COMPARATIVE STUDY AND ANALYSIS OF AUTOMOTIVE SUSPENSIONS SYSTEM

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Abstract: The purpose of this paper is to compare the Double Wishbone Leaf spring suspension system and the Macpherson Suspension system. On these suspension systems, ANSYS is used to achieve the goal of static structural analysis. Static loads are applied to the suspension system. The analyses, including structural analysis with static loading, to analyze the deflection, stress, and strain of the suspension systems, thorough which a complete comparative study can be accomplished.
A MATLAB Simulink block diagram was developed for the analysis. The algorithm developed here can be used to analyze an automobile's quarter car model using 2DOF, saving money on the circuit and rig testing.
Vibrations induced by road disturbances are not conveyed to the driver by automotive suspension systems. To simulate this a quarter vehicle model with two degrees of freedom (DOF) is produced. The impact of a speed bump as a step function is examined for the overshoot and settling time of sprung mass. A sine function is also used to replicate the characteristics of a bumpy road surface.

Keywords: Automotive, suspension, Quarter-car, Passive Suspension, MATLAB Simulink, Ansys Suspension

1. Introduction

When a car travels over rough roads, the vehicle will move up and down in response to the bumps in the road. When a high-speed vehicle collides with a road bump, it generates extremely high shock loads, which causes car components to lose control of their function and cause damage. To avoid this, car designers devised a suspension system for the vehicle. The suspension is the system of spring and shock absorber by which a vehicle is supported on its wheel.
The vehicle's suspension system offers a smooth ride and protects passengers from road shocks. In addition, the suspension system allows and maintains contact between the road surface and the wheels, ensuring steering stability, traction, and good handling. The suspension system on the car is not required if the route is flat and there are no abnormalities on the road. However, roads are not without bumps, potholes, and other flaws. These can interact with the vehicle wheels.

2. The function of the suspension system

As previously said, it is commonly considered that the main role of a suspension system is to absorb road roughness; nevertheless, due to varying operating conditions, the suspension of a vehicle must satisfy numerous requirements with somewhat contradictory goals. Because the suspension system connects the vehicle's body to the ground, all forces and moments between the two pass via it. As a result, the suspension system has a direct impact on a vehicle's dynamic behavior. The functions of a suspension system are frequently studied by automotive engineers using three key principles.

3. Design of suspension system

3.1 Leaf spring suspension system

The leaf spring is made up of many metal plates, also known as leaves, that are placed on top of each other in decreasing order of length. The camber is the curvature of the leaves that gives the leaf spring its semi-elliptical shape.

The master leaf, positioned on top, has its ends rolled to be attached to the body of the vehicle. These coiled ends of the master leaf are referred to as the eyes. The second master leaf is positioned just below the master leaf to support the master leaf, while the remaining leaves are known as graduated leaves [1]. The design of the leaf spring in solid works is shown below.

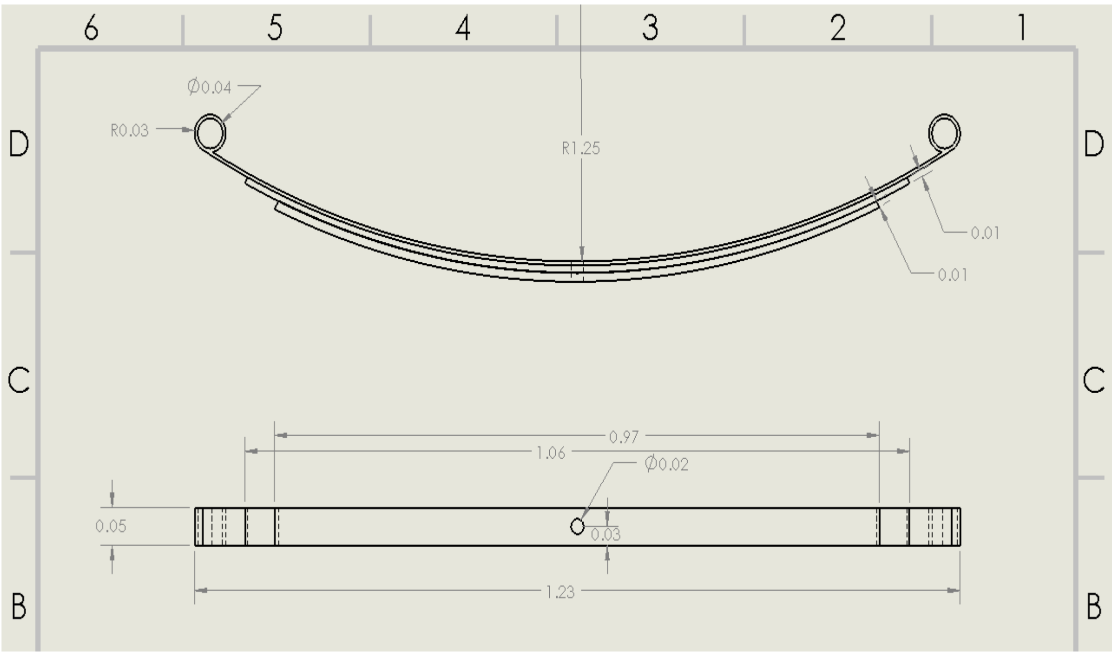


Figure 3.1 Leaf spring schematic diagram (all dimensions are in meters m).

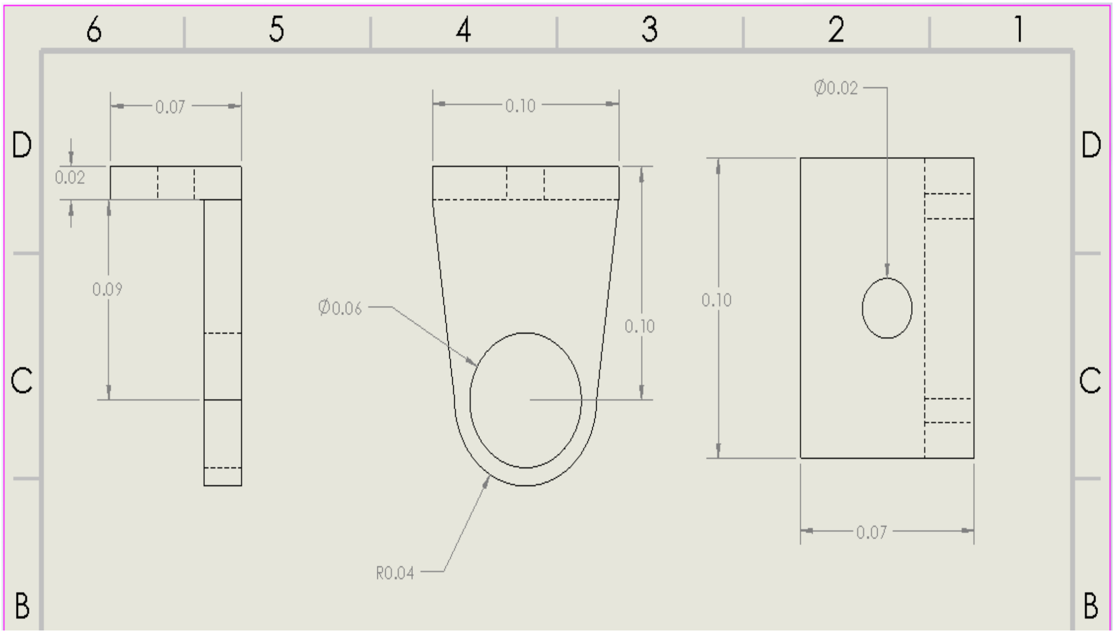


Figure 3.2: Schematic diagram of a central clamp (all dimensions are in meters m).

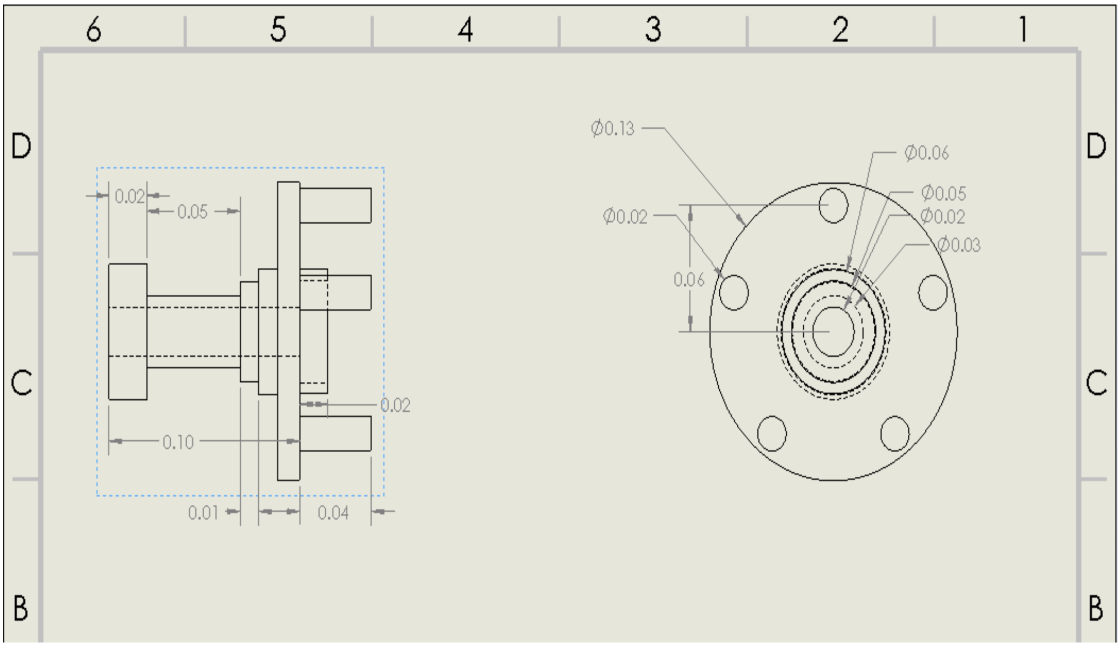


Figure 3.3: Schematic diagram of a wheel hub (all dimensions are in meters m).

