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Design Optimization of McPherson Strut Suspension System using Transient Analysis and Numerical Simulation

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Abstract: In past few decades, automobiles have evolved in all the subsystems in order to increase the efficiency, passenger comfort and safety. Suspensions are the integral part of any vehicle type. Suspensions had a thorough research and development of various advanced models to reduce the vibrations caused by the vertical acceleration of the vehicle body. Yet, the problem with maximum vibration elimination stays constant. This results into damage of vehicle body, instability of ride and increased probability of accidents. Among various suspension configurations available today in market, McPherson has been proved more efficient and safe over the years. However, it has much scope of improvement if contributed with thorough analysis and optimization.

This study based on quarter car model focuses on the design optimization of the McPherson strut suspension system using a combination of transient analysis and numerical techniques. Transient analysis allows for the simulation of dynamic responses under various driving conditions, providing insights into the system's behavior over time. Numerical optimization methods enable the exploration of a vast design space to identify the optimal configuration that meets specified performance criteria.

The findings of this research can benefit automotive engineers and designers in developing suspension systems that strike a balance between comfort, performance, and efficiency, thereby contributing to the advancement of vehicle dynamics and driving experience.

Keywords **– Quarter Car Model, ANSYS, MATLAB Simulink, Stiffness, Damping, Static, Transient, Vertical acceleration.**

I. Introduction

Suspension is the definition given to the system of springs, shock absorbers and linkages that joints a vehicle to its wheels and allows relative motion between the two. Suspension systems allow dual purpose contributing to the vehicle's road holding/handling and braking for good active safety and driving pleasure, and keeping vehicle occupants comfortable and reasonably well isolated from road noise, bumps, and vibrations, etc. If a road surface were perfectly flat and had no irregularities in it, then suspensions would not be required. But roads are far from flat, even freshly paved motorways/highways have subtle imperfections that can interact with the wheels. It's these imperfections that apply forces to the wheels and suspension components and causes handling imbalances in compromised set ups. The McPherson strut is a type of car Suspension system commonly used in many modern motor vehicles. This includes both front and rear suspensions, but usually located at the front of the car. The McPherson strut normally also has a steering arm built into the lower inner portion. This assembly is extremely simple and can be pre manufactured into a unit at the assembly line. By removing the upper control arm, it allows for more width in the engine bay, aiding in any maintenance work or engine design requirements. McPherson suspension system provides more space in the engine compartments. McPherson suspension system is use as the front suspension in the car and suspension system generally used for better cornering and also for the comfortable passenger ride. McPherson suspension system contains upper mounting point, shock absorber, spring, spring leg and lower control arm. It contains only one lower arm so it also known as single wishbone system.

The mathematical models are able to convert the system into mathematical equations so the equations will be solved and some rigid conclusions can be drawn for optimized performance. These models provide flexibility to optimize various parameters which include spring stiffness, damping, sprung and un-sprung masses and road profile input.

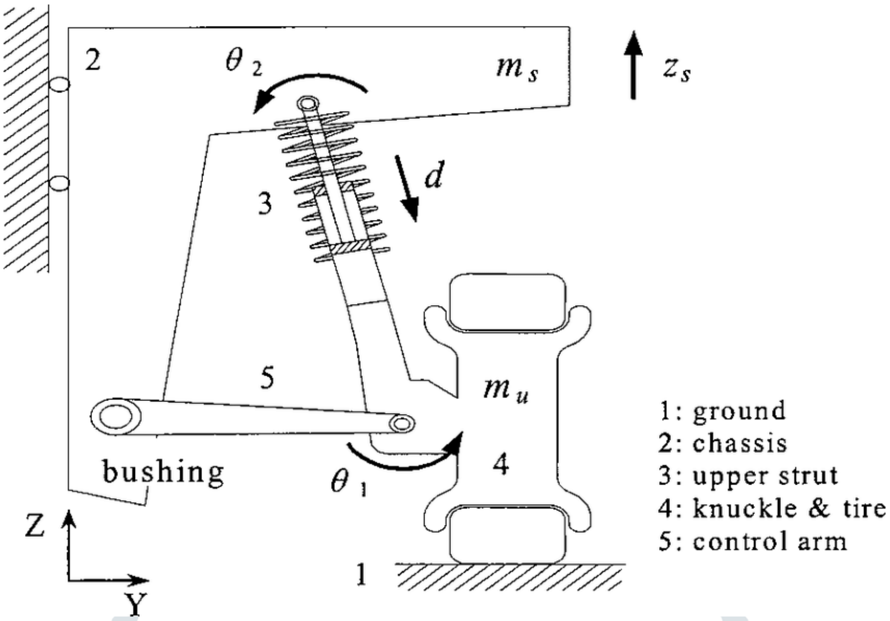


Fig. 01: Schematic of McPherson Strut Suspension System.

II. LITERATURE REVIEW

1. *A. Purushotham, Comparative Simulation studies on McPherson Suspension System, International Journal of Modern Engineering Research (IJMER), Vol.3, Issue.3, May June.(2013), Page 1377-1381*

Purushotham described that most of automobiles these days are using two suspension systems namely: double wishbone suspension system and McPherson suspension due to their good dynamic performance and higher passenger comfort. The McPherson strut setup is still being used on high performance cars such as the Porsche 911, several Mercedes-Benz models and lower BMW models due to its light weight, design simplicity and low manufacturing cost. This paper proposes a systematic and comprehensive development of a two-dimensional mathematical model of a McPherson suspension. The model considers not only the vertical motion of the chassis (sprung mass) but also rotation and translation for unsprung mass (wheel assembly). Furthermore, this model includes wheel mass and its moment of inertia about the longitudinal axis. The paper offers an implementation of the model using MATLAB Simulink, whose dynamics have been validated against a realistic two dimensional model developed with the ANSYS software. In this paper author concluded that the McPherson suspension system has been modeled after studying dynamic equations to study vibration characteristics of sprung mass of the automobile system with the inclusion of various design parameters such as stiffness, damping, masses, moment of inertia, etc. The commercial simulation software Simulink is used by author to implement dynamic equations to attain the acceleration and the displacement of the chassis of the automobile during the period in which the vehicle passes through various road conditions. Due to the complexity involved in the mathematical expressions and executing them into the Simulink software, the model has been simplified with a two dimensional approach. The ANSYS software is used to implement a simplified two dimensional practical model of McPherson suspension. The results obtained from ANSYS model are compared with the mathematical model implemented on Simulink. It is observed that the displacement and acceleration of the chassis of the automobile obtained in ANSYS are nearer to the values of mathematical model. With these developed models, the influence of suspension system parameters can be studied on the performance of passenger comfort.

2. *S. Pathmasharma, J. K. Suresh, P. Viswanathan and R. Subramanian , Analysis of Passenger Car Suspension System Using Adams, International Journal of Science, Engineering and Technology Research (IJSETR) Volume 2, Issue 5, May (2013), Page 1186-1193*

S. Pathmasharma described that in the past few decades rapid technological growth in the area of automobile engineering have witnessed. One way of improving the market share is to provide a vehicle with maximum comfort which is achieved by modifying the suspension system. Passenger car commonly use coil suspension system in their vehicles to absorb road shocks and provide comfort to passenger. Nonlinear springs are most commonly used in vehicle suspension system. Much research work has been carried out on coil spring with the objective of getting optimized designs to improved passenger comfort. In this paper author has discussed about the existing suspension system and improved design of suspension system. Author has done modeling using UG and dynamic analysis done in the Automatic dynamic of Mechanical System (ADAMS) in which analysis of suspension is carried out by author. In the paper main components like mounting head, track width and other parameters changed by author. Finally, author concluded that modification parameter of the suspension system improves the performance of the suspension system

III. VEHICLE SPECIFICATIONS:

Vehicle Name	: MARUTI SUZUKI SWIFT
Kerb Weight	: 960 kg
Gross Weight	: 1405 kg
Tire	: 165/80 R14 85T
Max. Speed Limit	: 190 kmph
Tire weight (approx.)	: 10.5 kg
Max. Load	: 515 kg
Tire aspect ratio	: 80
Load Index Rating	: 85

According to weight distribution, 60% on front side and 40% on rear side (advised condition).
Therefore, Mass acting on front suspensions: 843 kg and on Rear Suspensions: 562 kg.
Sprung Mass of 422 kg on each front suspension. Considering each wheel unsprung mass: 87.5 kg.

