DYNAMIC ANALYSIS OF FRONT AXLE CAR WISHBONE SUSPENSION USING FEM

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Abstract: Suspension system is the base for reduction of vibration and load transfer. Suspension system is designed to take structural loads and dynamic loads resulting from motion of the vehicle. Due to the advances in computer based CAD and CAE Software's, design simulation helps in proper design of the suspension system which is difficult with complex theoretical formulations. The finite element software's are developed to consider linear to nonlinear material behavior along with simple static to complex spectrum loads. Generally front axle is made independent for smoother steering and along for traversing the curves. In the present work, a front axle suspension used for a new design is considered for finding the structural stability.

***Index Terms* – Suspension system, Wishbone suspension, Static Analysis, FEM.**

I.INTRODUCTION TO INDEPENDENT SUSPENSION:

So- named because the front wheel's suspension systems are independent of each other (except where joined by an anti-roll bar). These came into existence around 1930 and have been in use in one form or another pretty much ever since then.

Fig 1.: Wishbone Suspension

DEPENDENT SUSPENSION:

TRANSVERSE LEAF-SPRING:

This system is an odd in that it combines independent double wishbone suspension with a leaf spring like you'd normally find on the rear suspension. Famously used on the corvette, it involves one leaf spring mounted across the vehicle, connected at each end to the lower wishbone. The center of the spring is connected to the front sub frame in the middle of the car. There are still two shock absorbers, mounted one to each side on the lower wishbones. Chevy insist that this is the best thing since sliced bread for a suspension system but there are plenty of other experts, manufactures and race drivers who think it's junk. It's never been clear if this was a performance and design decision or a cost issue, but this type of system is very rare.

- ### HYDROMANTIC SUSPENSION:

Hydromantic is a type of space efficient automotive suspension system used in many cars. The system replaces the separate springs and dampers of a conventional suspension system with integrated, space efficient, fluid filled, displacer units, which are interconnected between the front and rear wheels on each side of the vehicle. Each displacer unit contains a rubber spring, and damping is achieved by the displaced fluid through rubber valves. The displaced fluid passes to the displacer of the paired wheel, thus providing a dynamic interaction between front and rear wheels.

II.Literature on Suspension

An independent rear suspension is a suspension with each tire of the car free of the others. For the purposes of formula SAE racing, independent suspensions the best designs. One of the advantages to independent suspension systems is its superb ability to maintain contact with the road [5]. There are four main types of independent rear suspension designs that race car engineers consider when choosing the right design. Some designs are meant to cushion the driver from bumps and bad road conditions using significant suspension travel while others are designed to maximize the tires contact area with the driving surface. Formula car engineers are focused on maintaining high tire contact patch area [5]. The four types of suspension designs are as follows; Trailing arms, swing axel, Macpherson struts, and SLA. Each of these designs have slightly modified sub-designs [4].

The first design for discussion is trailing arms. A **trailing arm** suspension design is pivots on a line perpendicular to the center line of the car. This design has problems with toeing out of the tires during cornering and producing bending and deflection of links [4]. The design also produces problems with camber angle and loss of tire contact patch. **A swing axle** is a suspension design that pivots near the center of the chassis [5]. Other than being a relatively easy design, the swing axle has several negative attributes. The design typically produces a geometry with a high roll center, and as a result of that, the design will produce large jacking forces [5]. Jacking is the tendency for the car to lift a tire on the inside of the car during cornering [2]. With this said, it is important to reiterate that with these disadvantages, camber angle is reduced producing decreased cornering power. This is caused by the loss of tire contact patch area. Herb Adams says; "Since the jacking only happens during hard cornering, the loss of cornering power happens when it's needed most" [5].

The **Short Long Arm** design is widely used by many formula race car teams for the reason that its geometry through motion controls the camber angle. This geometry is the best design for keeping a large tire contact patch throughout the travel of the suspension. SLA suspension design consists of a double-A-arm with the top link shorter than the bottom [4]. This design produces a negative camber angle as the suspension is compressed, keeping the tires perpendicular to the ground which helps with higher cornering speeds [5].