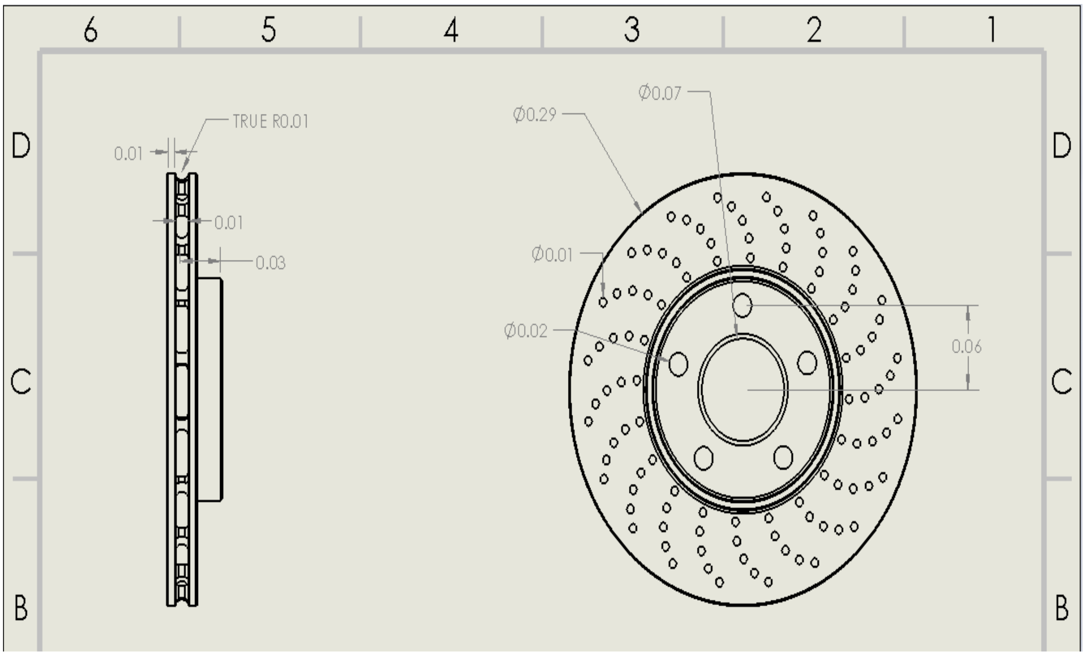


Figure 3.4: Schematic diagram of a disk brake (all dimensions are in meters m).

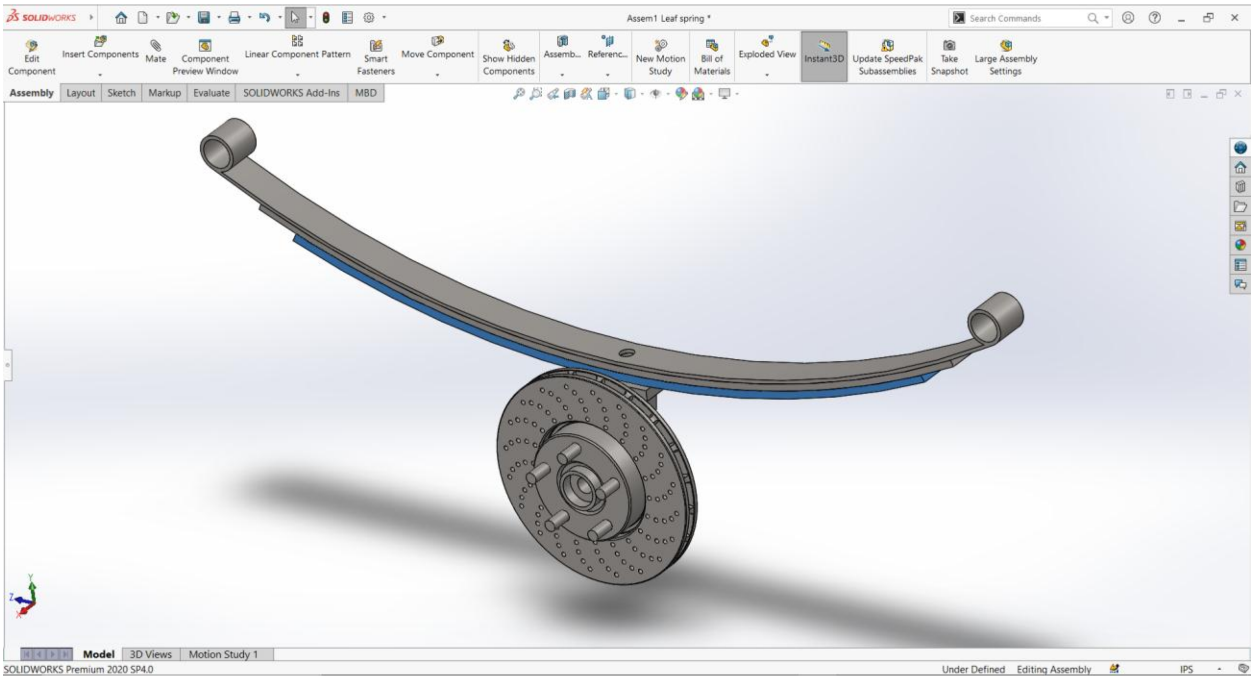


Figure 3.5: Leaf spring suspension assembly

The above-shown model's mass is equal to the sum of the masses of all the parts used in the assembly.

Mass of the assembly = 11.6951 + 1.9321 + 9.158 = 22.7852 kg

In addition to the mass of the assembly, we also must consider the mass of the other parts such as disk pads, tires, and wheels which are not modeled.

Hence the total mass = 22.7852 + 1.25 +9.07 = 33.1052 kg

3.2 Macpherson strut suspension system

The MacPherson strut is a form of vehicle suspension system which contains a single swing arm connected to the wheel. The spring and damper are connected to this swing arm and support the vertical load of the vehicle. Earle S.MacPherson, an American automotive engineer, devised and perfected the design, which is now commonly utilized in the front suspension of current vehicles [2].

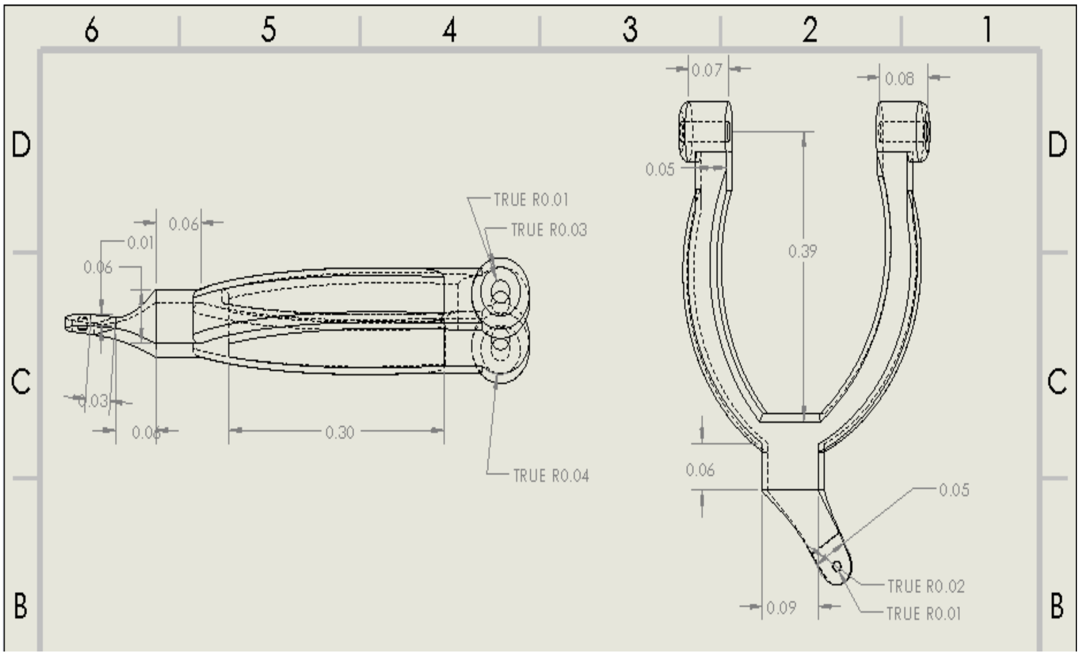


Figure 3.6: Schematic diagram of the lower control arm (all dimensions are in meters)

