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Modeling and Analysis of Cabin Suspension and Rear Suspension of 6x4 Tipper Vehicles for Ride Comfort Using Creo1.0, Ansys 13.0 and Matlab 2013

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Abstract

The paper observes the effect of pitch and bounce for ride comfort in 4x2 tipper vehicles. Equation of motion is derived using Lagrange principle for half-car model. CREO is used to model the cabin suspension. The results are optimized and analyzed using MATLAB Simulink model for response of the cabin suspension system .Optimized data is finalized for pitch and bounce amplitude. Effect of damping coefficient on pitch and bounce is studied. The study is done for onlyunladen condition. In this paper we are going to discuss about the ride comfort and suspension system of 4x2 tipper vehicle's designing and analyzing. Here a complete cad model and assembly of boggie suspension (Hotchkiss Suspension) system for tipper vehicles usingCreo 1.0 and analysis of the suspension model (multi leaf spring model) using Ansys 13.0 Workbench and APDL Window whatever required.

Keywords: Bounce amplitude, Cabin Suspension,CREO model, Ride Comfort, Pitch amplitude

List of symbols used in this paper

Mvehicle	Mass of vehicle considered
mcabin	Mass of cabin
$k_{cabin\ front}$	Front spring stiffness of cabin
$k_{cabin\ rear}$	Rear spring stiffness of cabin
$K_{vehiclefront}$	Front spring stiffness of vehicle
$K_{vehiclerear}$	Rear spring stiffness of vehicle
$l_{CGto\ rear}$	Distance from rear axle to CG of the vehicle
$l_{cg\ to\ front}$	Distance from front axle to CG of the vehicle
CG	Centre of gravity

1. Introduction

Ride Comfort

Ride comfort can be defined as the transportation of an automobile passenger in so easy manner that the trip will be a liking and not a suffering [1]. Since spring-suspension constitutes the basis of ride comfort in passenger cars only here in this paper we will put it for off-road vehicles and trucks. An analysis made by the National Automobile Chamber of Commerce of replies received to a questionnaire it circulated among owners of automobile industries is presented in proof of the honest interest the motoring public has in the riding-quality of tipper vehicles and off road vehicles and also the variable factors that have an influence on spring-suspension are stated to be the type of spring used, its physical dimensions, the amounts of sprung and unsprung weight, frame construction, wheelbase dimension and the kind of material used.

Ride comfort involves the application of vibrational loads to full vehicles, seats and other automotive systems and components through which vibration can be felt by a driver or passenger. As loads are applied, the test operator physically occupies the test specimen, providing subjective feedback on the ride comfort experience. ISO 2631 routines for measuring and controlling an individual's vibration exposure [2].

As ride comfort becomes increasingly important in vehicle designs, the expertise and technology required to help you stay ahead of the competition through the accurate measurement and tracking of human vibrational exposure. Rather than performing evaluations of several vehicles traversing the same surface on the proving ground, this project can make improvements in ride comfort of tipper vehicles.

This report have complete CAD model and assembly of bogie suspension system for tipper vehicle using Creo 1.0 and optimize the cabin suspension with linear equations for **Bounce and Pitch** using Matlab 2013 with Simulink model. In this paper design and analysis of the rear suspension system of 6x4 tipper vehicle using Ansys 13.0

2. Theory

In the present work a 4x2 tipper vehicle cabin suspension is modeled and designed for optimum value of pitch and bounce amplitude. Using international standards for human body comfort while sitting in a moving vehicle, let us see what different kind of forces we generally face.

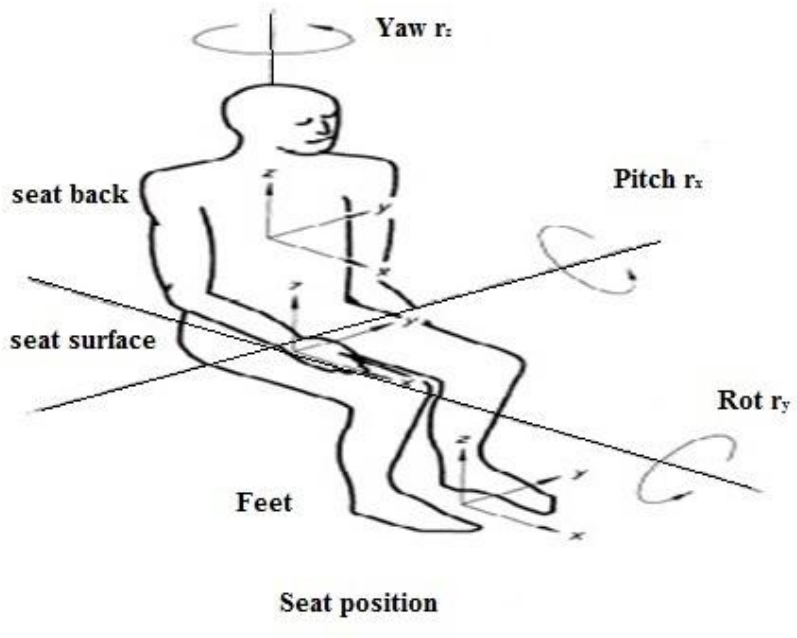

Fig 2.1: Basicentric axis of a human body [2]

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IS0 2631 defines techniques for the measurement of periodic, random and transient full-body vibration. It shows that the principal factors that combine to define the degree to which a vibration exposure will be acceptable. Informative annexes indicate current opinion and provide guidance on the possible effects of vibration on health, comfort and perception and motion sickness.

The frequency range considered is-

0.5 Hz to 80 Hz for health, comfort and perception, and

0.l Hz to 0.5 Hz for motion sickness [2].

Although the potential effects on human performance are not covered, most of the guidance on whole-body vibration measurement also applies to this area. This part of IS0 2631 also defines the principles of preferred methods of mounting transducers for determining human exposure. It does not apply to the evaluation of extreme magnitude single shocks such as occur in vehicle accidents. This part of IS0 2631 is applicable to motions transmitted to the human body as a whole through the supporting surfaces: the feet of a standing person, the buttocks, back and feet of a seated person or the supporting area of a recumbent person. This type of vibration is found in vehicles, in machinery, in buildings and in the vicinity of working machinery.