One important type of dependent suspension systems is live axels. Live axles are relatively easy to design and to manufacture. Additionally, this design has been known to perform well in formula competitions. According to Adams, "On smooth roads, it is usually difficult to see any advantage for an independent rear suspension." [5] This means that live axle systems should not be overlooked for designing a car rear suspension. For a live axle suspension, the roll center of the car will be in the very center of the differential, meaning that it is higher than other designs mentioned above [2]. In live axles, there are methods of controlling side to side motion suspension such as pan hard and watts linkages [5].

III. Theoretical framework

Newton's law can be applied to a free-body diagram of an individual system, as shown in

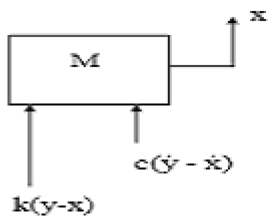


Fig:2 Free body diagram

A summation of forces yields the following governing differential equation of motion:

$$m\ddot{x} + c\ddot{x} + kx = c\dot{y} + ky \text{ .... (1.1)}$$

A relative displacement can be defined as z = x-y. The following equation is obtained by Substituting this expression into equation (1):

$$m\ddot{z} + c\dot{z} + kz = -m\ddot{y} \text{ .... (1.2)}$$

Additional substitutions can be made as follows,

$$\omega_n^2 = \frac{k}{m} \text{ .... (1.3)}$$

$$2\zeta\omega_n = \frac{c}{m} \text{ .... (1.4)}$$

Note that ζ is the damping ratio, and that ω_n is the natural frequency in radians per second.

Further more ,ζ is often represented by the amplification factor Q, where Q=1/ (2ζ).

Substitution of these terms into equation (2) yields an equation of motion for the relative Response

$$\ddot{z} + 2\,\zeta\omega_n\,\dot{z} + \omega_n^2 z = -\ddot{y}(t)\text{..................(1.5)}$$

Equation (1.5) does not have a closed-form solution for the general case in which y(t) is an arbitrary function. A convolution integral approach must be used to solve the equation. The convolution integral is then transformed into a series for the case where y(t) is in the

Form of digitized data. Furthermore, the series is converted to a digital recursive filtering relationship to expedite the calculation. The resulting formula for the absolute acceleration is

$$\ddot{x}_i = +2\exp[-\zeta\omega_n \Delta t]\cos[\omega_d \Delta t]\,\ddot{x}_{i-1}$$

$$-\exp[-2\,\zeta\omega_n \Delta t]\,\ddot{x}_{i-2}$$

$$+2\,\zeta\omega_n \Delta t\,\ddot{y}_i$$

$$+\omega_n \Delta t \exp[-\zeta\omega_n \Delta t]\,\{[\omega_n/\omega_d (1-2\,\zeta^2)]\sin[\omega_d \Delta t]-2\zeta\cos[\omega_d \Delta t]\}\,\ddot{y}_{i-1} \quad \text{......(1.6)}$$

IV.PROBLEM DEFINITION & FINITE ELEMENT MODEL DEVELOPMENT

Suspension is a very important component in the automobile design. Proper suspension helps in reducing the stresses developed in the chassis and body of the automobile. Also human comfort is decided by the level of suspension. Since suspension connects the chassis to the axle structure, any failure of suspension is very critical. So present analysis is carried out for finding the structural integrity of a new design of car.

PROBLEM DEFINITION

Cad Design and Analysis of Front axle independent suspension system to with stand structural and dynamic loads for structural integrity of Wishbone suspension. Also to estimate the factor of safety in the different components. Here the objectives include

1) Literature on Spectrum analysis and Loads on Suspension systems
2) Cad drafting and three dimensional modelling of the Suspension system
3) Meshing and Importing to Ansys for analysis
4) Analysis for the given loads to check the structural integrity
5) Spectrum response analysis

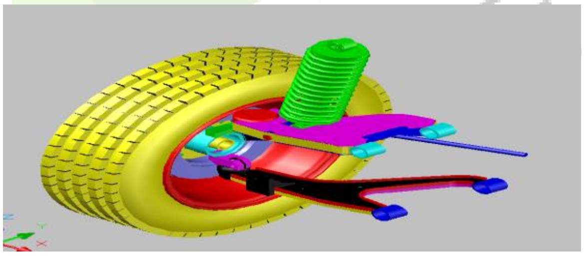


Fig 3 : New design of Suspension

MATERIAL

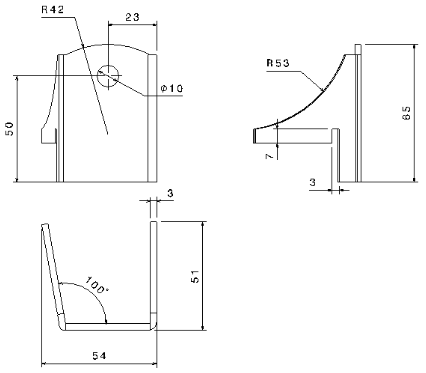
Material: Structural steel
Yield stress: 550 N/mm^2.
Poison's ratio: 0.3
Allowable stress: 275 N/mm^2.
FOS: 2