IV. QUARTER CAR SUSPENSION MODEL:

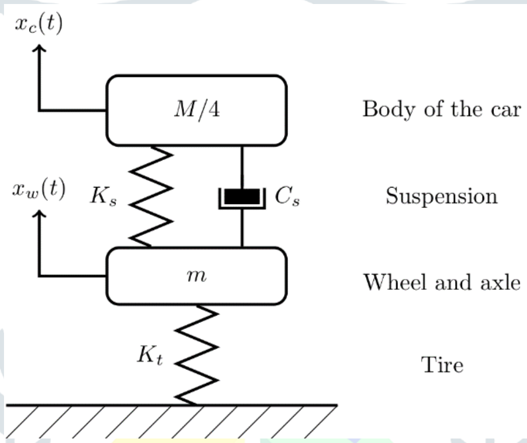


Fig. 02: Quarter Car Suspension Model

Above fig. shows a simplified model with 2 DOF. It consists of sprung mass supported by a primary suspension, which is in turn connected to unsprung mass. As shown above, the Tire is represented as spring with equivalent stiffness, although a damper is most of the times included to represent small amount of damping inherent to the visco-elastic nature of the Tire. The abrupt road profile can also be represented with positive and negative amplitude.

Notations:
Ms = Sprung Mass (kg)
Mu = Unsprung Mass (kg)
Cs = Damping Coefficient (N-s/m)
Ks = Spring Stiffness (N/m)
Kt = Tire equivalent stiffness (N/m)
Xs = Sprung Mass body displacement. (m)
Xu = Unsprung Mass body displacement (m)
Xr = Vertical Travel (m)

<u>GOVERNING EQUATIONS:</u>

<u>Car Body Bounce:</u>

Ms*Xs” + Cs (Xs’- Xu’) + Ks (Xs – Xu) = 0 ....(1)

<u>Wheel Bounce:</u>

Mu*Xu” + Cs (Xu’ - Xs’) + Kt (Xr’ - Xu’) + Ks (Xu – Xs) = 0 ....(2)

According to Newton’s II Law of Motion,

Σ F = m*a

<u>For Mass ‘Ms’:</u>

Σ F = -Fd – Fs_2 = Ms*Xs”

= - Cs (Xs’ - Xu’) - Ks (Xs - Xu) = Ms*Xs”

Rearranging,

Ms*Xs” + Cs (Xs’ - Xu’) + Ks (Xs – Xu) = 0 ....(3)

For Mass ‘Mu’:

$\Sigma F = - Fd + Fs_2 - Fs_1 =$ Mu*Xu”

Cs (Xs’ - Xu’) + Ks (Xs – Xu) - Kt (Xu – Xr) = Mu*Xu”

Rearranging,

Mu*Xu” - Kt (Xr- Xu) - Ks (Xs – Xu) - Cs (Xs’ - Xu’) = 0 ....(4)

Now, from equations (3) and (4), we have vertical acceleration equations:

Xs” = 1/Ms * Cs (Xu’ - Xs’) + Ks (Xu – Xs)

Xu” = 1/Mu * Kt (Xr – Xu) + Ks (Xs – Xu) + Cs (Xs’ - Xu’)

V. MATLAB SIMULINK MODEL:

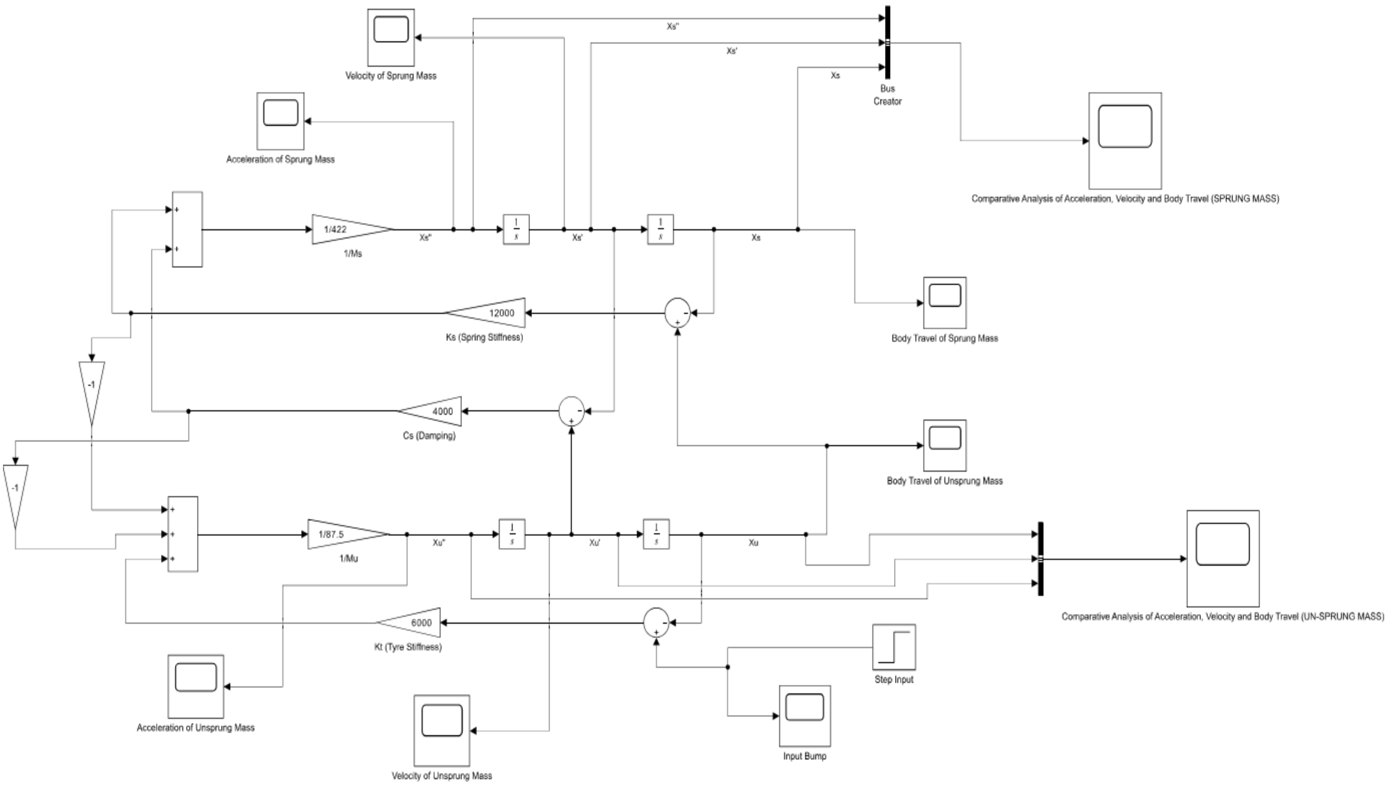


Fig. 03: MATLAB SIMULINK Model for Quarter Car Suspension System.

SIMULINK Results for different conditions can be achieved by:

1. Change in Sprung mass.
2. Change in stiffness of suspension spring.
3. Change in damping coefficient.
4. Change in un-sprung mass.
5. Change in Tire stiffness.
6. Road Profile Input in terms of mathematical equation.

MATLAB RESULTS:

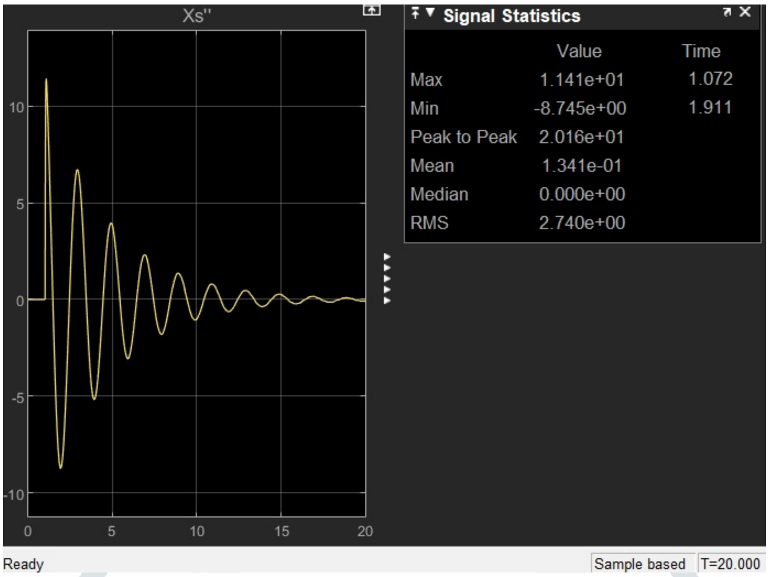


Fig. 04: Sprung Mass Acceleration VS Time (20 seconds)