For better understanding purpose we have here line diagram of half lmd (low and medium duty) vehicles suspension system. We are here only going for 4x2 vehicles only.

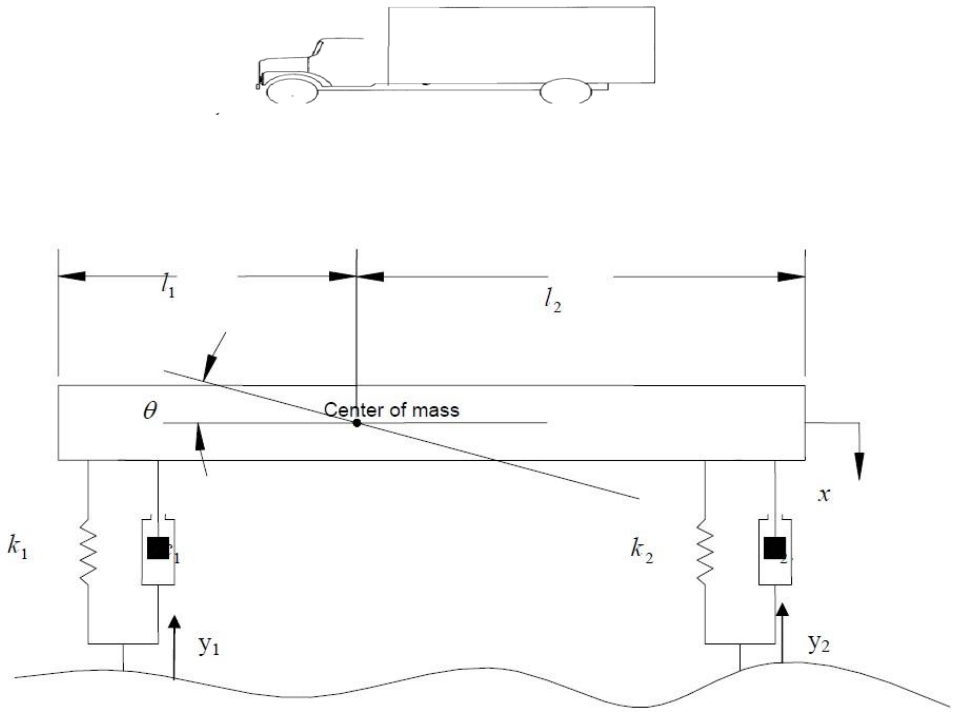


Fig 2.2: Half Car Suspension model of truck

2.2 Equations used-

The governing system of differential equations which describe the bounce and pitch motiton of the system in fig 3.2 are found using LaGrange's equations. The generalized form of the pitch and bounce motion are x(t) and θ(t) having notations [3][4][5].

The Kinetic energy described in equation 1 as:

$$T = 1\frac{1}{2}mx^2 + \frac{1}{2}J\theta^2 \quad (1)$$

The Potential energy is described as:

$$U = \frac{1}{2}k_1(y_1 - x + l_1\theta)^2 \quad (2)$$

Raleigh's dissipation function for viscous dissipation in dampers is:

$$Q = \frac{1}{2}c_1\left(y_1\dot{} - \dot{x} + l_1\dot{\theta}\right)^2 + \frac{1}{2}c_2\left(y_2\dot{} - \dot{x} - l_2\dot{\theta}\right)^2 \quad (3)$$

Lagrangean function L= T-U evaluated from (1) & (2) and together with (3) we get:

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}}\right) - \frac{\partial L}{\partial \dot{x}} = \frac{\partial Q}{\partial \dot{x}} \quad (4)$$

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\theta}}\right) - \frac{\partial L}{\partial \dot{\theta}} = \frac{\partial Q}{\partial \dot{\theta}} \quad (5)$$

The application of equations (4) and (5) yields:

$$m\ddot{x} + (c_1 + c_2)\dot{x} + (l_2c_2 - l_1c_1)\dot{\theta} + (k_1 + k_2)x + (l_2k_2 - l_1k_1)\theta = k_1y_1 + k_2y_2 + c_1y_1\dot{} + c_2y_2\dot{}$$

$$J\ddot{\theta} + (c_2J_2 - c_1J_1)\dot{x} + (l_2^2c_2 + l_1c_1)\dot{\theta} + (k_2l_2 - k_1l_1)x + (l_1^2k_1 + l_2^2k_2)\theta = k_2l_2y_2 - k_1l_1y_1 + c_2l_2y_2\dot{} - c_1l_1y_1\dot{}$$

Fig 2.3: Suspension attached to cabin

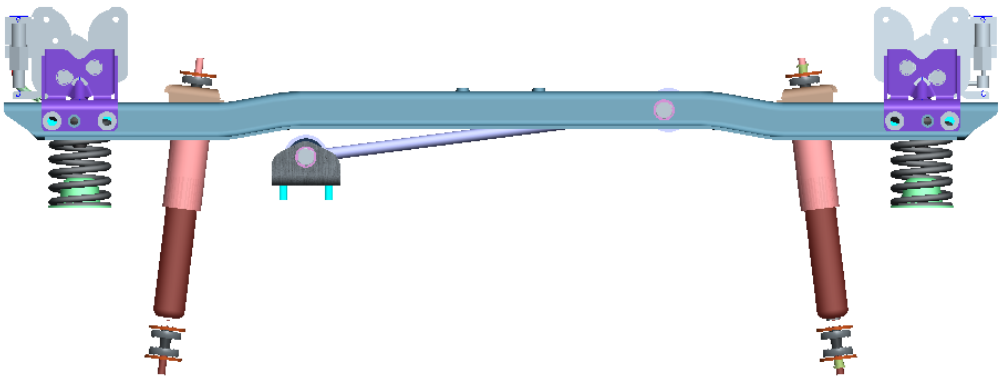


Fig 2.4: CAD model of Cabin suspension

3. SIMULINK MODEL

Optimization problems can be handled through MATLAB optimization package; however we will show here that SIMULINK model presented is an excellent tool to optimize the response of the system[1][5][7].