DESIGN SPECIFICATIONS

1) Stresses should be within the Yield Stress(Factor of Safety should be more than 2
2) Maximum deflection should not cross 0.937 mm.
3) As per IS standards allowable deflections for the beam is 1mm for 350mm span.

Cad Models

The problem specification Cad modeling is done and draft model is as represented as follows. The structure comprises two wishbone (Upper and bottom), damper arrangement.



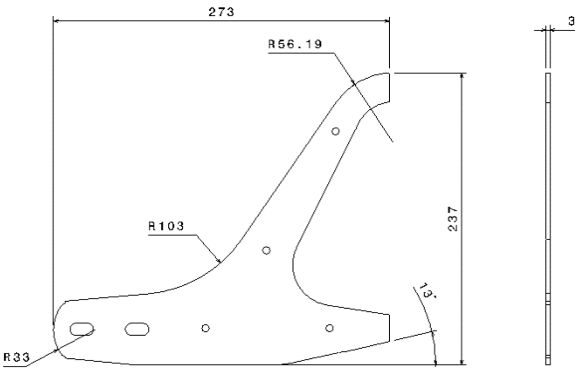


Fig:4 Dimensional view of the Component Fig:5 Bottom wishbone Dimensions

The figure 6 &7 shows component dimensions of the problem. All dimensions are represented in mm. Bend radius of the pipe is given as 380mm. Sleeve length is around 300mm to accommodate the hanger flange. Rib dimensions are increased towards the bottom to increase the strength of the joint at the base.

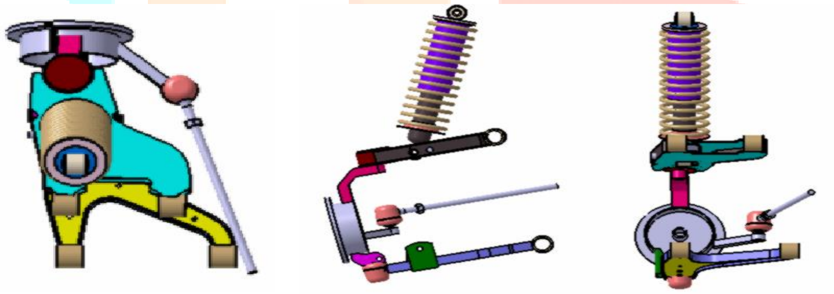


Fig: 6 Front Independent Suspension System for Car

The three dimensional views shows Suspension system cad design with part details. The ribs are used to improve the strength of the structure. The components like coil spring, strut, upper and lower wishbone and torsion arm is shown in the figure. Individual parts are separated and figures are captured for representation and shown in the drafting views.

Load cases:

Totally 3 different load cases are carried out to analyze the suspension. Each load case with its load considerations are represented as follows.

Table 1: Different Load cases

Sl. No	Load case no
1	Spectrum load in Longitudinal Axis
2	Spectrum load in Lateral Axis

The table represents applied boundary conditions on the problem. Various load cases are considered based on the loads coming in the operational and working conditions. The spectrum loads are generated due to road vibration.

MESHED PLOTS

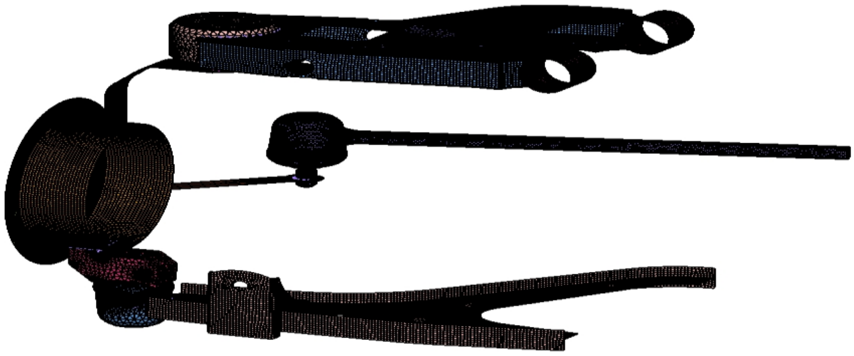


Fig 7: Structural Mesh (Combination of shell and solid elements)