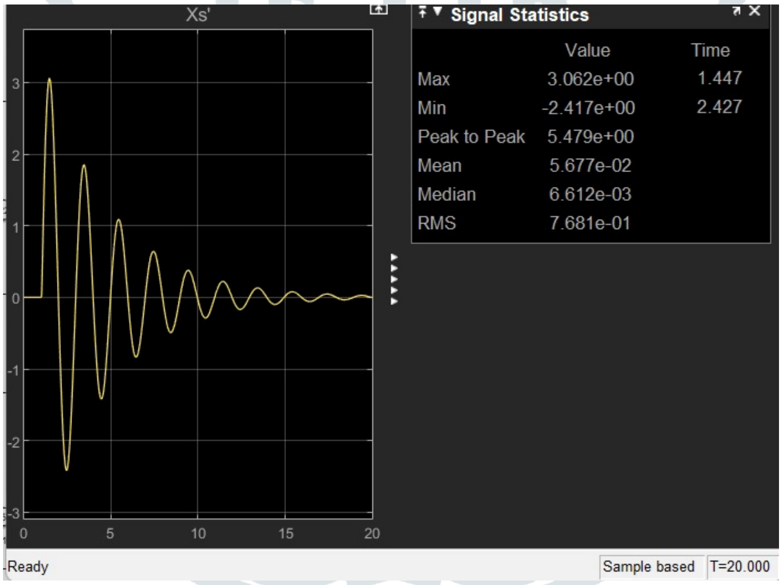


Fig. 04: Sprung Mass Velocity VS Time (20 seconds)

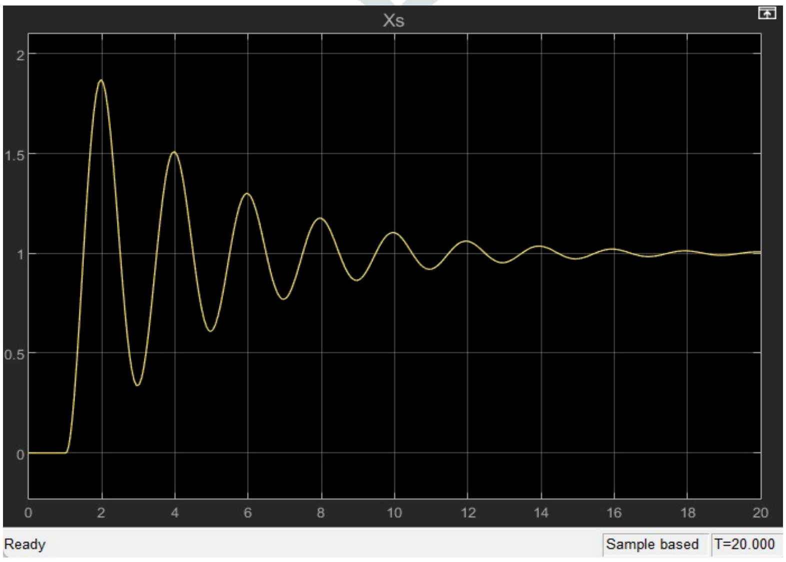


Fig. 06: Sprung Mass Body Travel VS Time (20 seconds)

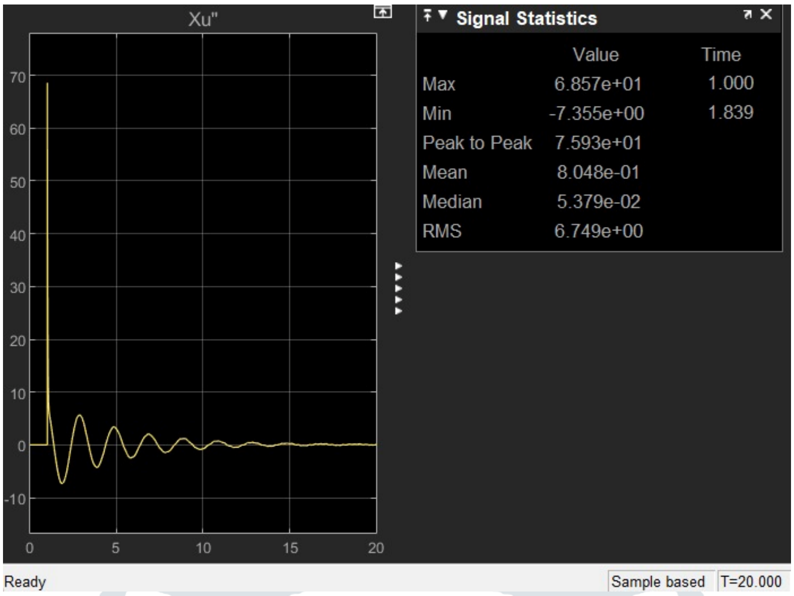


Fig. 07: Unsprung Mass Acceleration VS Time (20 seconds)

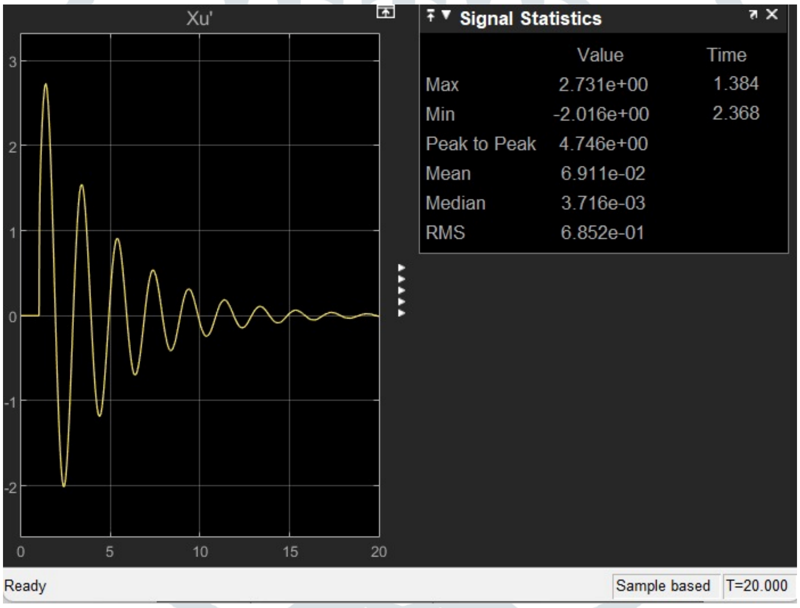


Fig. 08: Unsprung Mass Velocity VS Time (20 seconds)

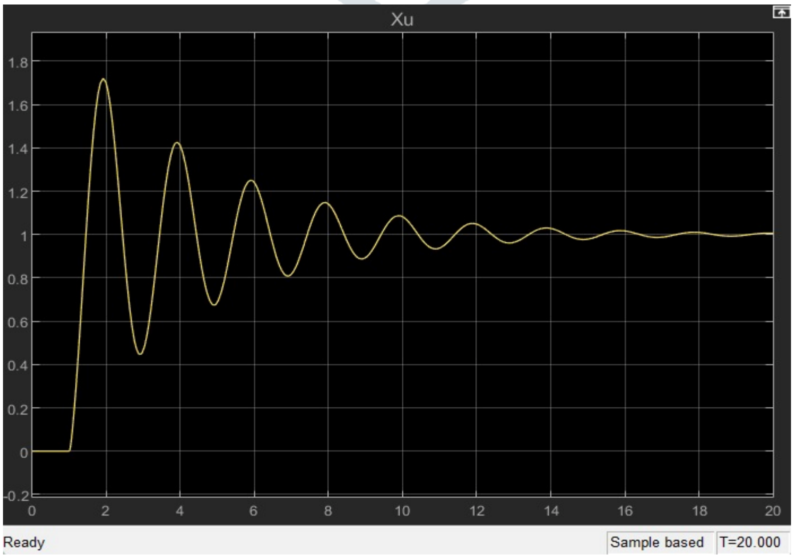


Fig. 09: Unsprung Mass Body Travel VS Time (20 seconds)