1st optimization: The data which is used for the optimization is of the vehicle 4x2 Haulage vehicle. But for the ease of calculations it is converted into 4x2 taking the front axle and two rear axles in unladen conditions.

Mvehicle= 10000 kg, mcabin=500 kg, kcabin front=5 kgf/mm, kcabin rear= 3.3 kgf/mm, Kvehiclefront=35.71 kgf/mm (eq), Kvehicle rear= 208.33 kgf/mm (eq)*, Cg= 2085mm (from front) length. We will choose a c = 6000 N.s/m in the model using equation (6), depicted below, in Figure 3.1

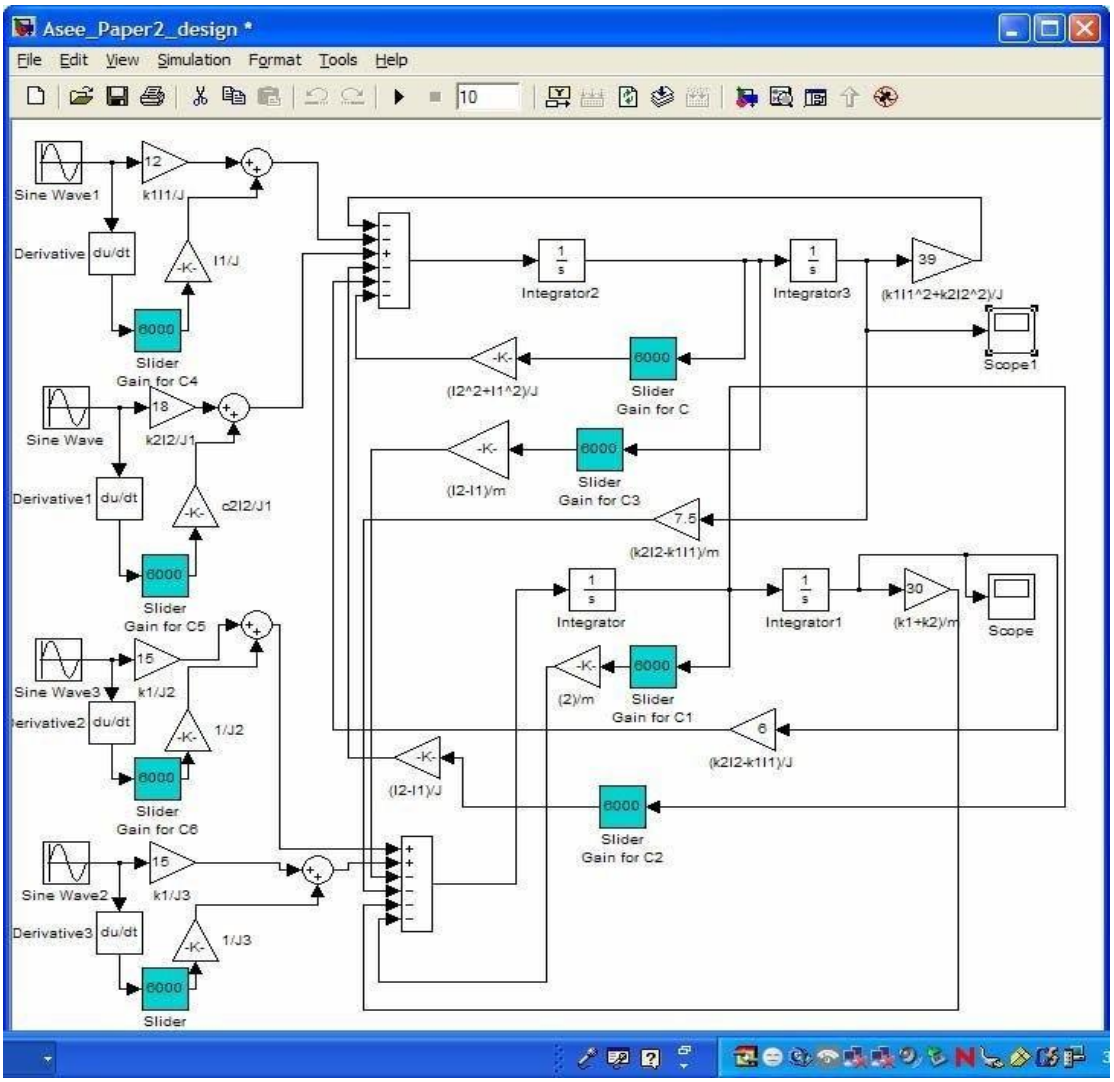


Fig 3.1: 1st Iteration Simulink Model

The responses for bounce and pitch motions are shown in Figures 6 and 7 respectively.

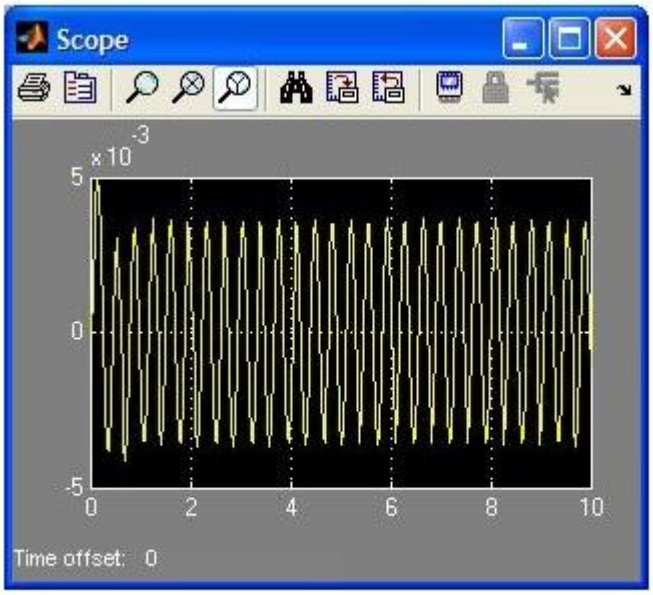


Fig 3.2: Bounce Amplitude (7mm)