Initially the mid surfaces are extracted using Hypermesh and later split the geometry to ease the quad meshing. Generally quad mesh gives better results. All the quality criteria like aspect ratio, jacobian, skew, and warpage are checked for better results. The connections are maintained through RBE3 and coupling constrains.

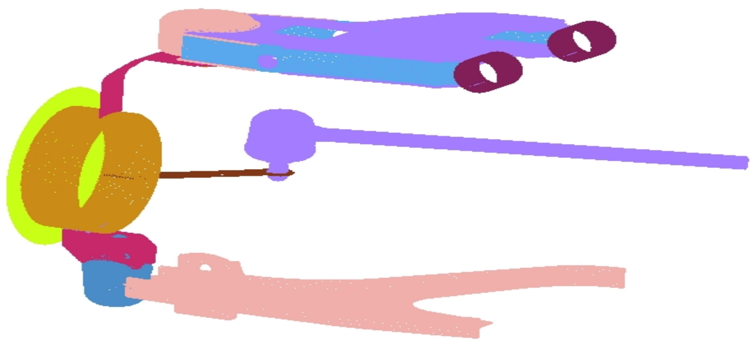


Fig 8: Wire Frame form of mesh

Individual components are represented with different colors for visualization and for applying different cross sections. Most of the wish bone structures are made of 3mm thickness and Connections members to the knuckle is through bigger sized sections (Top 20mm and bottom 25mm). Even the torsion bar connection is also of 15mm thickness.

ASSUMPTIONS:

1) Materials are assumed to be isotropic and homogeneous.
2) A linear element solid45 is considered for stress analysis.
3) RBE3 element is used for load transfer.
4) A complete contact is considered between the members
5) Total 5 load cases are considered for analysis.

V.RESULTS FOR ANALYSIS

Structural stress estimations are very important to find the stress distribution and safety of the problem. The advantage of Finite element method is its ability to view the results on individual components and also at the place of interest. Only the requirement here is the finer mesh for better results. Since the suspension is very critical component, the mesh is imported to ansys and appropriate properties are assigned before going for actual execution.

The design of new suspension is analyzed for the given structural loads and dynamic loads. The results are presented for stress and deformation conditions. Following load cases are represented for the problem

Case 1: Spectrum analysis for vertical loads

Case 2: Spectrum analysis for longitudinal loads
Case 3: Spectrum analysis for lateral loads.

MODAL ANALYSIS

Modal analysis has been performed to find resonance frequency in the suspension system due to the rotating member (axle) and vibrational loads. Identification of initial natural frequencies of the system helps in avoiding the particular frequency in application. The modal results are as follows.

THEORETICAL CALCULATION TO FIND OPERATIONAL FREQUENCY

Maximum speed = 135 KMPH Diameter of wheel (d)= 0.6 m
Velocity v = 135*1000/3600 = 37.5 m/sec