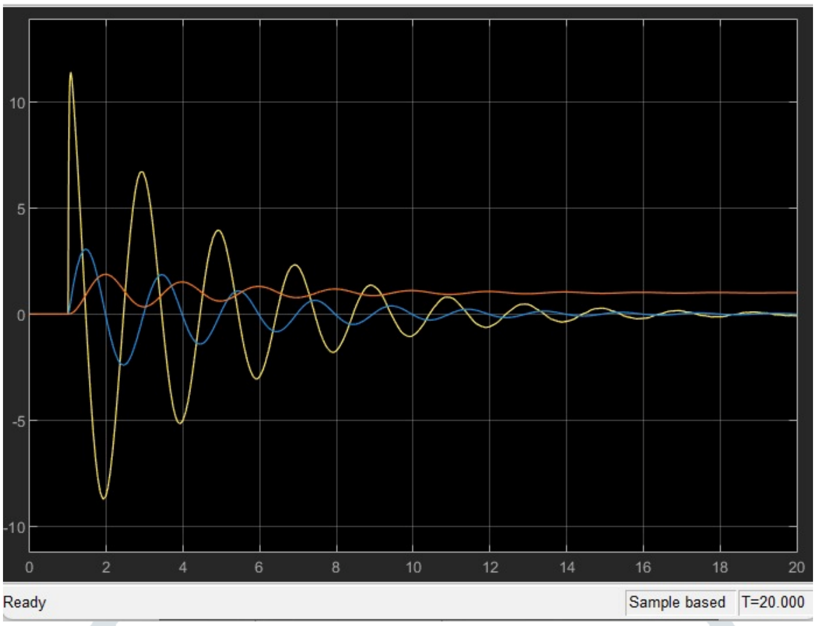


Fig. 10: Comparative Analysis (SPRUNG MASS)

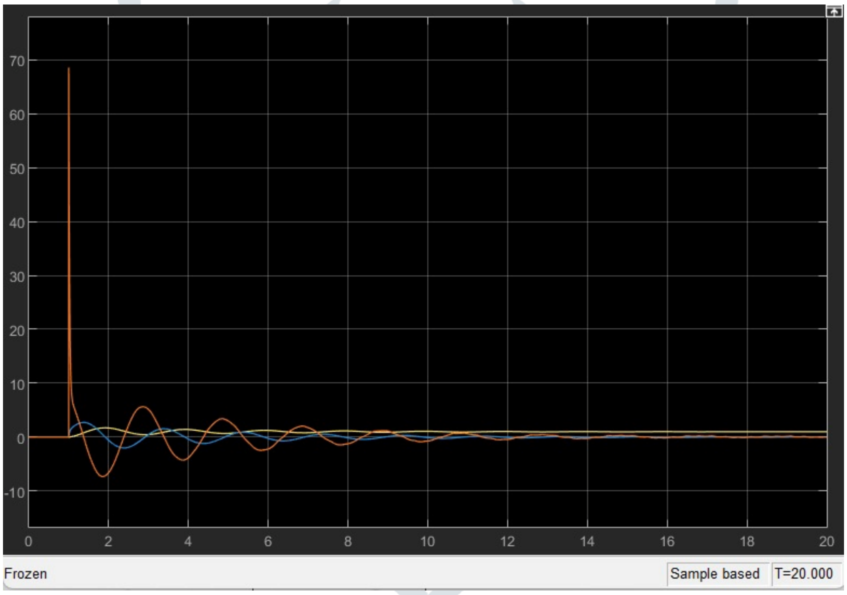


Fig. 11: Comparative Analysis (UN-SPRUNG MASS)

VI. MATLAB RESULTS CONCLUSION:

Above graphs were plotted considering the actual suspension model with a step input. The comparative analysis plots showcase the gradual reduction of vertical acceleration (Z-Axis Body Travel) due to shock absorption in spring and dampers. The plots were recorded for a time period of 20 seconds. Behaviour of real-time suspension model for given road profile for varied parameters can be recorded and analysed using the designed MATLAB Simulink Model for maximum optimized design. Technical specifications required for maximum elimination of vibrations can be studied through this model. This would help to decide spring material, stiffness, damping coefficient in order to increase ride comfort of passenger taking weight of complete vehicle body in consideration.

VII. ANSYS

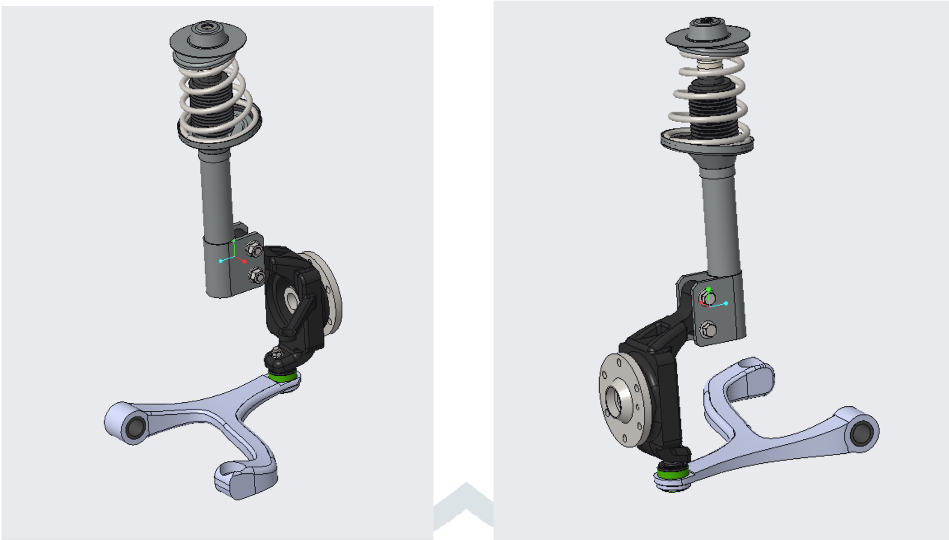


Fig 12: Suspension assembly

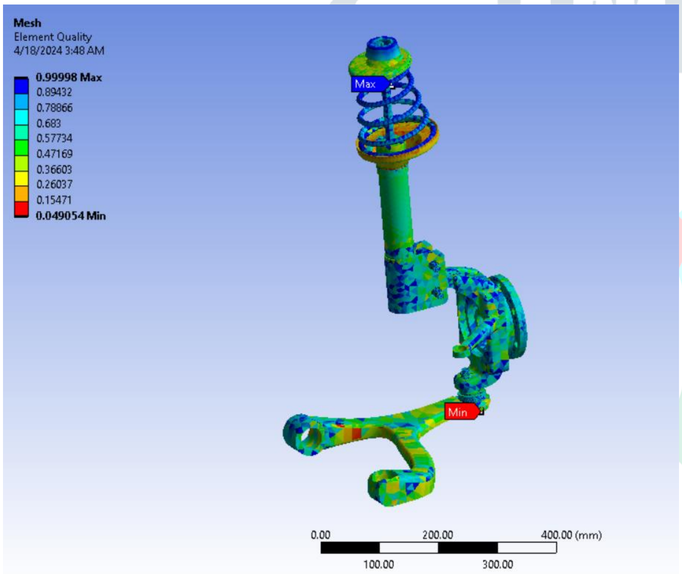


Fig 13: Element Quality

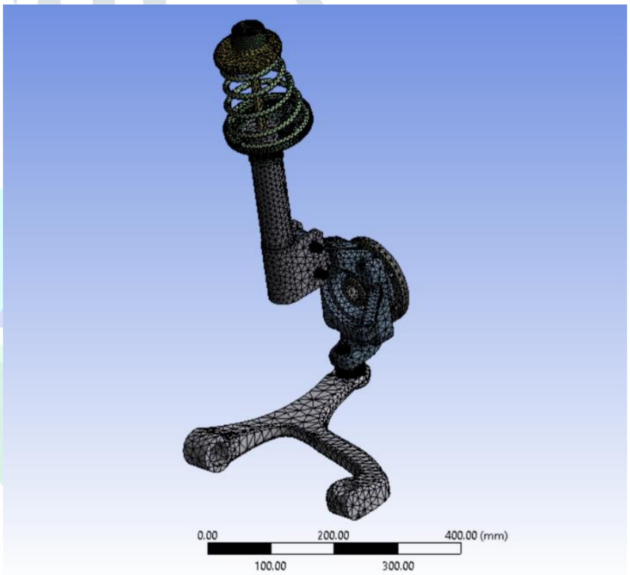


Fig 14: Mesh

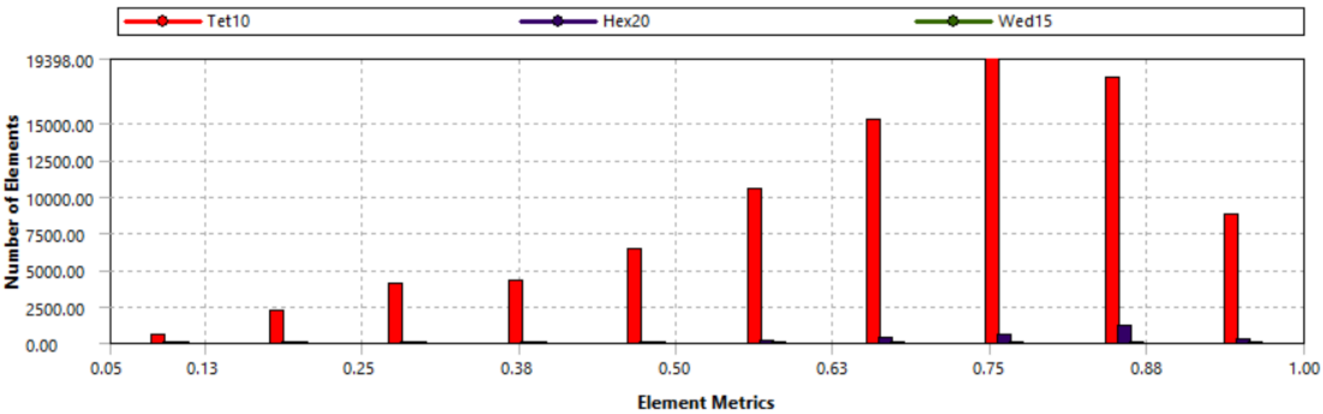


Fig 15: Mesh Matrix

LOAD APPLIED:

Impact load faced by Tire due to road profile.
Force at A: 5000 N

Vehicle weight
Force at B: 4500 N

The above mentioned forces are been considered for static structural analysis of the system.

For transient structural analysis the direction and location of force applied remains same as shown in Fig.16. The magnitude of force-A change with time.

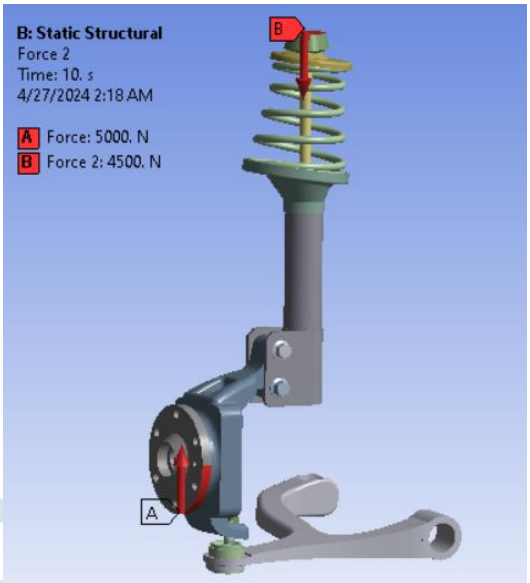


Fig 16: Direction of forces

RESULTS:

A) STATIC STRUCTURAL ANALYSIS

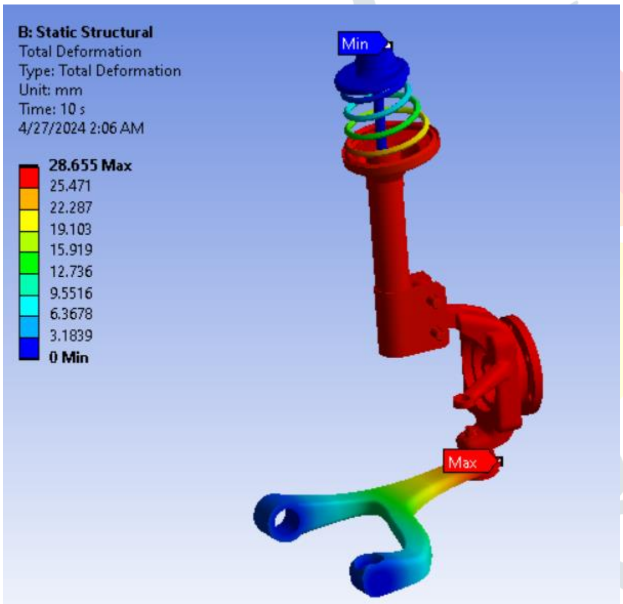


Fig 17: Total Deformation

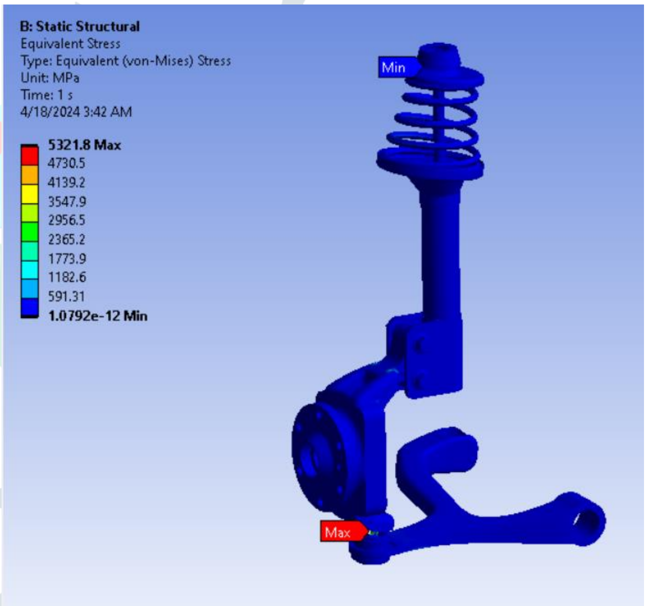


Fig 18: Equivalent stress

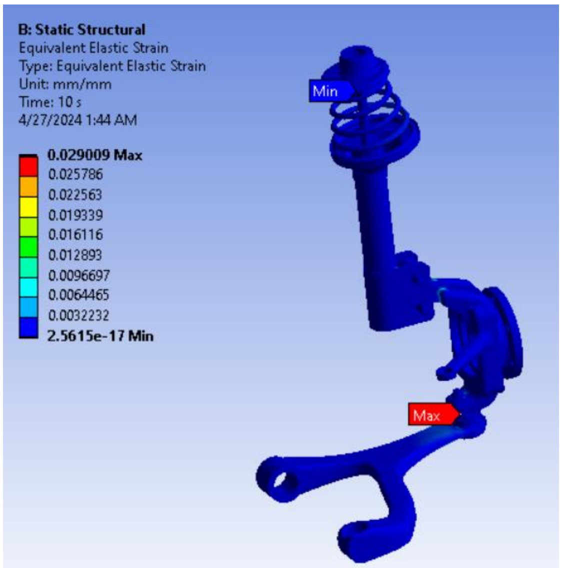


Fig 19: Equivalent Strain

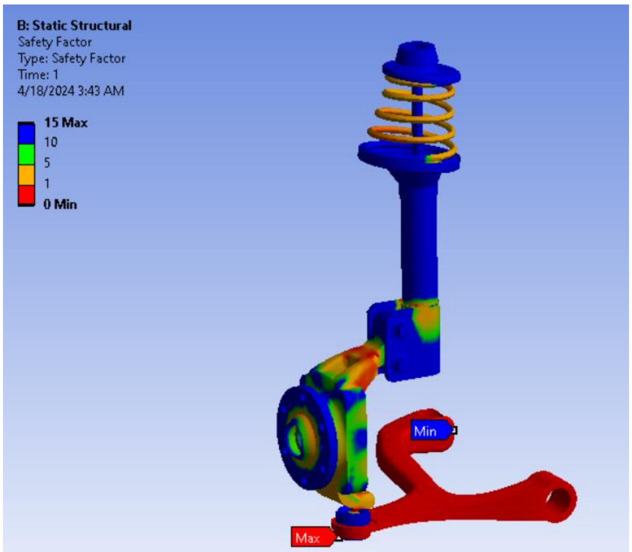
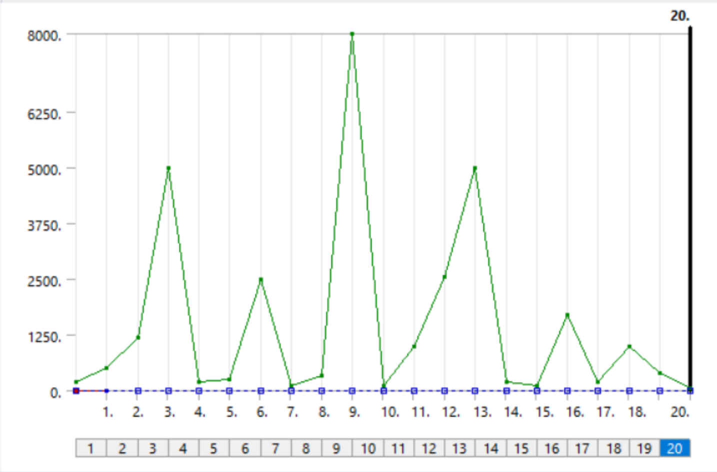


Fig 20: Factor of Safety

B) TRANSIENT STRUCTURAL ANALYSIS

Time stamp: 20 seconds

Following are the load Inputs for Transient analysis for force A as shown in Fig.16



	Steps	Time [s]	X [N]	Y [N]	Z [N]
1	1	0.	0.	200.	= 0.
2	1	1.	0.	500.	0.
3	2	2.	= 0.	1200.	= 0.
4	3	3.	= 0.	5000.	= 0.
5	4	4.	= 0.	200.	= 0.
6	5	5.	= 0.	250.	= 0.
7	6	6.	= 0.	2500.	= 0.
8	7	7.	= 0.	100.	= 0.
9	8	8.	= 0.	350.	= 0.
10	9	9.	= 0.	8000.	= 0.
11	10	10.	= 0.	100.	= 0.
12	11	11.	= 0.	1000.	= 0.
13	12	12.	= 0.	2560.	= 0.
14	13	13.	= 0.	5000.	= 0.
15	14	14.	= 0.	200.	= 0.
16	15	15.	= 0.	100.	= 0.
17	16	16.	= 0.	1700.	= 0.
18	17	17.	= 0.	200.	= 0.
19	18	18.	= 0.	1000.	= 0.
20	19	19.	= 0.	400.	= 0.
21	20	20.	= 0.	50.	= 0.