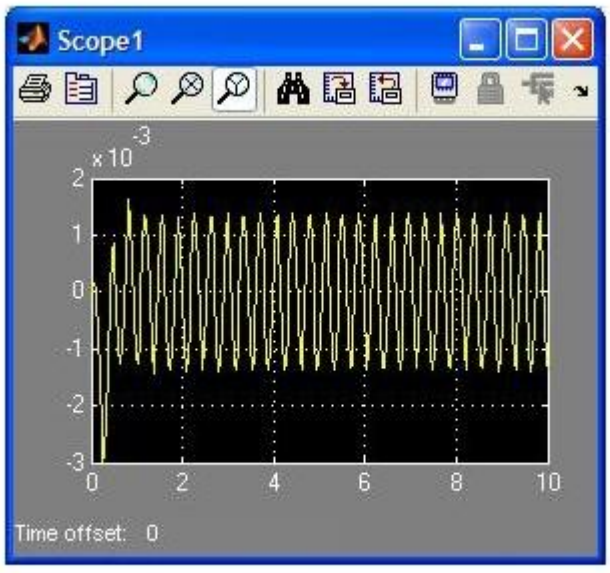


Fig 3.3: Pitch Amplitude (2.5x10-3 rad)

2nd Optimisation: Now let us try different values for damping coefficients by changing the "c" values in the slider gain.

Mvehicle= 10000 kg, mcabin=500 kg, kcabin front=5 kgf/mm, kcabin rear= 3.3 kgf/mm, Kvehiclefront=35.71 kgf/mm (eq), Kvehicle rear= 208.33 kgf/mm (eq), lCG to rear (2) = 4065mm, lcg to front (1) = 2085mm, Centre of gravity (Cg) = 2085mm (from front) length.

We will choose a c = 3000 N.s/m in the model using equation (6), depicted below, in Figure 3.4.

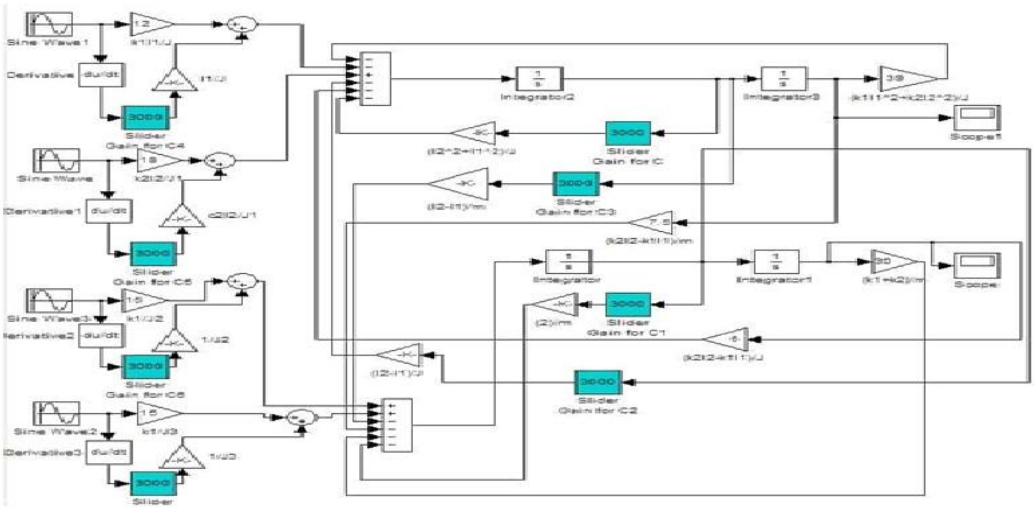


Figure 3.4: 2nd Iteration Simulink Model

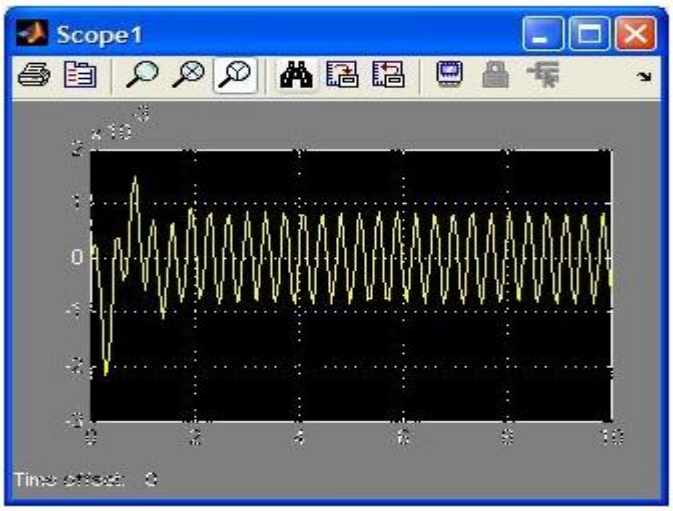


Fig 3.5: Bounce Amplitude (0.0248m)

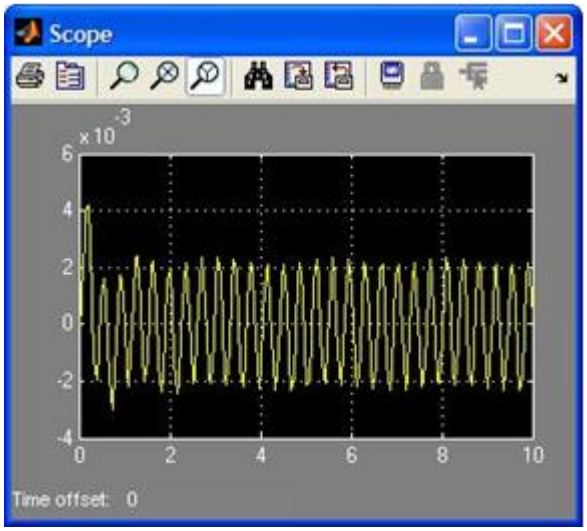


Fig 3.6: Pitch Amplitude (0.2061rad)

3rd Optimisation: Now let us try different values for damping coefficients by changing the "c" values in the slider gain.