$V = \pi \, dn^1$
$N^1 = 37.5/\pi * .6$
$N^1 = 19.89$ HZ

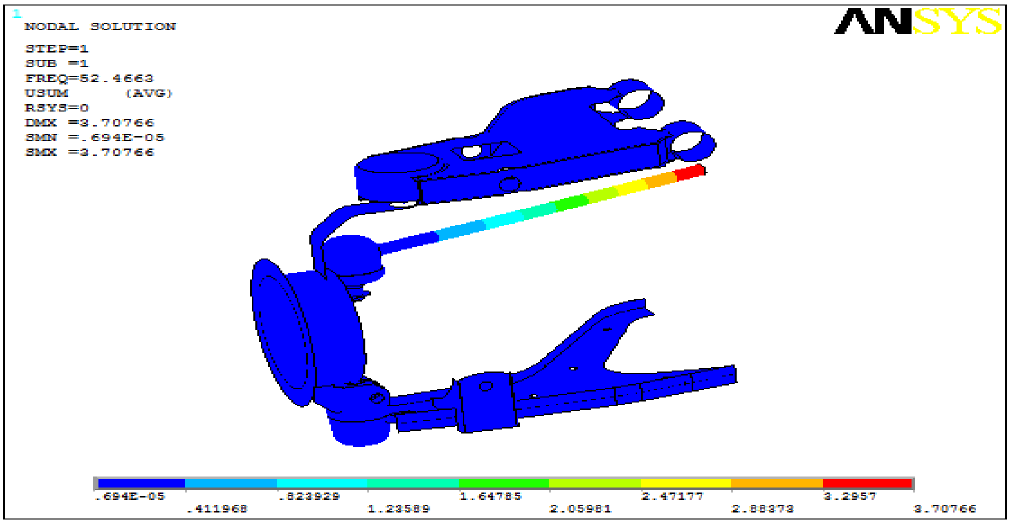


Fig 9: Mode shape corresponding first fundamental frequency 52.46Hz

The figure 9 shows mode shape corresponding to the fundamental frequency 52.46 Hz. Maximum vibrations is observed at the unsupported torsion bar. Since axle portion is fixed, displacements are generally less at the constrained region. The problem works like a simply supported beam where maximum deformation acts at the center. Other mode shapes also shown in the following figures. Mode shapes helps in finding weak directions and regions in the problem under loading.