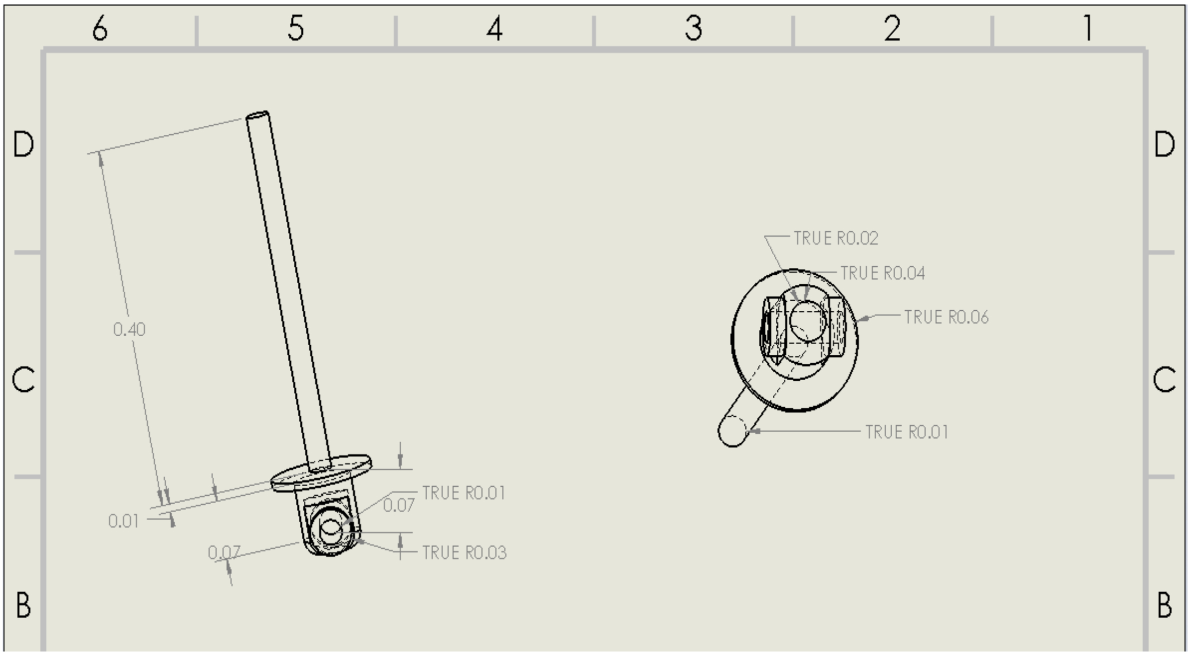


Figure 3.7: Schematic diagram of a strut (all dimensions are in meters m).

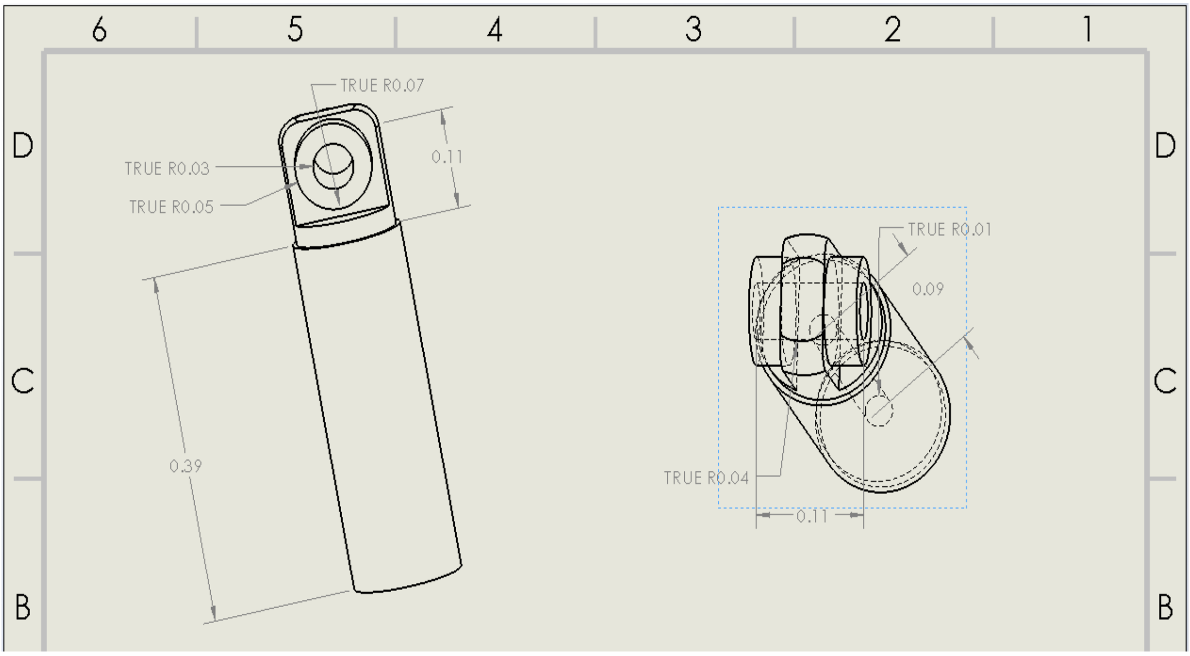


Figure 3.8: Schematic diagram of shock absorber (all dimensions are in meters m).

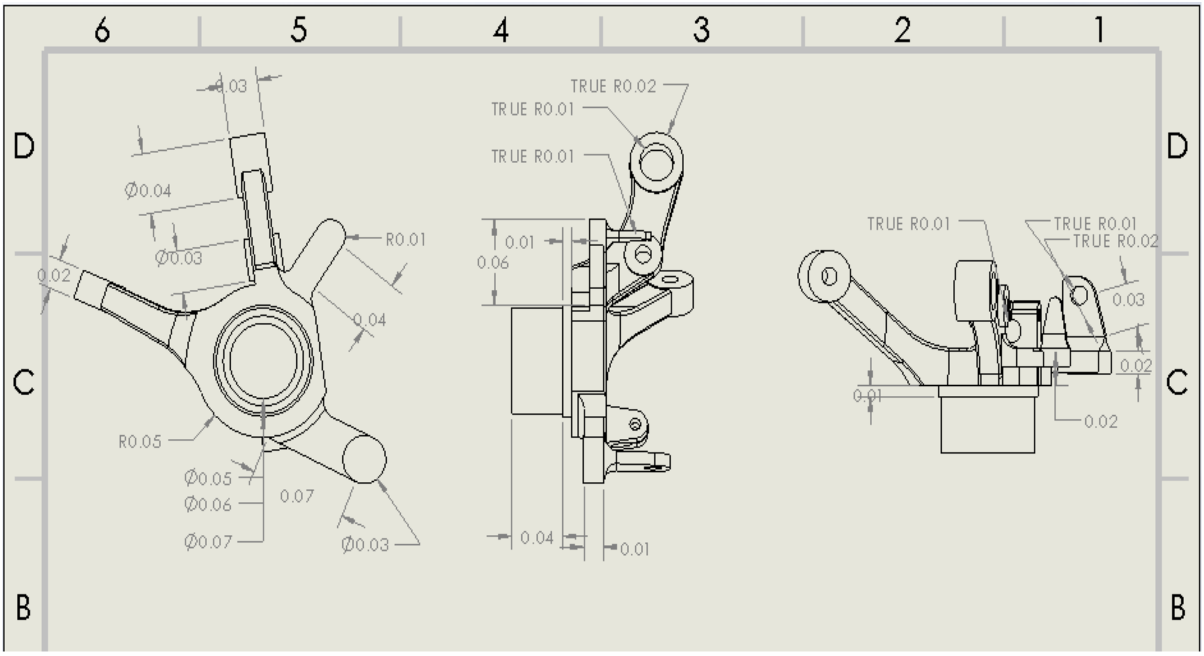


Figure 3.9: Schematic diagram of steering knuckle (all dimensions are in meters m).