Fig 21: Load inputs

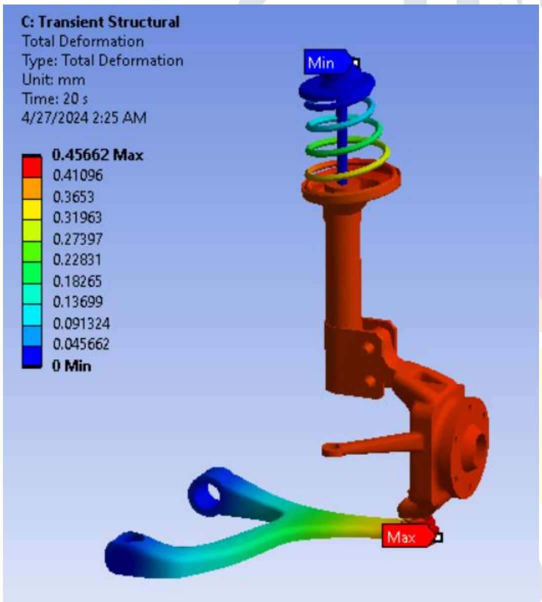


Fig 22: Total Deformation

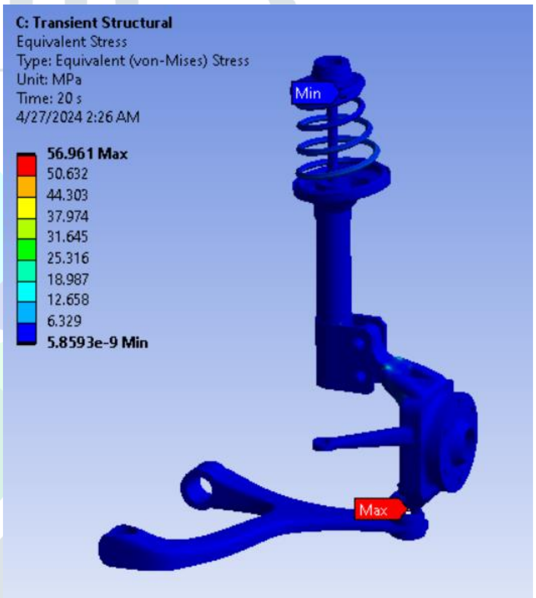


Fig 23: Equivalent Strain

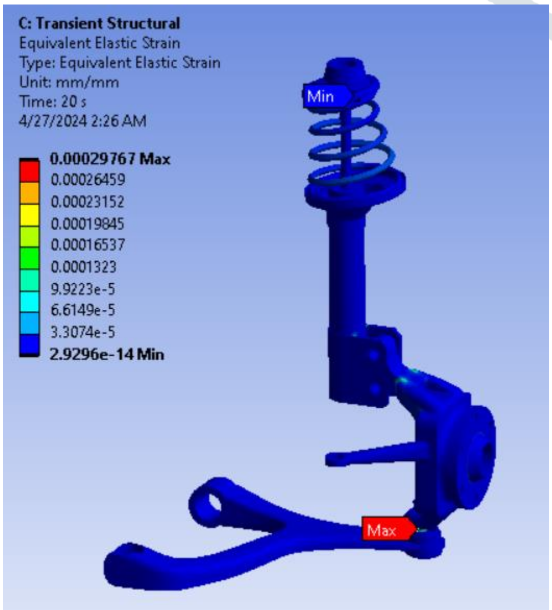


Fig 24: Equivalent Strain

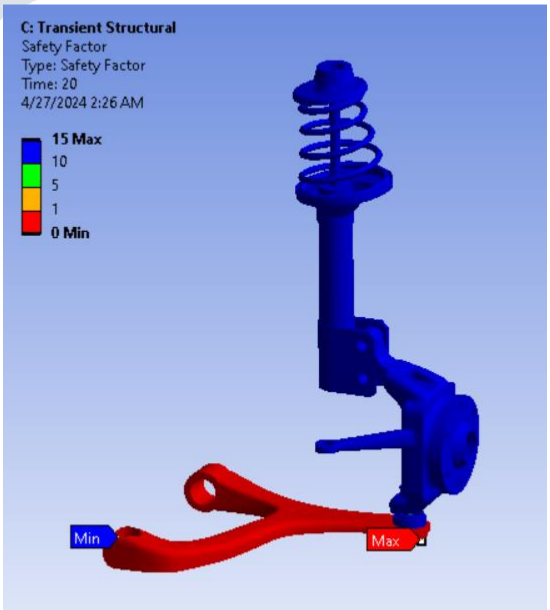


Fig 25: Factor of Safety

VIII. DESIGN MODIFICATIONS

After analyzing the results of existing system, following geometric design modifications are made to enhance system performance. These changes aim to improve efficiency, ride comfort, and handling.

An additional damper assembly has been integrated into the system, prompting modifications to the lower control arm. The modified control arm has a mount for the additional damper as shown in fig. 26 and fig 27.In the modified design, the location previously occupied by the main damper now accommodates a smaller damper.