Mvehicle= 10000 kg, mcabin=500 kg, kcabin front=5 kgf/mm, kcabin rear= 3.3 kgf/mm, Kvehiclefront=35.71 kgf/mm (eq), Kvehicle rear= 208.33 kgf/mm (eq), lCG to rear (2) = 4065mm, lcg to front (1) = 2085mm, Centre of gravity (Cg) = 2085mm (from front) length.

We will choose a c = 4500 N.s/m in the model using equation (6), depicted below, in Figure 11.

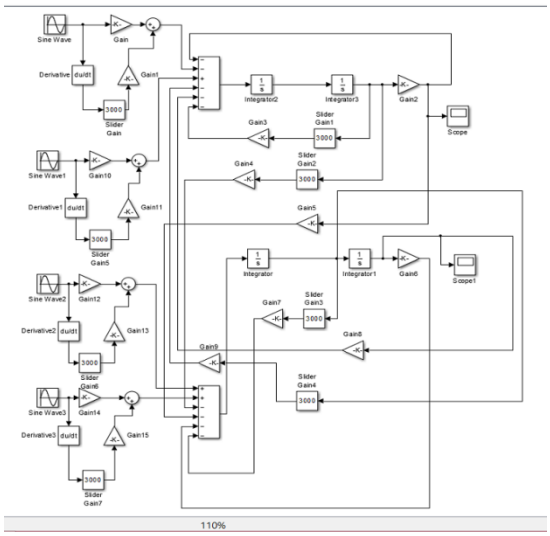


Fig 3.7: 3rd iteration Simulink Model

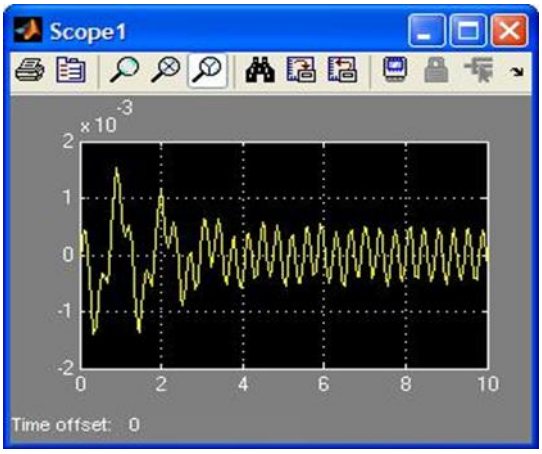


Fig 3.8: Bounce Amplitude (2.6mm)

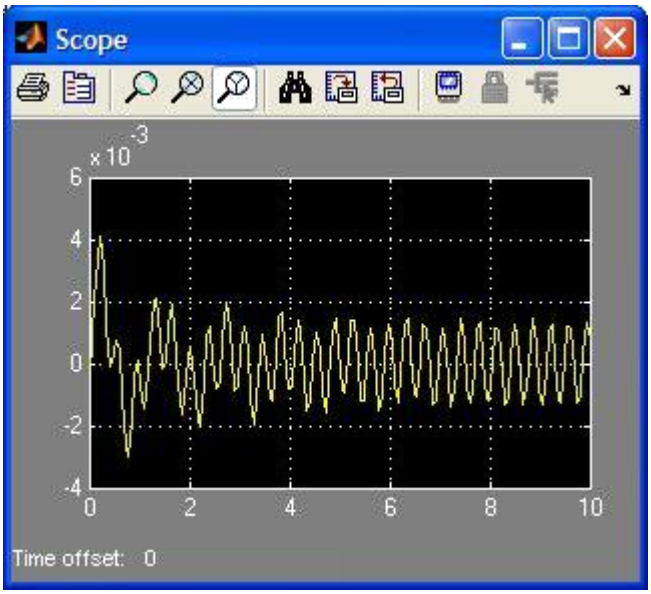


Fig 3.9: Pitch Amplitude (1x10-3 radians)

RESULTS AND DISCUSSION

Comparison between results:

Table: 3.1

Iteration No.	Damping co-efficients	Bounce Amplitude	Pitch Amplitude
1.	3000Ns/m	0.0248m	0.2061 rad
2.	4500Ns/m	0.0264m	1x10-3rad
3.	6000 Ns/m	0.0067m	2.5x10-3 rad

Now we can see in this range of pitching and bouncing frequency of cabin or whole half car model about CG is accepted only when it is less.

So our first interpretation is the true one. Because after both of the iteration we have seen the bounce amplitude increased drastically from 0.0067m to 0.0264m and it is also noticed that further reducing the damping coefficient on same scale we get a little variation.

Therefore from the above results we can say that this is the final accepted result of Bounce and Pitch amplitude in unladen condition with sine curve input.

Pitch Amplitude = 0.2061 radians or 11.41degree.

3 CAD model of rear leaf spring:

As in this paper we are also designing the rear leaf spring of tipper vehicle of 4x2 model, we have also created the new CAD model for the vehicle. This suspension system modified from normal bell crank multi leaf spring to single Hotchkiss (bogie) suspension type.