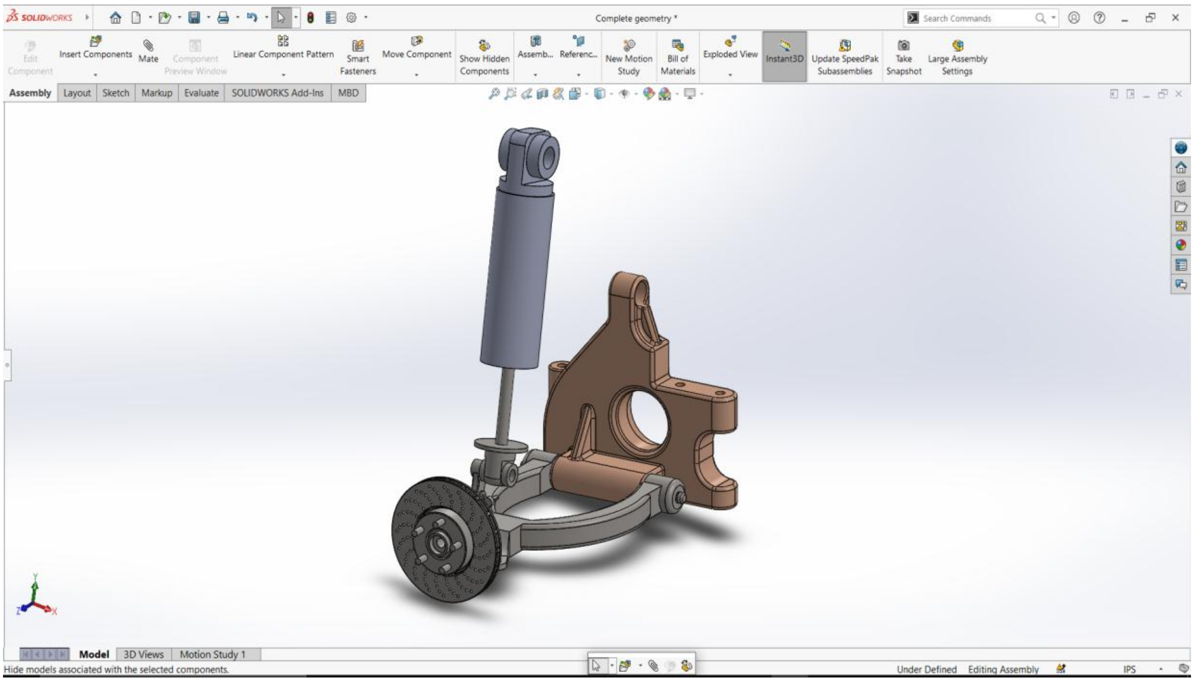


Figure 3.10: Macpherson suspension assembly

The above-shown model's mass is equal to the sum of the masses of all the parts used in the assembly.

Mass = 17.7 + 3.38 + 2.3 + 9.58 = 32.96kg

In addition, other parts such as breaks, wheels, tires are not modeled but their mass must be added hence

Total mass = 32.96kg + 1.25 + 9.07 = 43.28kg

3.3 Double wishbone suspension

A double-wishbone suspension is an independent suspension design in which the wheel is connected by two (sometimes parallel) wishbone-shaped arms to the body of the vehicle. There are two mounting points on the chassis for each wishbone or arm, as well as one joint at the knuckle to control vertical movement. The shock absorber and coil spring are attached to the lower wishbones as it supports most of the vertical load. Engineers can precisely manage the motion of the wheel throughout suspension travel using double wishbone designs, controlling factors including camber, caster, toe pattern, roll center height, scrub radius, scuff, and more to achieve the best vibrational performance from these suspension systems [3].

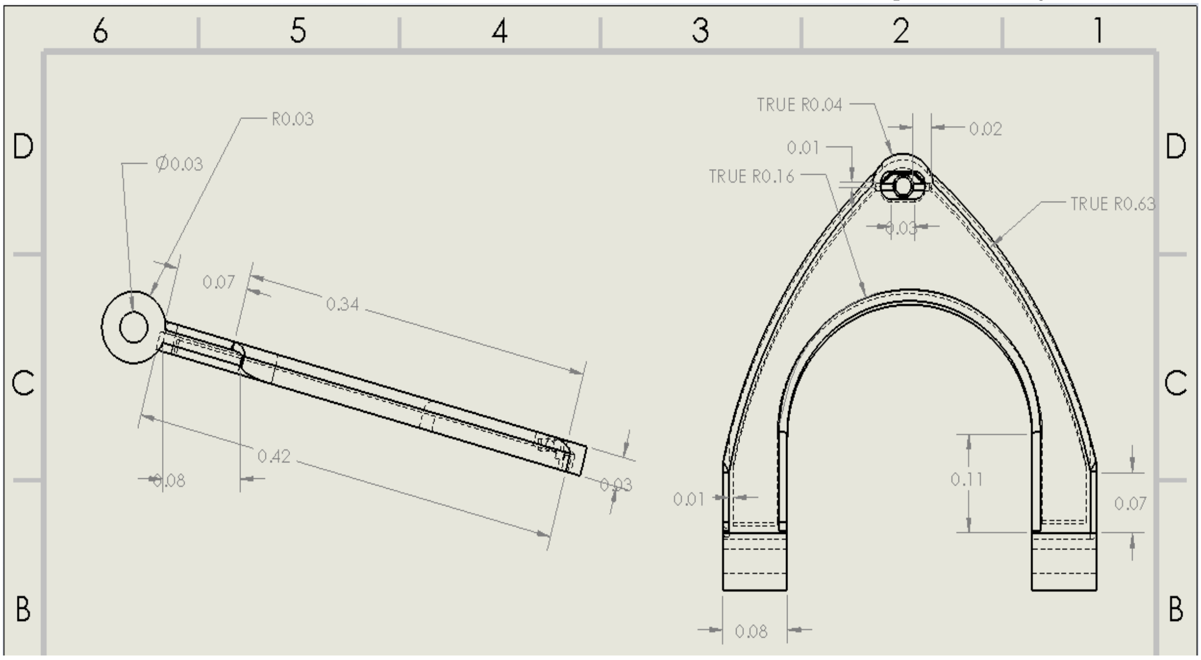


Figure 3.11: Schematic diagram of Upper control arm (all dimensions are in meters)

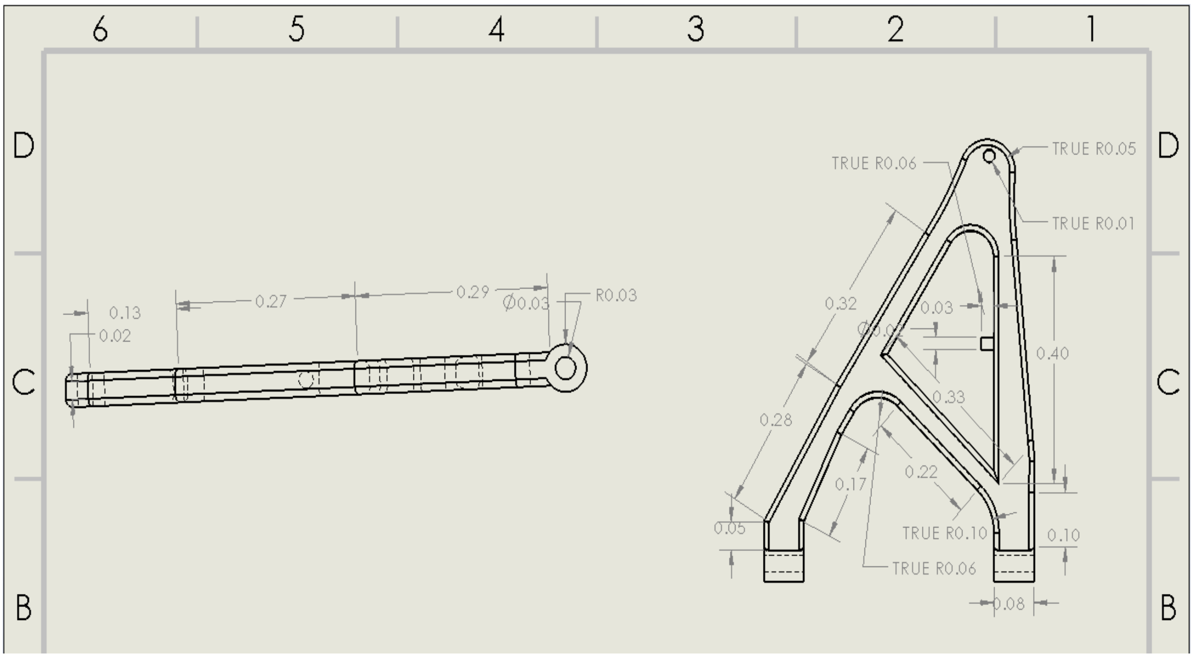


Figure 3.12: Schematic diagram of the lower control arm (all dimensions are in meters)