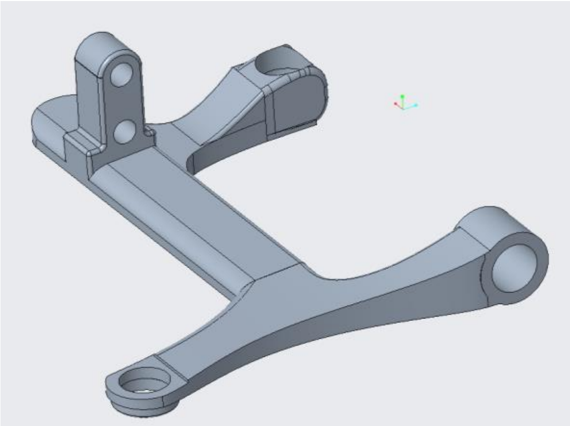


Fig 26: Modified control arm

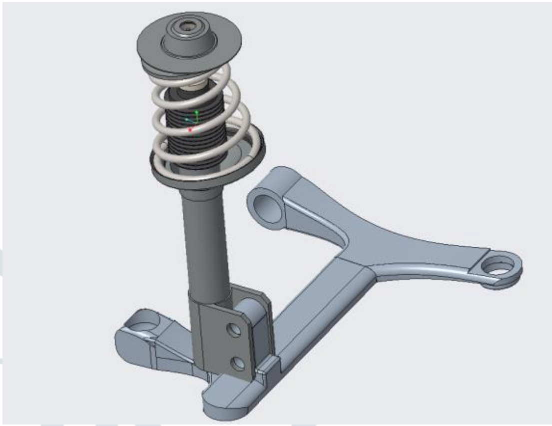


Fig 27: Additional damper installed

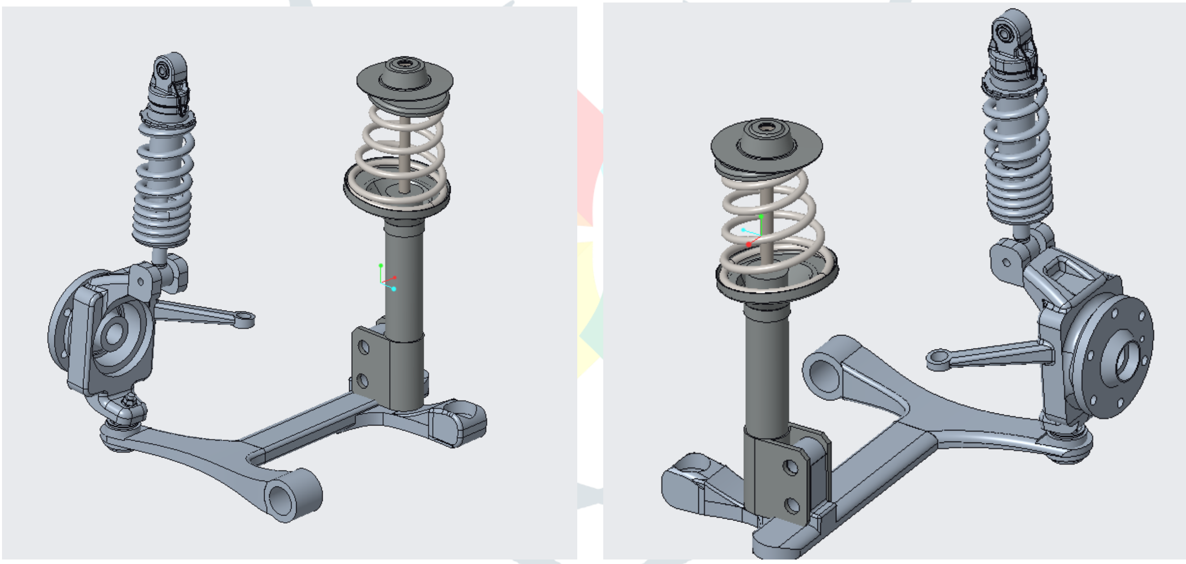


Fig 28: Modified assembly of McPherson Suspension

LOAD APPLIED:

Impact load faced by Tire due to road profile.
Force at A: 5000 N

Vehicle weight
Force at B: 2500 N
Force at C: 2000 N

The above mentioned forces are been considered for static structural analysis of the system.

For transient structural analysis the direction and location of force applied remains same as shown in Fig.16. The magnitude of force-A change with time.

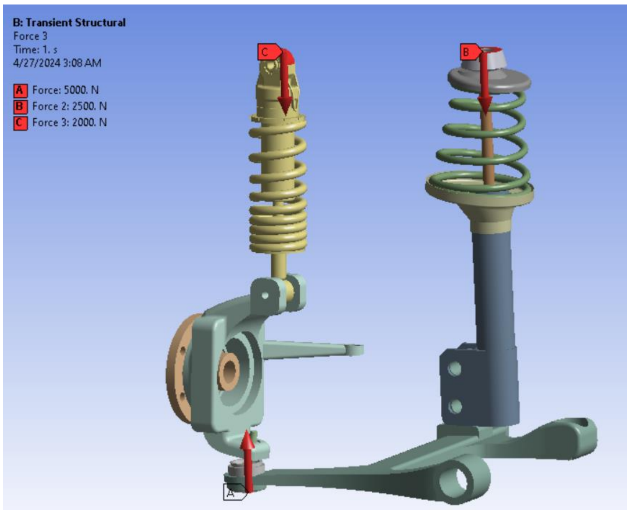


Fig 29: Direction of Forces

A) STATIC STRUCTURAL ANALYSIS: MODIFIED GEOMETRY

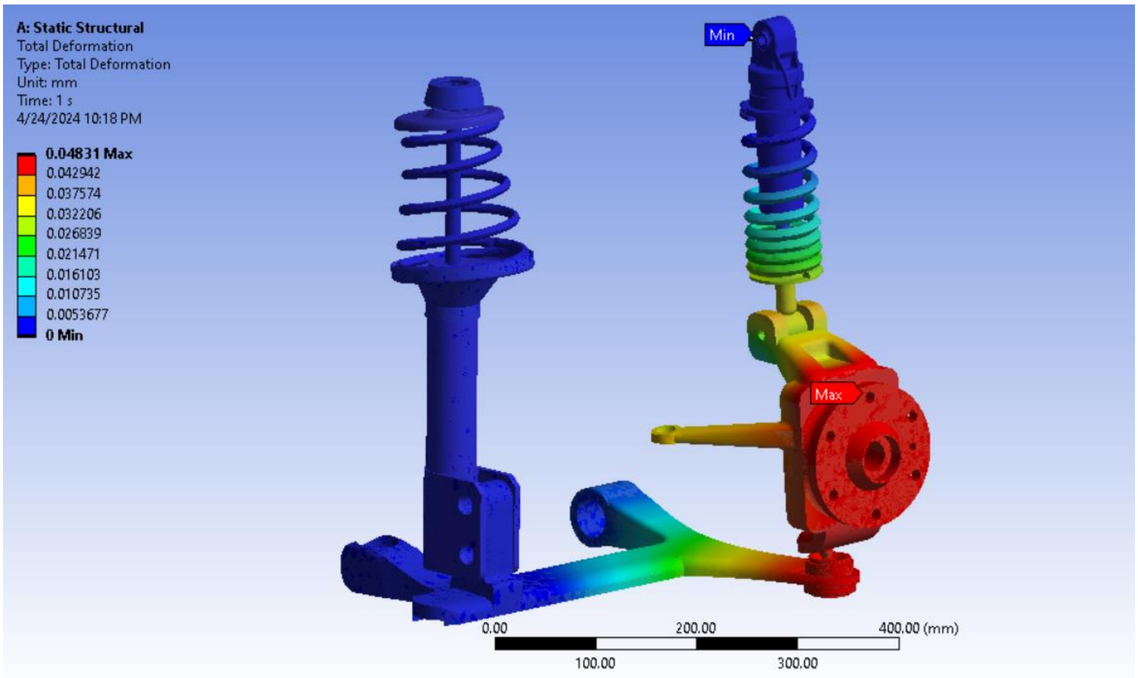


Fig 30: Total deformation

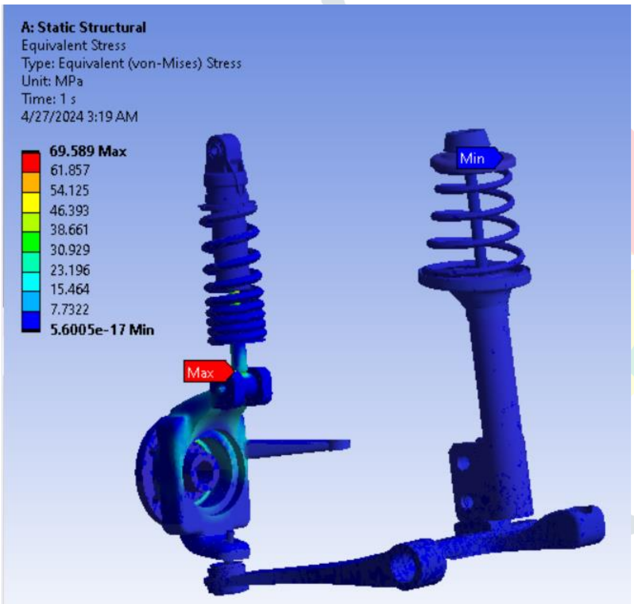


Fig 31: Equivalent Stress

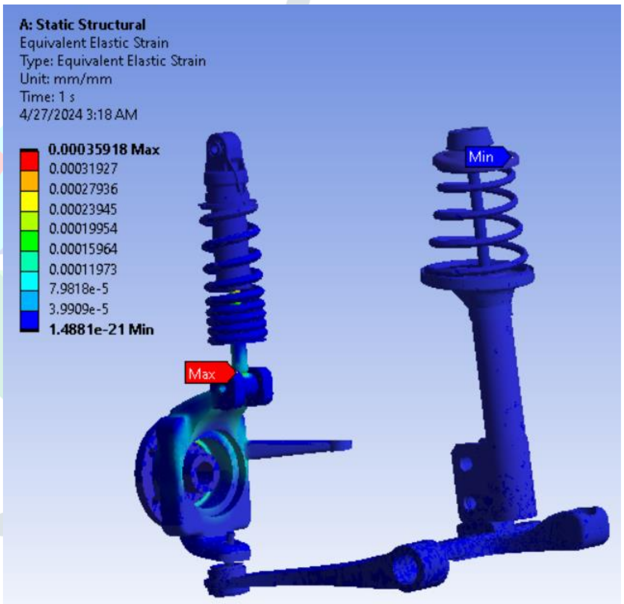


Fig 32: Equivalent Strain

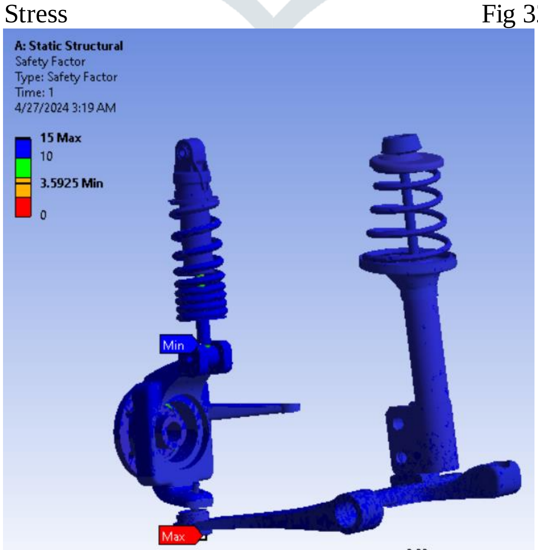


Fig 33: Factor of Safety

B) TRANSIENT STRUCTURAL ANALYSIS: MODIFIED GEOMETRY
Time stamp: 20 sec
The load input for transient are similar to the inputs of previous geometry, Refer fig.21.