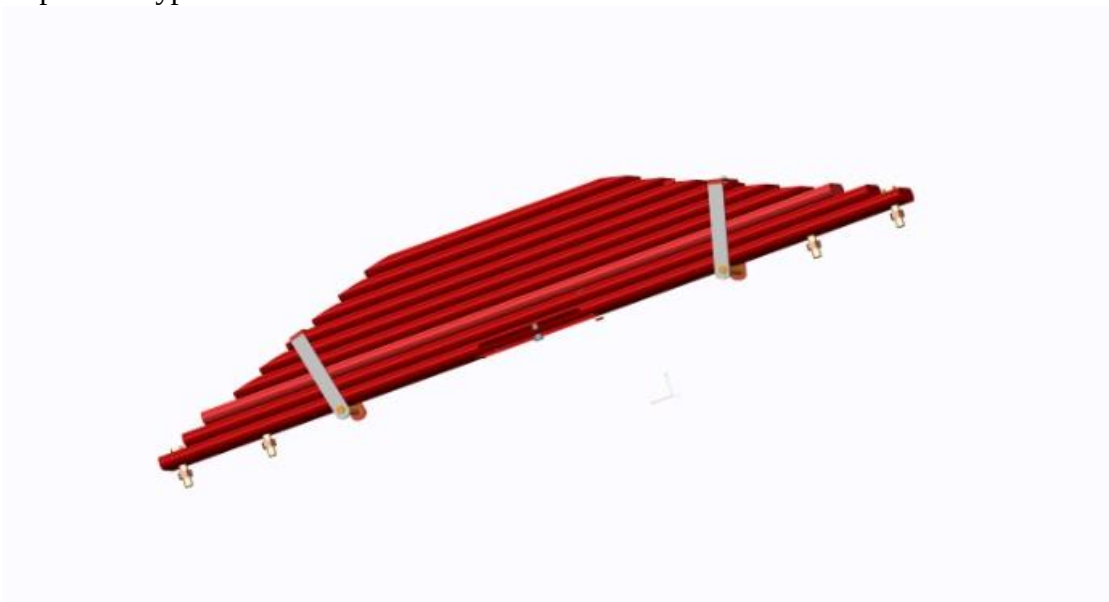


Fig 3.10: New CAD model of Leaf Spring for 6x4 tippers

Fig 3.11: Leaves with Trunniun Hub

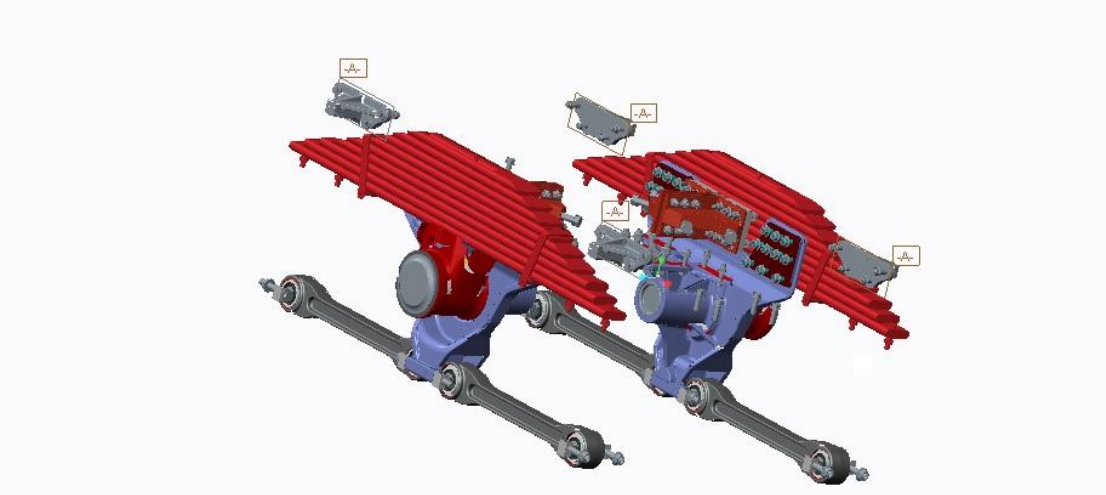


Fig 3.12: New CAD model of Bogie suspension for 6x4 tippers without U bolt and bilateral torque rod

4 Results and Discussion:

4.1 Design Parameter of Leaf Spring

Leaf steel spring use in this work includes:

i. Span, 1025 mm;
ii. Arc height of axle seat (camber), 90.8 mm;
iii. Width of leaves, 60mm;
iv. Thickness of leaves, 16 mm:
v. Full bump loading 7500kg.

4.2 Material Properties

Parameter Value

i. Material selected- Steel 65Si7/SUP9
ii. Young's Modulus, I – 2.1* 105 N/mm
iii. Poisson's Ratio – 0.266
iv. BHN 400-425
v. Tensile strength Ultimate – 1272 Mpa
vi. Tensile strength Yield – 1158 Mpa
vii. Spring stiffness – $K_{vehiclefront}$=35.71 kgf/mm(eq), $K_{vehicle\ rear}$= 208.33 kgf/mm(eq)
viii. Mass – 54.165 kg
ix. Normal Static loading – 35000 N
x. Density – 0.00000785 Kg/mm (785 Kg/m^3)
xi. Behavior Isotropic (assumed).

4.3 Multi Leaf Spring Boundary Conditions:

The multi leaf spring is modeled in position having maximum deflection i.e. flat position. The boundary condition is the collection of different forces, pressure,

velocity, supports, constraints and every condition required for complete analysis. As per specifications the spring is drawn at flat condition, therefore the load is applied in downward direction to achieve initial no load condition[7][8][10].

As no load camber is considered as 153°, a joint rotation of 27° is considered for both revolute joints, during static analysis.

Existing leaf spring is designed under the given conditions and load. In this report we have calculated only static structural analysis and performed for:

(i) total deformation andvon misses stress

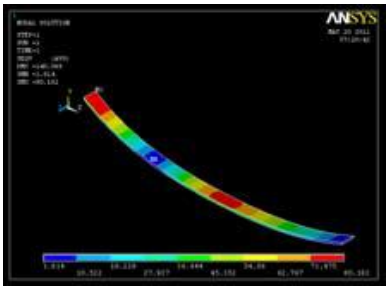


Fig 4.1: Von- Mises Stress Analysis of main leaf

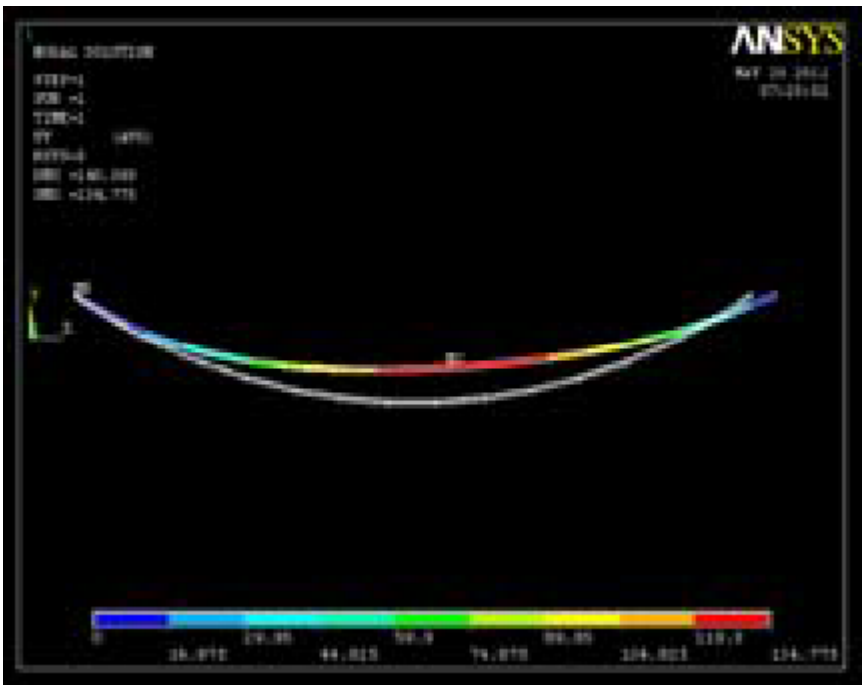


Fig 4.2: Total deformation of main leaf

From the above results we have:

(i) Von- misses stress = 213.25 kgf/mm^2 at 35000N load which is just par of

permit zone or safe zone.

(ii) Deflection = 257 mm

Analysis using ansys:

Applying load conditions and boundary conditions and explaining in detail/

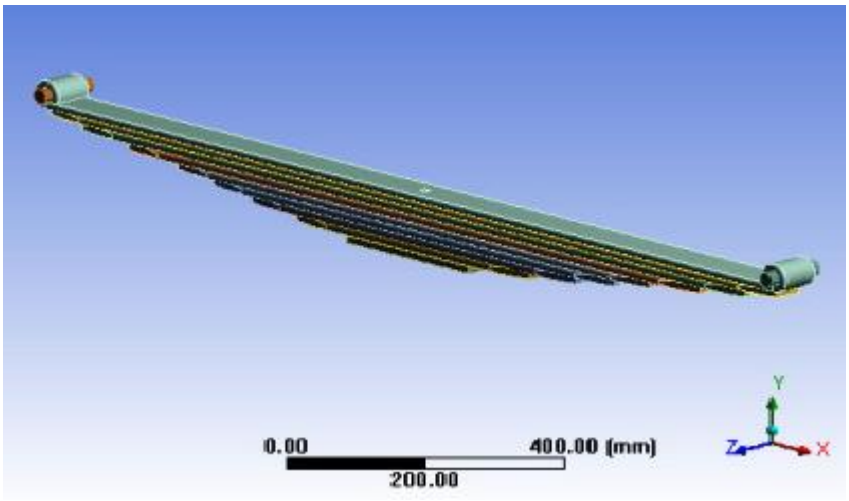


Fig 4.3: CAD imported to ansys

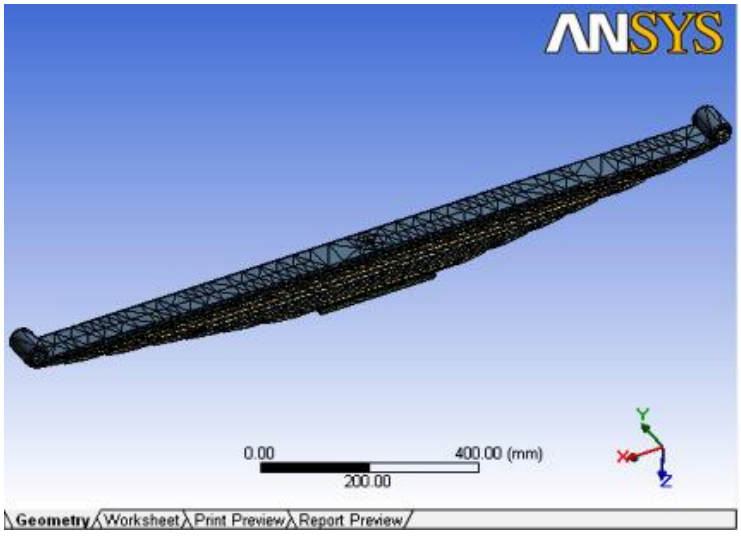
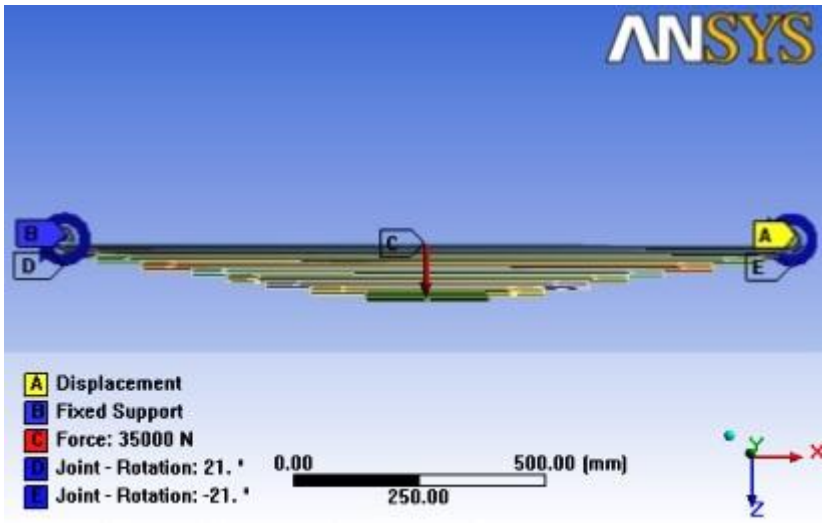
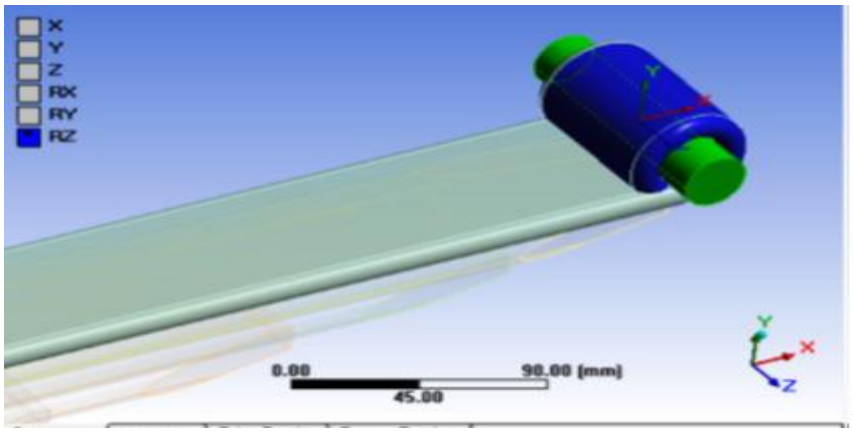


Fig 4.4: Mesh (free mesh) in Ansys



(a)



(b)

Fig 4.5: Boundary Conditions Applied

Now we will calculate the results for two loads which are 35000N and 17000N for conformation of permissible zone of total deformation and von misses stress generated.