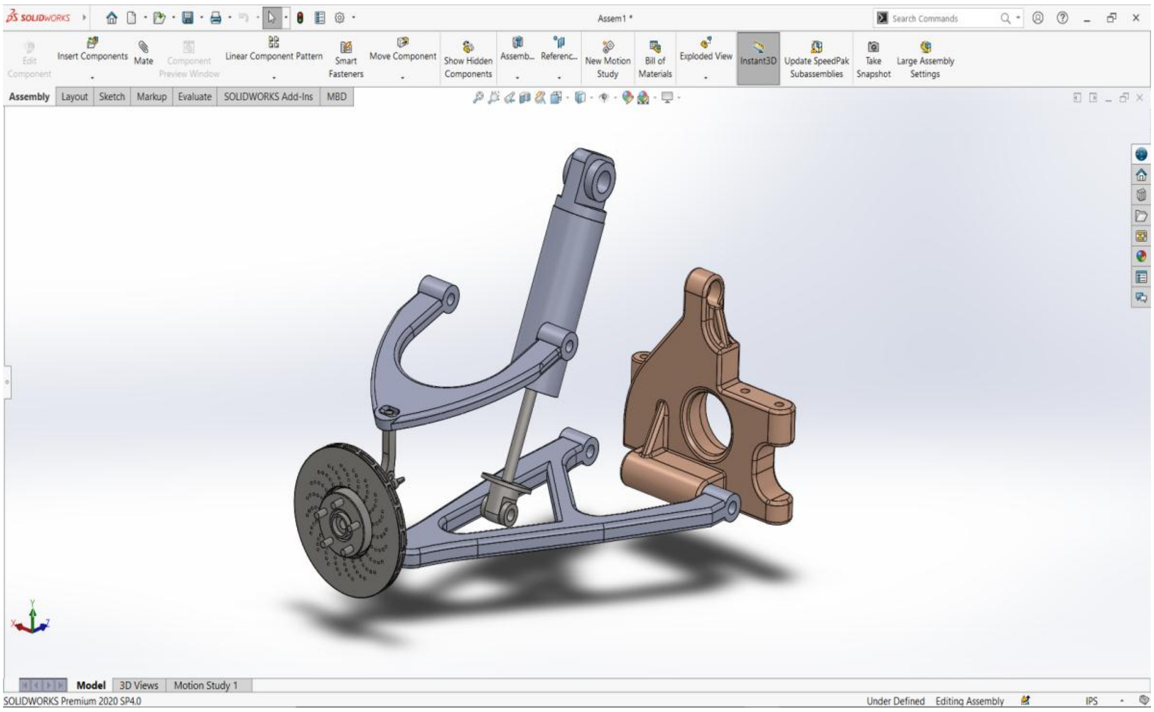


Figure 3.13: Double wishbone suspension assembly

The above-shown model's mass is equal to the sum of the masses of all the parts used in the assembly.

Mass = 12.425 + 20.8 + 3.38 + 2.3 + 9.58 = 48.485kg

In addition, there are other parts such as breaks, wheels, tyre which are not modelled but their mass must be added hence

Total mass = 48.485 + 1.25 + 9.07 = 58.805kg

4. Finite Element Analysis

The Finite Element Analysis (FEA) is the numerical simulation of a physical phenomenon using the Finite Element Method (FEM). Engineers utilize FEA software to reduce the number of physical prototypes and experiments, as well as optimize components throughout the design phase, to build better products faster and for less money.

Since mathematics are required to fully comprehend and quantify physical phenomenon, such as structural or fluid behavior, thermal transfer, wave propagation, biological cell growth, etc. Partial Differential Equations are used to describe these processes as (PDEs). Numerical approaches have also been developed over the last few decades to allow computers to solve these PDEs, and one of the most popular methods is the Finite Element Analysis

4.1 Result of finite element analysis of leaf spring suspension system

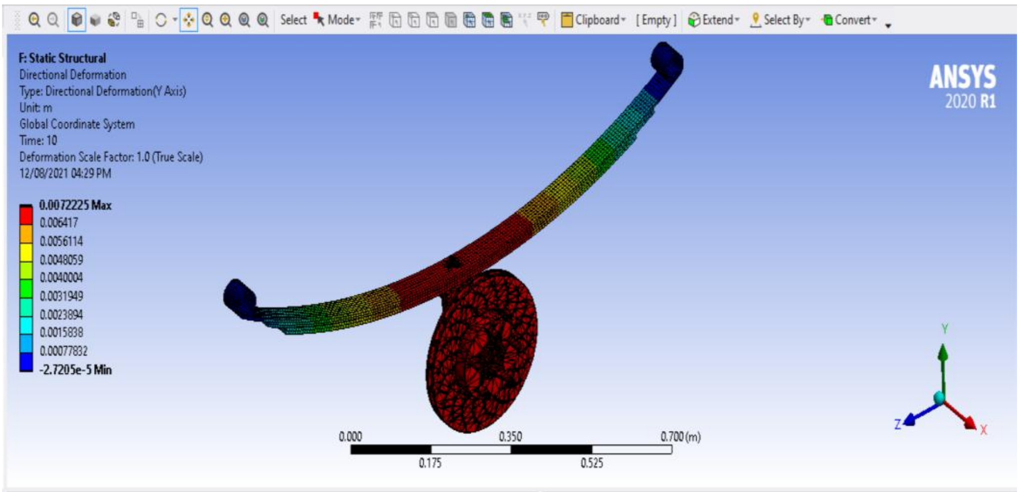


Figure 4.1: Directional displacement of leaf spring

Discussion: The above figures show an increase in deflection as force increases. Furthermore, the wheel hub and brake disk experiences the most deflection since force is directly applied to it and they are furthest from the support.

Table 4.1: Result of simulation

SNO	Time [s]	Maximum [m]	Force [N]	Stiffness (N/m)
1	0	0	0	0
2	1	0.00225	1800	800000
3	2	0.0028	2160	770000
4	3	0.00336	2510	747000
5	4	0.00391	2870	733000
6	5	0.00446	3220	722000
7	6	0.00501	3580	714000
8	7	0.00557	3930	706000
9	8	0.00612	4290	701000
10	9	0.00667	4640	696000
11	10	0.00722	5000	693000

Suspension Stiffness for our defined load of 1500kg = 730000N/m

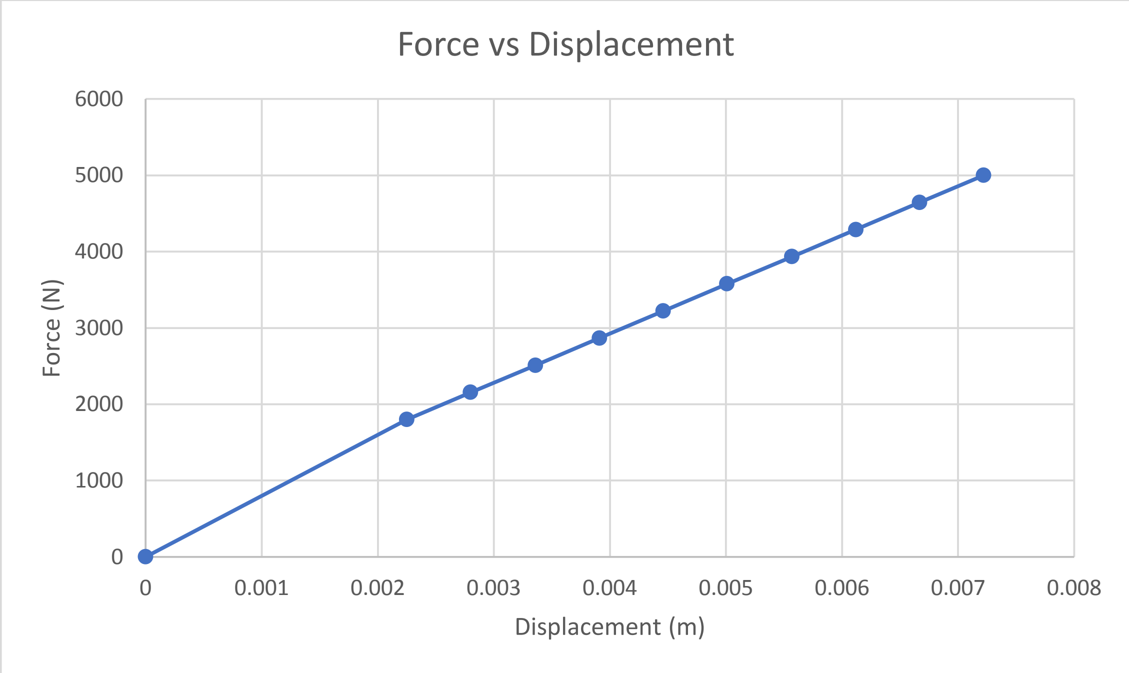


Figure 4.2: Simulation solution directional deformation

Discussion: The above graph shows the force vs displacement relation of the system at different loads. It also determines that the displacement of the suspension system increases as force is increased. The displacement increases propotionally with the applied force.