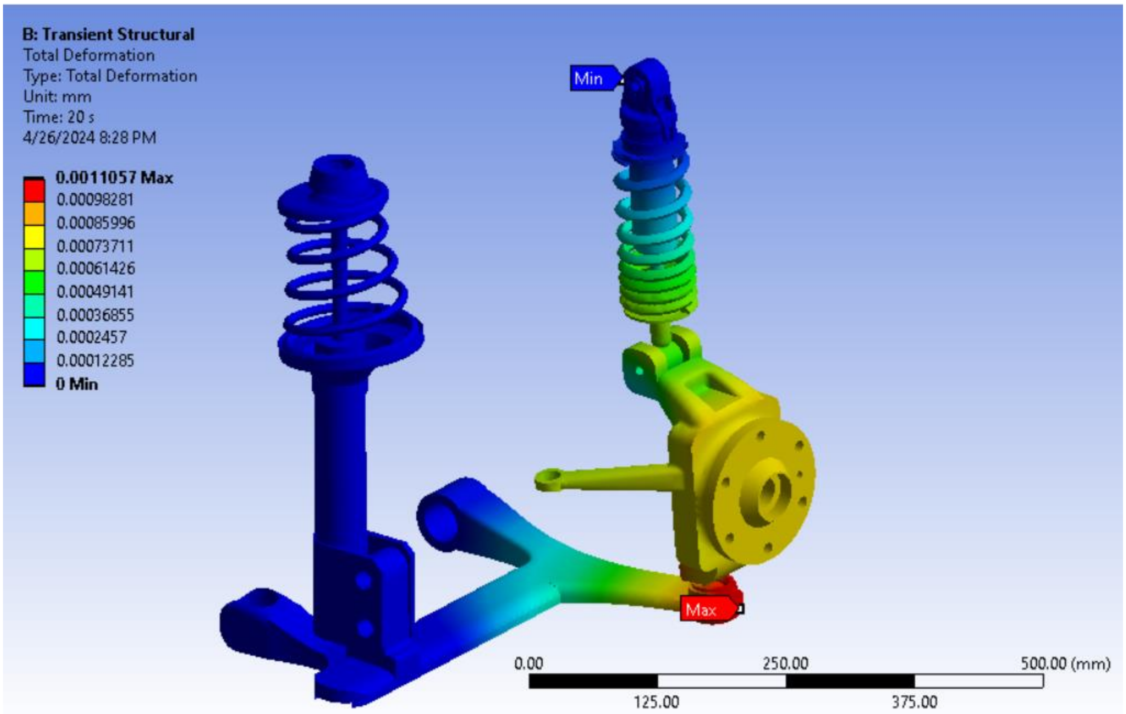


Fig 34: Total Deformation

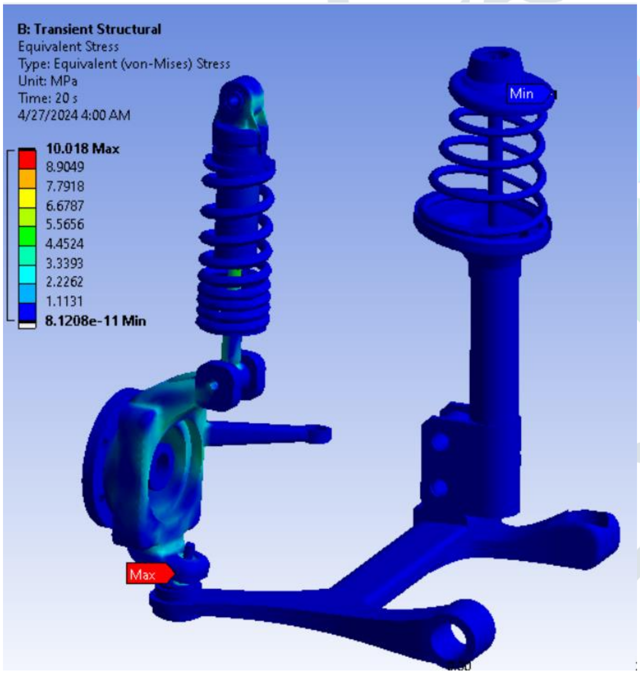


Fig 35: Equivalent stress

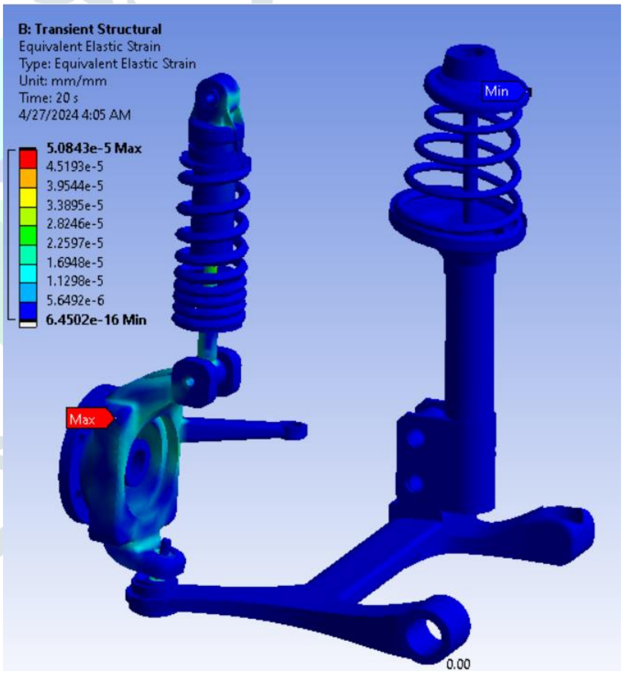


Fig 36: Equivalent strain

PARAMETERS		ORIGINAL GEOMETRY		MODIFIED GEOMETRY	
		STATIC	TRANSIENT	STATIC	TRANSIENT
Total Deflection (mm)	Maximum	28.655	0.45662	0.04831	0.0011
	Minimum	0	0	0	0
Equivalent Stress (MPa)	Maximum	5321.8	56.961	69.586	10.018
	Minimum	1.0792×10^{-12}	5.589×10^{-9}	5.6×10^{-17}	8.108×10^{-11}
Equivalent Strain (mm/mm)	Maximum	29.009×10^{-3}	29.767×10^{-5}	35.918×10^{-5}	5.0843×10^{-5}
	Minimum	2.5615×10^{-17}	2.929×10^{-14}	1.488×10^{-21}	6.4502×10^{-16}

IX. CONCLUSION:

The optimized design proposed for maximum strength and vibration absorption is presented in this study. Comparative analysis of existing geometries of McPherson Strut suspension and modified geometry concludes that 82% reduction was observed in Transient Equivalent Stress analysis. The Transient Total Deflection was successfully reduced by 97%. In addition, Equivalent strain was reduced by 85.84%.

These results prove that the proposed design has comparatively higher strength, less deflection and less equivalent strain which allows more safety and ride comfort with long component life.
The damping properties of the suspension damper need careful calibration to align with other dynamics parameters of the vehicle, including sprung mass, un-sprung mass, Tire stiffness, suspension stiffness, and road input. This ensures that when the vehicle encounters a bump, it doesn't experience excessive vibration for an uncomfortably long duration, thus avoiding passenger discomfort and potential vehicle failure. To simplify analysis and focus on understanding the interplay of these parameters, we hold road input constant and assume the vehicle maintains a steady velocity. The proposed MATLAB Simulink model allows to optimize the design as per available materials and conditions. The vibrations can be analysed for abrupt road profile and can be reduced by increasing the spring stiffness and damping coefficient.

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