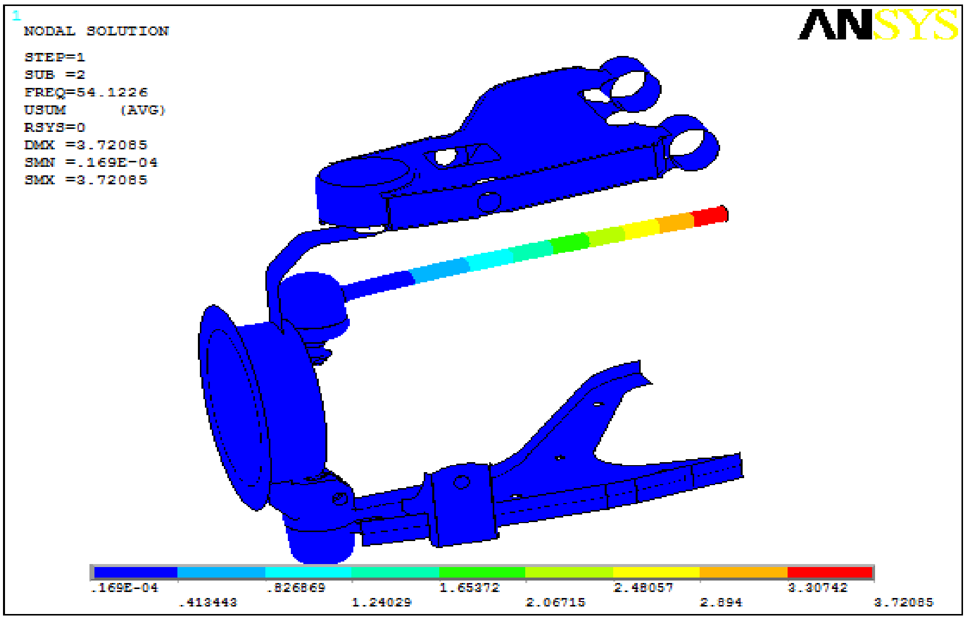


Fig 10 Mode shape for second 54.1226 Hz

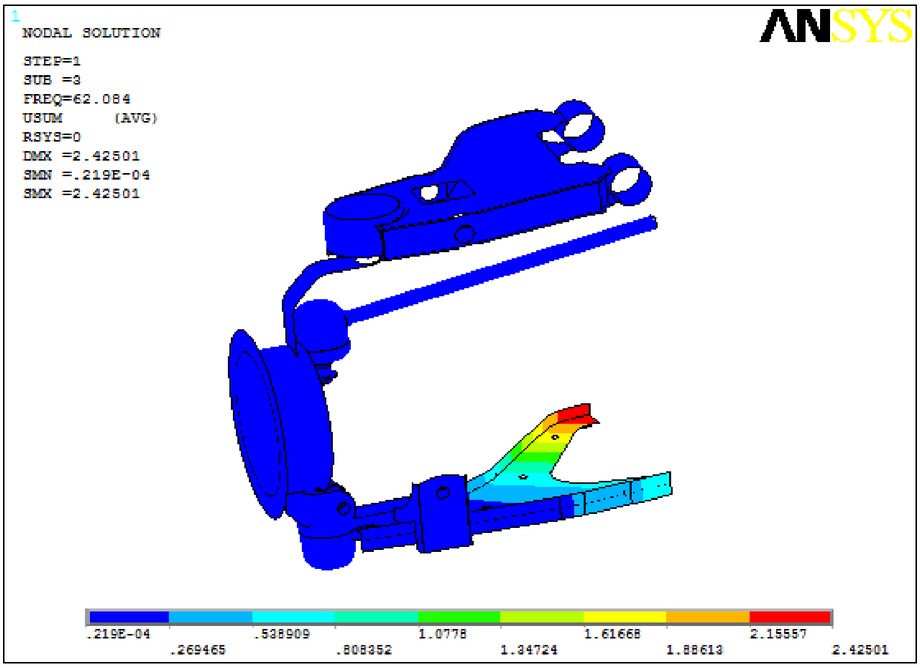


Fig 11: Mode shape for mode shape r62.084 Hz

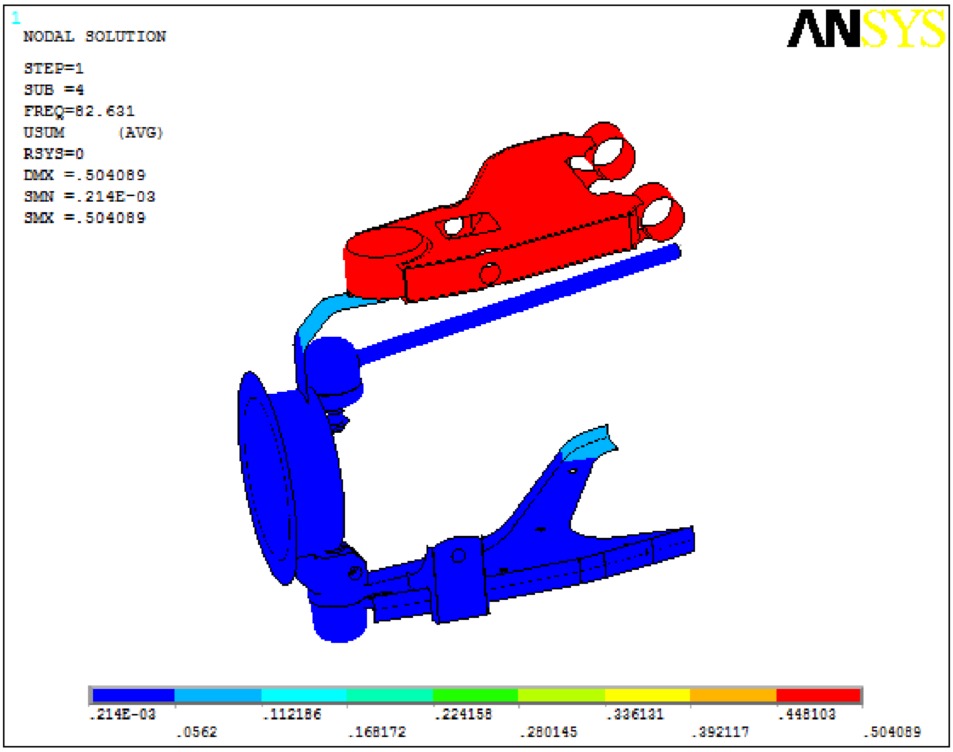


Fig 12: Mode shape for fourth 82.631 Hz

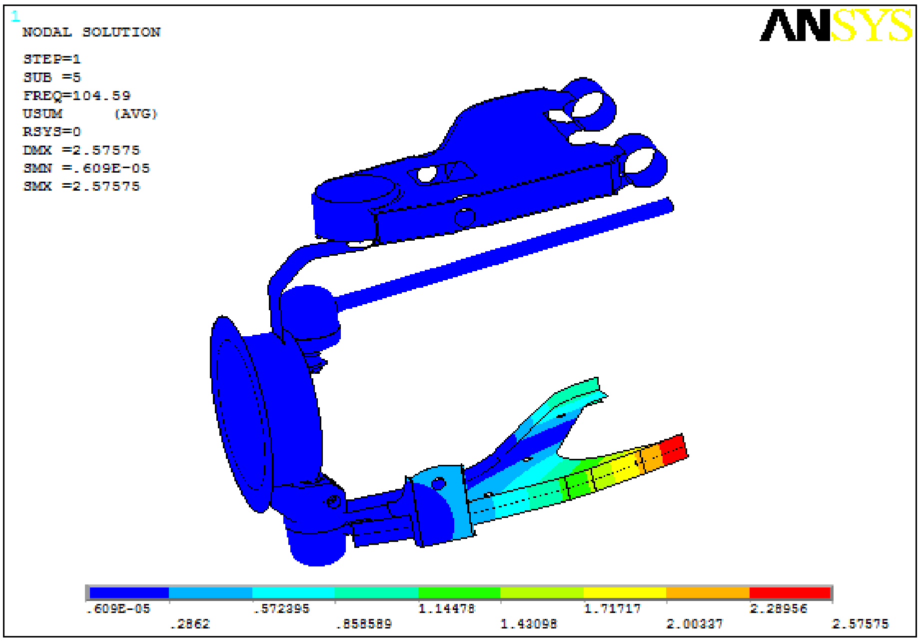


Fig 13: Mode shape for fifth 104.59 Hz

TABLE 1: DYNAMIC COMPARISON TO AVOID RESONANCE

Details	Operational Frequency(Hz)	Natural Frequency(Hz)
1	20	52.4

The table 1 shows modal frequency comparison. Normal running frequency is 20Hz for the given vehicle and the first natural frequency is 52.4Hz. So Resonance will not take place in the suspension system due to its higher value.

So the overall analysis shows complete safety of the suspension system for the given loads. All the relevant pictures are represented along with graphical plots.

ACCELERATION SPECTRAL DENSITY (ASD) DATA

(*Based on the road excitation profile from the MIL STD 810-D*)

Observations for all spectrum data are as follows.

- The results are plotted only for structurally important components which is experiencing high resultant stresses.
- All the stress values plotted are in Mega-pascals (MPA).
- Singular stresses are avoided for better results.
- RMS Stress values indicate the "Root Mean Square" of the stress values observed during a Random vibration analysis. The actual stress values observed will not cross this value during 68.3% of the random vibration duration.
- The actual stress values observed in any component will be less than three times RMS stresses (3 Sigma) observed in it for 99.73% of time. The design is considered safe against yield failure, if the 3 Sigma (3RMS) stresses developed in it falls below the yield strength of its material.