4.2 Result of finite element analysis of Macpherson suspension system

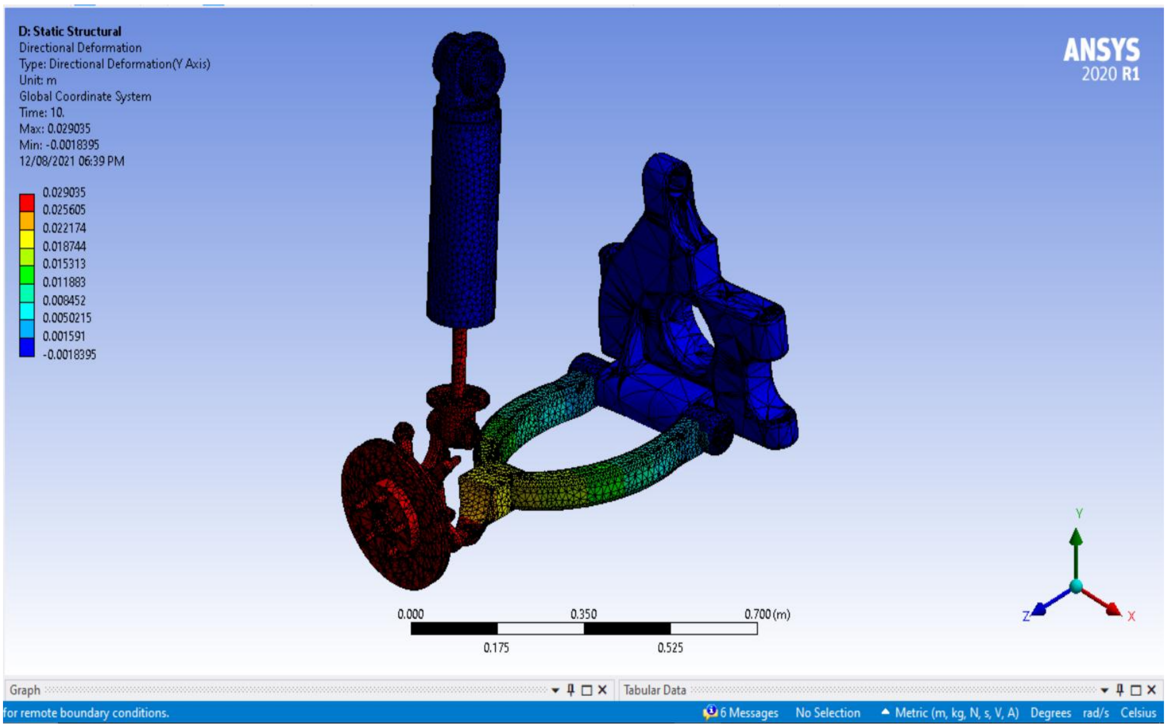


Figure 4.4: Simulation solution directional deformation

Discussion: The above figures show an increase in deflection as force increases. Furthermore, the wheel hub and brake disk experiences the most deflection since force is directly applied to it and they are furthest from the support.

Table 4.2: Simulation solution

S.NO	Time [s]	Maximum [m]	Force (N)	Stiffness (N/m)
1	1	0.00982	1800	183000
2	2	0.0115	2160	188000
3	3	0.0131	2510	191000
4	4	0.0149	2870	192000
5	5	0.0168	3220	192000
6	6	0.0189	3580	190000
7	7	0.0211	3930	186000
8	8	0.0235	4290	182000
9	9	0.0262	4640	177000
10	10	0.029	5000	172000

Equivalent Stiffness of the assembly at the defined load of 1500kg = 192000N/m

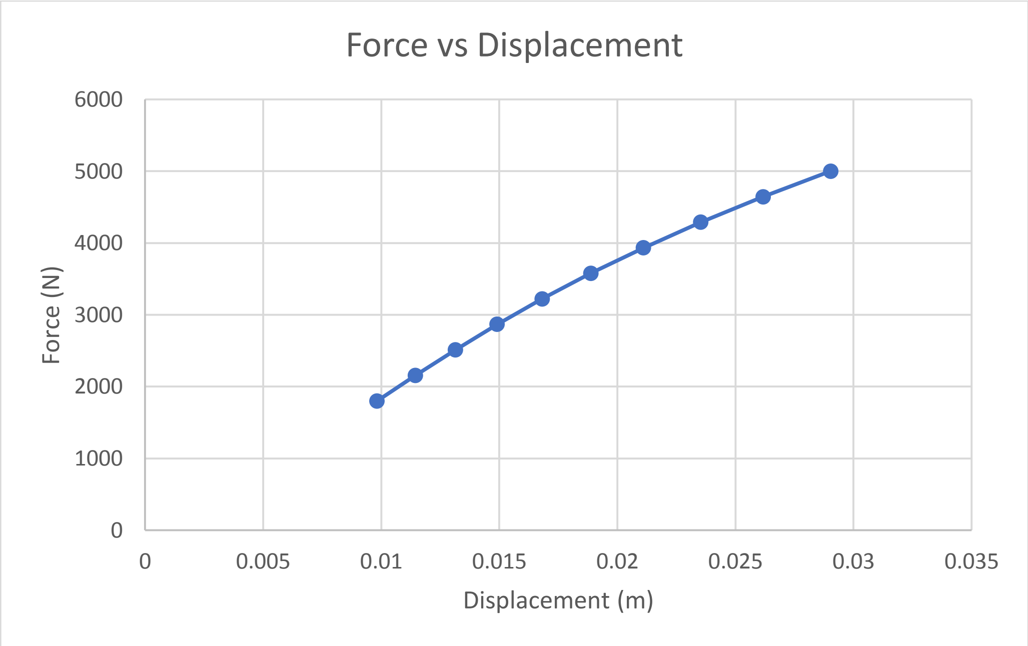


Figure 4.5: Simulation solution

Discussion: The above graph shows the force vs displacement relation of the system at different loads. It also determines that the displacement of the suspension system increases as force is increased. The displacement increases propotionally with the applied force.

4.3 Result of finite element analysis on double-wishbone suspension

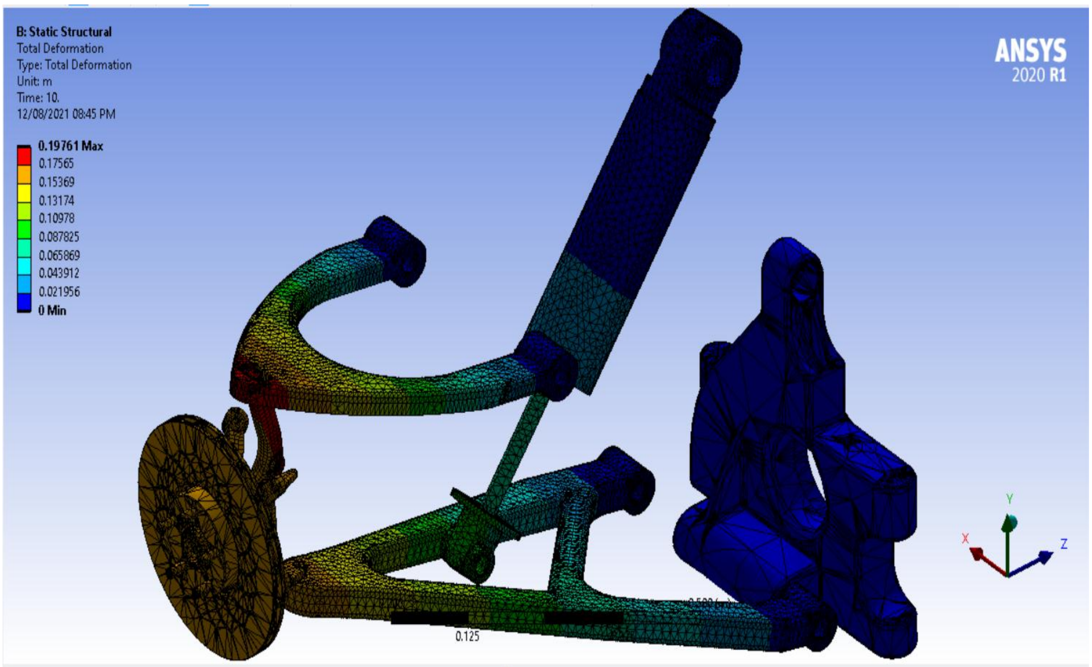


Figure 4.6: Simulation solution total deformation

Discussion: The above figures show an increase in deflection as force increases. Furthermore, the wheel hub and brake disk experiences the most deflection since force is directly applied to it and they are furthest from the support.

