure the ride index is below operators cab seat, to observe the impact of new spring design, ride index measured at centre block of the front bogie. Centre pin is the location where carbody is connected to the bogie and load transfer occurs through the same point. Hence, initial observational point for ride index considered at centre block.

This new spring design is focused on the vertical stiffness. The impact of this stiffness on the ride index in vertical direction only studied in the present investigation. The variation in lateral direction ride index is not focused since it has involved the yaw and roll motion into the consideration.

In the post processing Sperling method used to find out the Z direction ride index.

Figure 11 shows the 1.92 ride index for existing spring characteristics and Figure 12 show 1.78 ride index in Z direction for newly designed non linear spring characteristics.

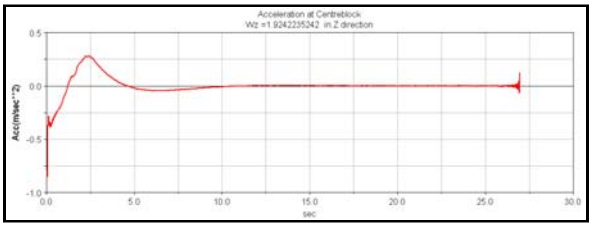

Fig. 26 Ride index with existing design

Ride index with newly design spring is not deteriorating from the existing ride index , however, improvement of 8% observed over the existing ride index.

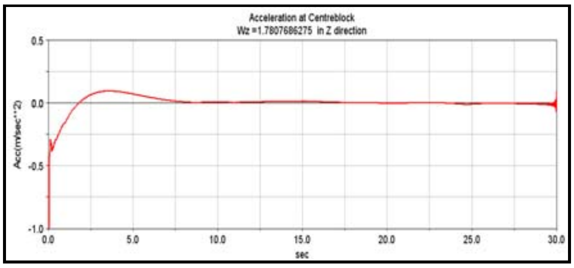

Fig. 27 Ride index with new spring design

5. CONCLUSIONS

Numerical simulation showed that the stress level in the composite set which has both outer and inner spring was above allowable limit of 412 MPa where as a newly designed single non linear spring replacing the composite spring was found to have stress level of 160 MPa. Existing spring stress values observed above the allowable limit during hunting curve track condition. The dynamic analysis set the design limits for new coil spring design and the newly designed non linear coil spring stiffness is on higher side but stress observed below allowable limit and also non linear coil spring design showing 8% improvement from the existing vertical ride index. It is concluded that the new spring design can enhance durability and ride index.

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