- The 'Damage ratio' is a number calculated using Miner's rule which indicates the probability of failure under fatigue loading vibration for a fixed duration. The design is considered safe against fatigue, if the cumulative damage ratio is less than one.

SPECTRUM LOADS

Table 2: Input data for Spectrum loads in tabular form

Frequency (Hz)	Acceleration Amplitude (g^2/Hz)		
	Vertical (Y)	Longitudinal (Z)	Transverse (X)
0	0	0	0
4	0.01973	0.00855	0.00034
10	0.01973	0.00855	0.00034
20	0.01973	0.00855	0.00171
30	0.01973	\	0.00171
40	0.01973		\
78	\		0.00005
79			0.0005
100	0.0036	0.00119	0.0005
RMS Value	1.088g	0.606g	0.238g

Table 3: THE RESULTS ARE TABULATED BELOW

Description	Deflection	Static Stress	Spectrum Stress				
			1 Sigma Stress (Mpa)	3 Sigma Stress (Mpa)	Total Stress	Yield Stress	Factor of Safety
Spectrum-Longitudinal	0.15*3	157	7.28	21.84	157+21.84=178.84	550	3.07
Spectrum – Lateral	0.0895*3	157	4.4	13.2	157+13.2=170.2	550	3.23

The table 3 shows stress and deflection development for different load cases. The results show almost a constant maintenance of factor of safety more than 2. Since it is satisfying the requirement for factor of safety, the structure can with stand the given loads.

BUCKLING ESTIMATES FROM BOTH THEORETICAL AND FINITE ELEMENT SOLUTION

The problem type is one end fixed and other end subjected compressive load.

So buckling load is $P_{cr} = \pi^2 EI/4L^2$

Moment of inertia of the rod: $\pi d^4/64 = 3.14*36^4/64 = 82448 mm^4$.