Table 4.3: Simulation solution total deformation

S.No	Time [s]	Maximum [m]	Force (N)	Stiffness (N/m)
1	1	0.0124	1800	145000
2	2	0.0154	2155.6	140000
3	3	0.0199	2511.2	126000
4	4	0.0285	2866.8	100000
5	5	0.0453	3222.4	71200
6	6	0.0694	3578	51600
7	7	0.0972	3933.6	40500
8	8	0.12723	4289.2	33700
9	9	0.15917	4644.8	29200
10	10	0.1924	5000.4	26000

Overall stiffness to be determined at 1500kg = 51600N/m

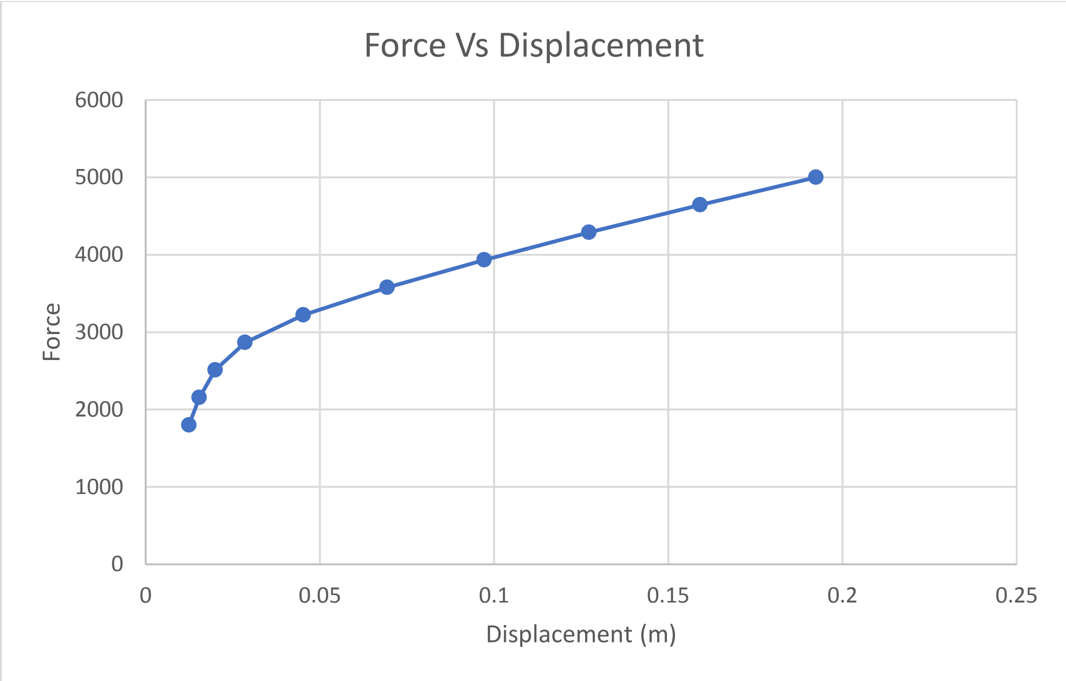


Figure 4.7: Simulation solution total displacement

Discussion: The above graph shows the force vs displacement relation of the system at different loads. It also determines that the displacement of the suspension system increases as force is increased. Since the strut is mounted at an angle the force vs displacement has a higher slope at the start but as the load increases the camber angle increases and alligns with the anlge of the strut resulting in an near propotional increase.

5.1 System Modelling of the suspension system

Modeling of automotive suspension is of great interest for automotive and vibration engineers. Vehicle ride quality is a prime concern for engineers when a vehicle passes over a speed bump. For our analysis 2 DOF quarter car model (Fig. 1) has been developed with the following assumptions [4]:

- The vehicle is a rigid body with the suspension
- The suspension consists of suspension spring, absorber, sprung, the un-sprung mass of the body
- Tire stiffness and tire absorptivity is considered separately

Parameters used for mathematical modeling are as follows:

M = Sprung Mass
m = Un-sprung Mass
Ks = Suspension spring stiffness
Kt = Tire stiffness
Cs = Damping coefficient of absorber
Ct = Damping coefficient of tire
w = Road input (height of speed bump)
X1 = Sprung mass vertical movement
X2 = Un-sprung mass vertical movement
Δ = Suspension Travel

The equation of motion for the sprung and unsprung mass of the model considering it moving over a speed bump will become [5] [6]:

$$\ddot{X}_1 = -\frac{1}{M}K_s(X_1 - X_2) - C_s(\dot{X}_1 - \dot{X}_2)\ldots\ldots(1)$$

$$\ddot{X}_2 = \frac{1}{m}K_s(X_1 - X_2) + C_s(\dot{X}_1 - \dot{X}_2) - K_t(X_2 - w) - C_t(\dot{X}_2 - \dot{w})\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

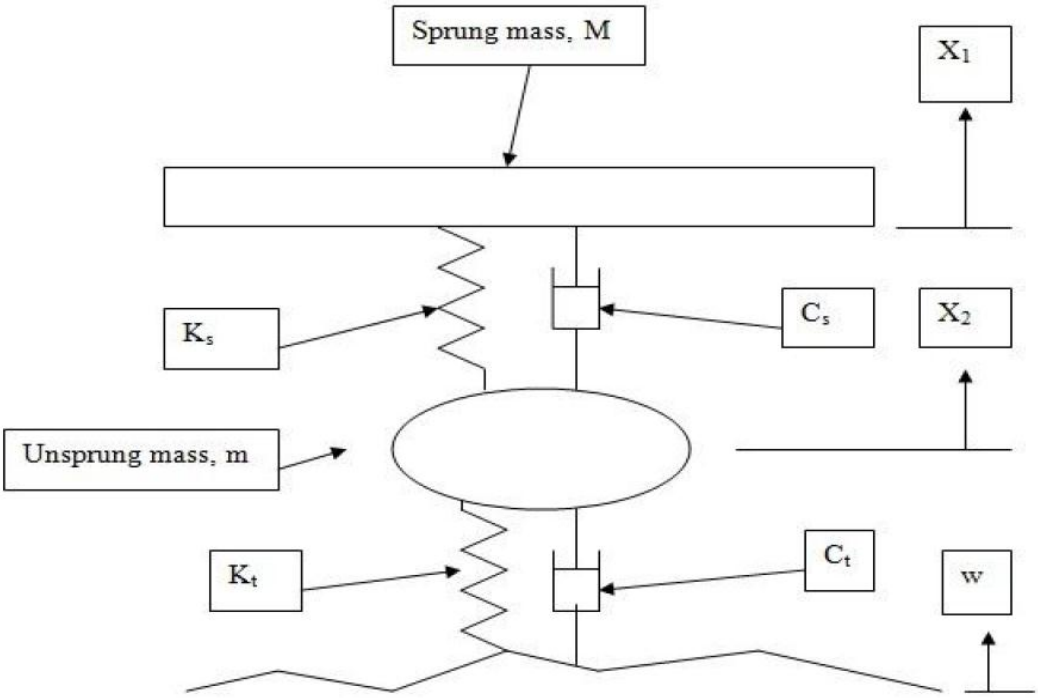


Figure 5.1: Quatre car passive suspension model

Derivation for the governing equation of two degrees of freedom mass damping system

For the sprung mass M, the following forces are applied

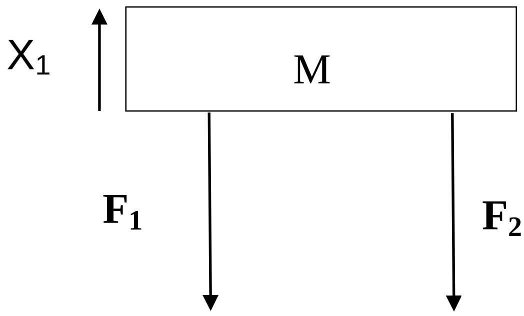
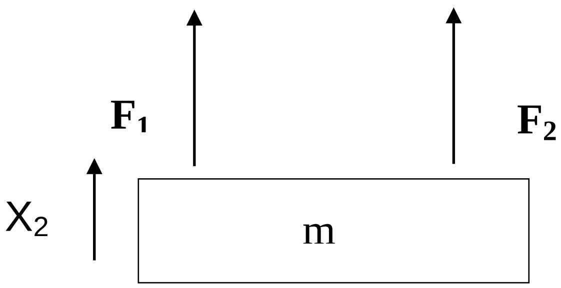


Figure 5.2: Sprung mass

For the unsprung mass, the following forces are applied



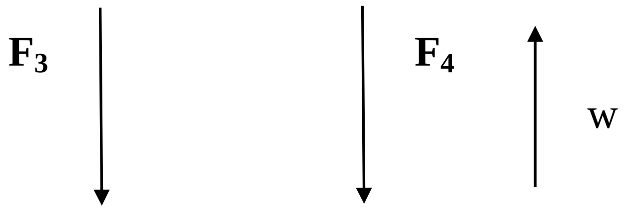


Figure 5.3: Unsprung mass

$F_1 = K_s (X_1 - X_2)$ ……. (This equation describes the force on spring due to displacement)
$F_2 = C_s(\dot{X}_1 - \dot{X}_2)$ … (This equation describes the force on a damper due to displacement)
$F_3 = K_t (X_2 - w)$
$F_4 = C_s(\dot{X}_2 - \dot{w})$
Now according to newton second law
F = ma
m = mass
a = acceleration
For mass M

$$\sum F_{X1} \rightarrow M\ddot{X} = -F_1 - F_2$$

$$M\ddot{X} = -K_s(X_1 - X_2) - C_s(X_1 - X_2) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

For Mass m

$$\sum F_{X2} \rightarrow m\ddot{X} = F_1 + F_2 - F_3 - F_4$$

$$m\ddot{X} = K_s(X_1 - X_2) + C_s(X_1 - X_2) - K_t(X_2 - w) - C_t(\dot{X} - \dot{w}) \ldots\ldots\ldots (4)$$

5.2 Simulation under MATLAB Simulink
MATLAB Simulink can solve ordinary differential equations of both linear and nonlinear types. MATLAB Simulink block diagrams are created for the 2DOF quarter car model to evaluate sprung mass displacement, as well as suspension travel reactions of the suspension system when it goes over a speed bump and a rough road.
In this section, we need to simulate the dynamic response of each of the suspension systems. The suspension system is simulated with the application of MATLAB Simulink. The mathematical modeling is done in the previous section [7].
The Simulink library and logic of the Simulink simulation as shown below in fig.2 is developed according to the mathematical equations (3) and (4).