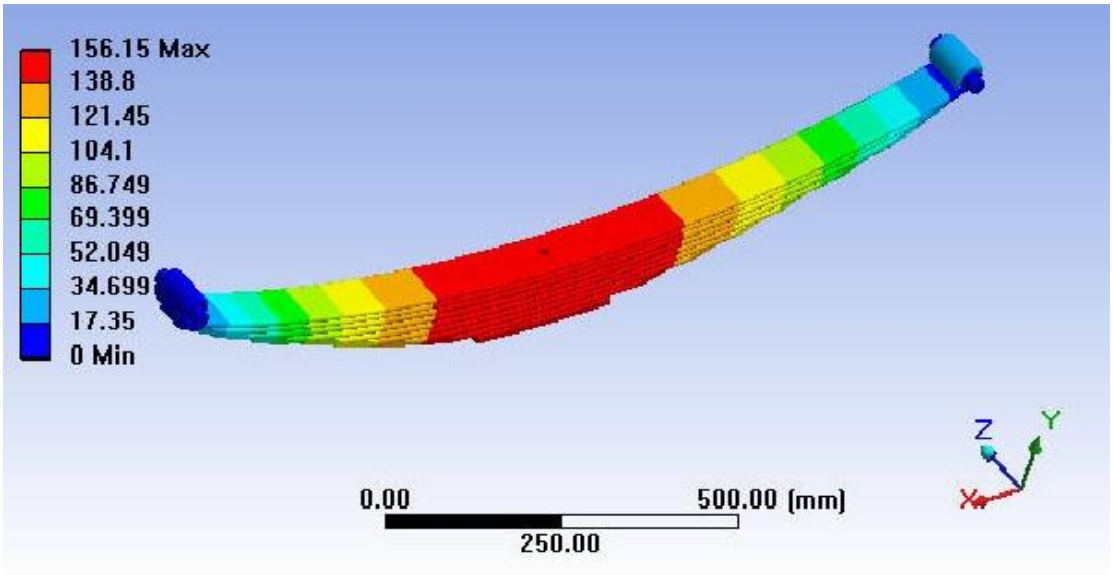


Fig 4.6: Total deformation at 35000N load

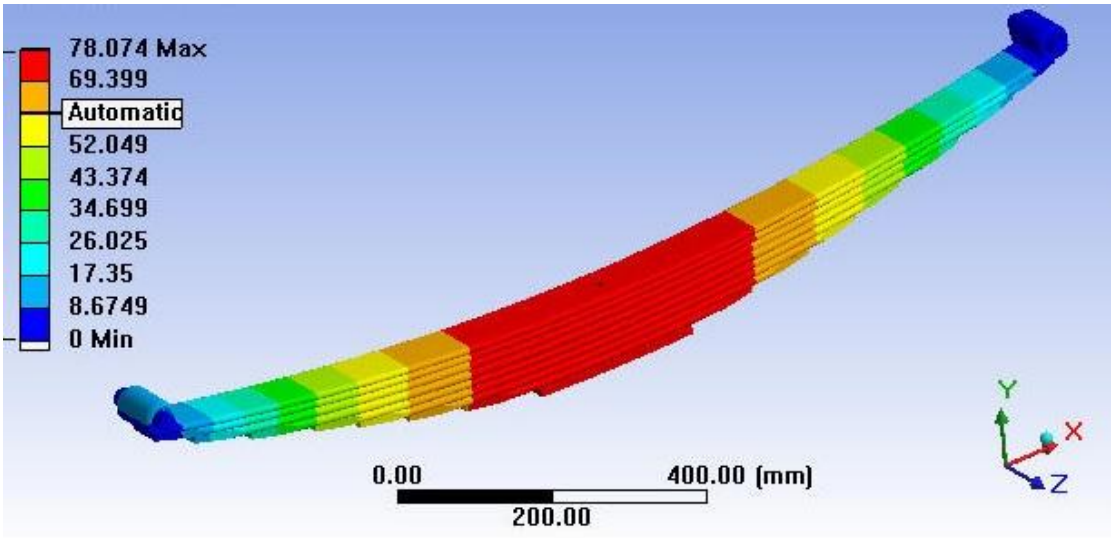


Fig 4.7: Total deformation at 17000N

Table:2

No. of Leaves	Total Deformation	Von-Misses stress
7 Leaves (35000N)	257 mm	213.25 Kgf/mm^2
9 Leaves (35000N)	156mm	141.56 Kgf/mm2
9 Leaves (17000N)	78.074mm	53.77 Kgf/ mm2

Conclusions:

From all of the above analysis now we can conclude that we have fulfilled all the basic necessary needs of ride comfort analysis. The design analysis carried out for ride comfort and for that I have used Matlab 2013 (Simulink model) to find out frequency of vehicle, pitch amplitude and bounce amplitude.

Then we carry out benchmarking of existing 6x4 tipper vehicle's bogie suspension (Hotchkiss suspension). It is of 7 leaves model with bilateral torque rod previously. But as per results we got by analyzing the main leaf in ansys we found it as not suitable for 35000N load. So we have changed that model with 9 leaves and analyzed it. In that analysis we got deformation of leaves as 156mm which is far better than the previously leaf deformation of 257mm.

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