Buckling load Pcr=1407835N

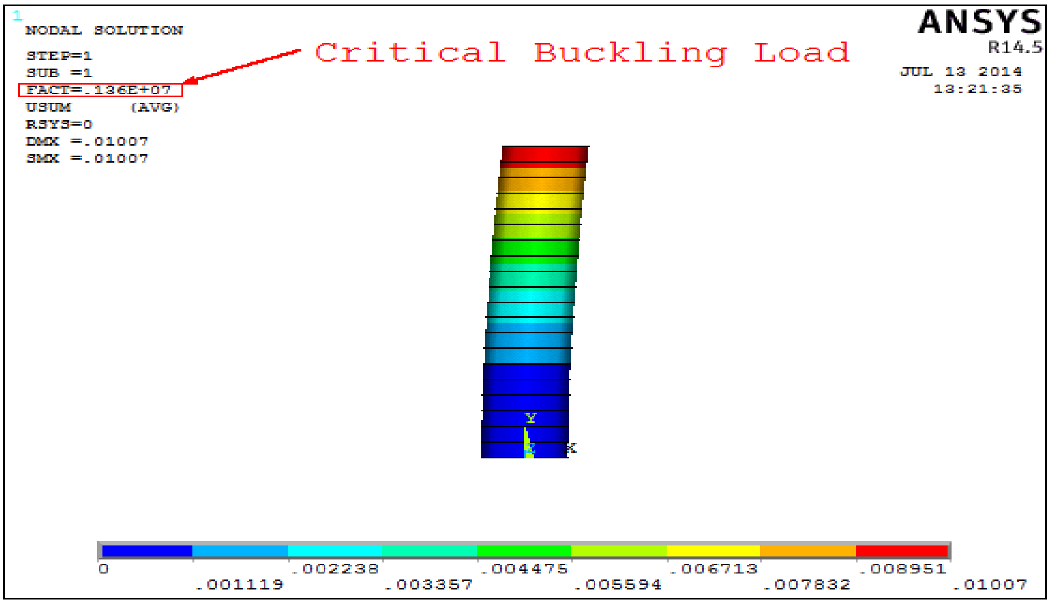


Fig 14: Buckling Load

With reference to finite element method using one dimensional concept
Pcr=1358700N
Difference: 1407835-1358700=49135N
Percentage Error =49135*100/1407835=3.49%

VI. Conclusion

The front independent suspension used in an automobile system is analyzed for structural stability condition. 5 practical load cases are considered to check the safety of the structure. Here main objective is to find the load carrying capacity along with modal frequency conditions. Upper and bottom wishbone, torsion bar, connecting members are the main members in the suspension assembly. Initially model is built with Catia V5 total 5 load cases are consider

1. Spectrum Analysis for Vertical loads
2. Spectrum Analysis for longitudinal loads
3. Spectrum Analysis for Lateral loads

In the first two load cases the stress obtained from FE Analysis is compared with allowable stress. The induced stresses are 8.89 N/mm^2 and 157 N/mm^2 for first and second load cases respectively. The induced stresses are less compared to the allowable stress (275 N/mm^2), hence the structure is safe for self-weight and structural loads. Then modal Analysis has been performed to find Resonance frequency in the suspension system due to the rotating members and vibrational loads. In the Modal Analysis. The operational frequency or Normal running Frequency obtained theoretically is 20 HZ for the given vehicle and first Natural frequency is 54.12 HZ. Hence the overall Resonance will not take place in the suspension system due to its higher value. Three cases of spectrum loads as per the mil standards are applied on the suspension system. They are spectrum vertical, spectrum longitudinal and Spectrum lateral. The static stress considered for all three cases is 157 N/mm^2. Spectrum Stresses are 250.3N/mm^2, 178.84N/mm^2, 170.2 N/mm^2 for the first, second and third load cases. This can be mainly attributed to lesser cross section at the hinge pin regions. Generally vonmises stress is considered due to its applicability for predicting the failure of the ductile components. Yield Stress considered is 550 N/mm^2. The Factor of safety obtained is 2.19, 3.07, 3.23 for first, second and third load cases respectively.

FUTURE SCOPE

1) The suspension system material can be changed to advanced materials.
2) The Suspension can be analysed for transient or shock loads.
3) The Suspension members can be optimized for the structural loads.
4) The topology optimization can be carried out for still better strength.

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