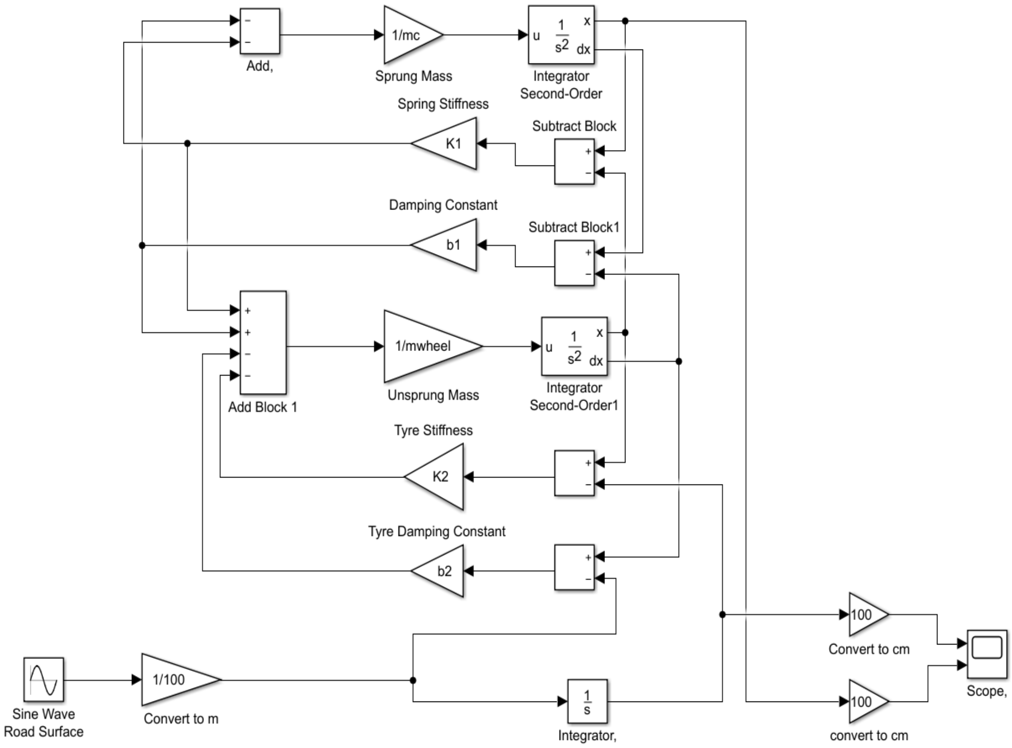


Figure 5.4: Simulink block diagram of two degrees of freedom mass-spring-damper system

Table 5.1: Input Parameters used in system simulation

S no	Parameters	Symbol	Quantity for leaf spring suspension	Quantity for Macpherson suspension	Quantity for Double wishbone suspension
1	Mass of the vehicle body	m_c	1500kg	1500kg	1500kg
2	Mass of the suspension system	m_{wheel}	33.105kg	43.28kg	58.82kg
3	Coefficient of stiffness of spring	K_1	730000N/m	192000N/m	51600N/m
4	Damping Coefficient of the damper	b_1	66181.56Ns/m	33941.125Ns/m	17595Ns/m
5	Coefficient of stiffness of the tire	K_2	500000N/m	500000N/m	500000N/m
6	Damping Coefficient of tire	b_2	15000 Ns/m	15000Ns/m	15000Ns/m

5.3 Simulation results for Double Wishbone suspension system

The input parameters used in Simulink to generate the input signal are as follows.

$$\theta = amp * Sin(freq * t + Phase) + Bias$$

Amp = 400
Freq = 200
t = 1s to 5s
phase = 0
Bias = 0

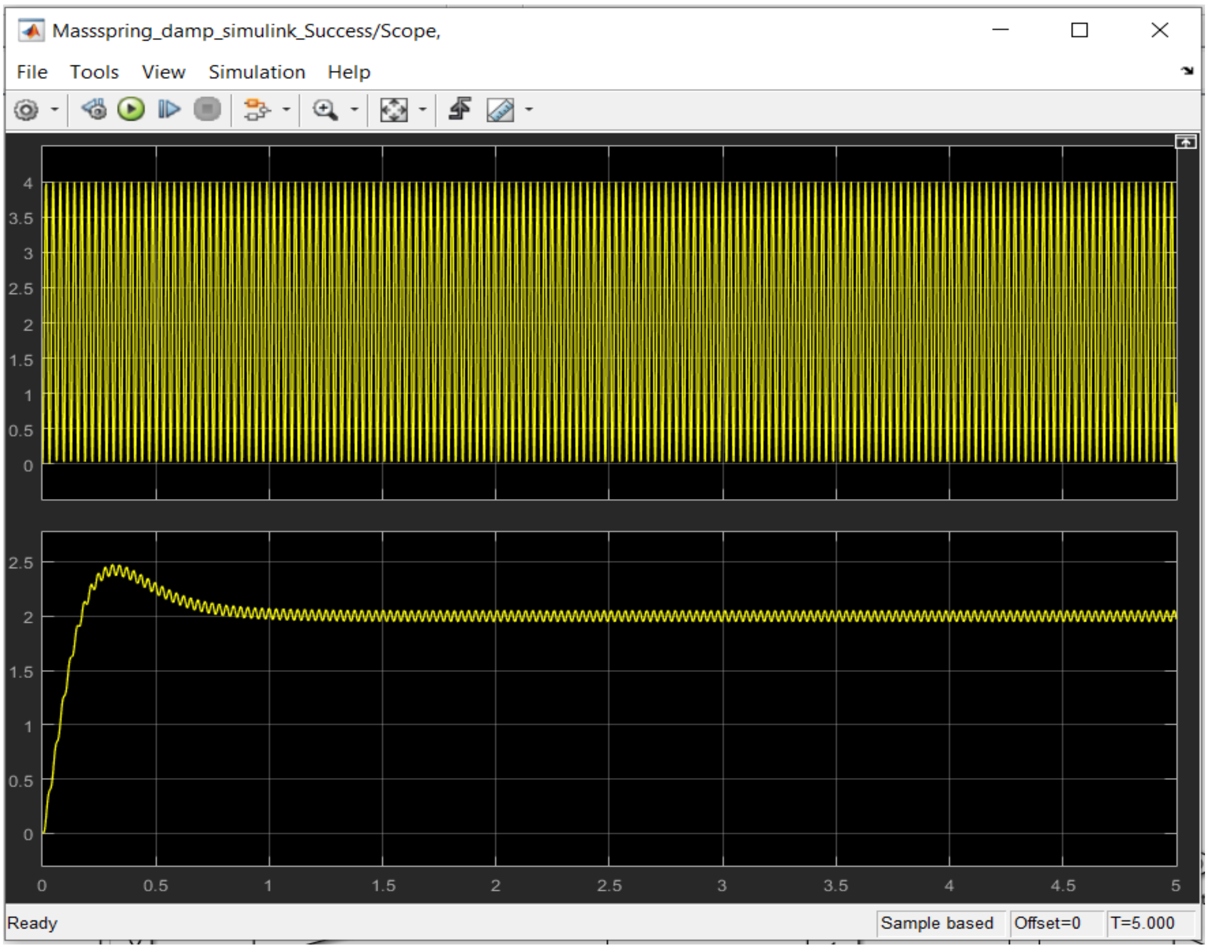


Figure 5.5: Input signal vs Sprung mass displacement

Discussion: Analytical results of suspension system for ¼ car model for a rough road with a bump of 4 cm and a frequency of 200 Hz shows that the sprung mass did not overshoot, and displacement was limited to 2.4 cm instead of the applied 4cm. Furthermore, after the initial bump, the suspension system settles around 2cm of displacement and oscillates around it with an amplitude of 0.05cm [8] [9]. Since this situation represents a car driving over a rough surface at an increased speed, from a comfort point of view this is the best result compared to the previous suspension system since this produced the lower initial bump and then settled into a lower frequency. Additionally, this type of suspension system settled into a fixed frequency instantly. The Y-axis shows displacement, and the X-axis shows time.

5.4 Simulation results for Leaf spring suspension system

The input parameters used in Simulink to generate the input signal are as follows.

$$\theta = amp * Sin(freq * t + Phase) + Bias$$

Amp = 400
Freq = 200
t = 1s to 5s
phase = 0
Bias = 0

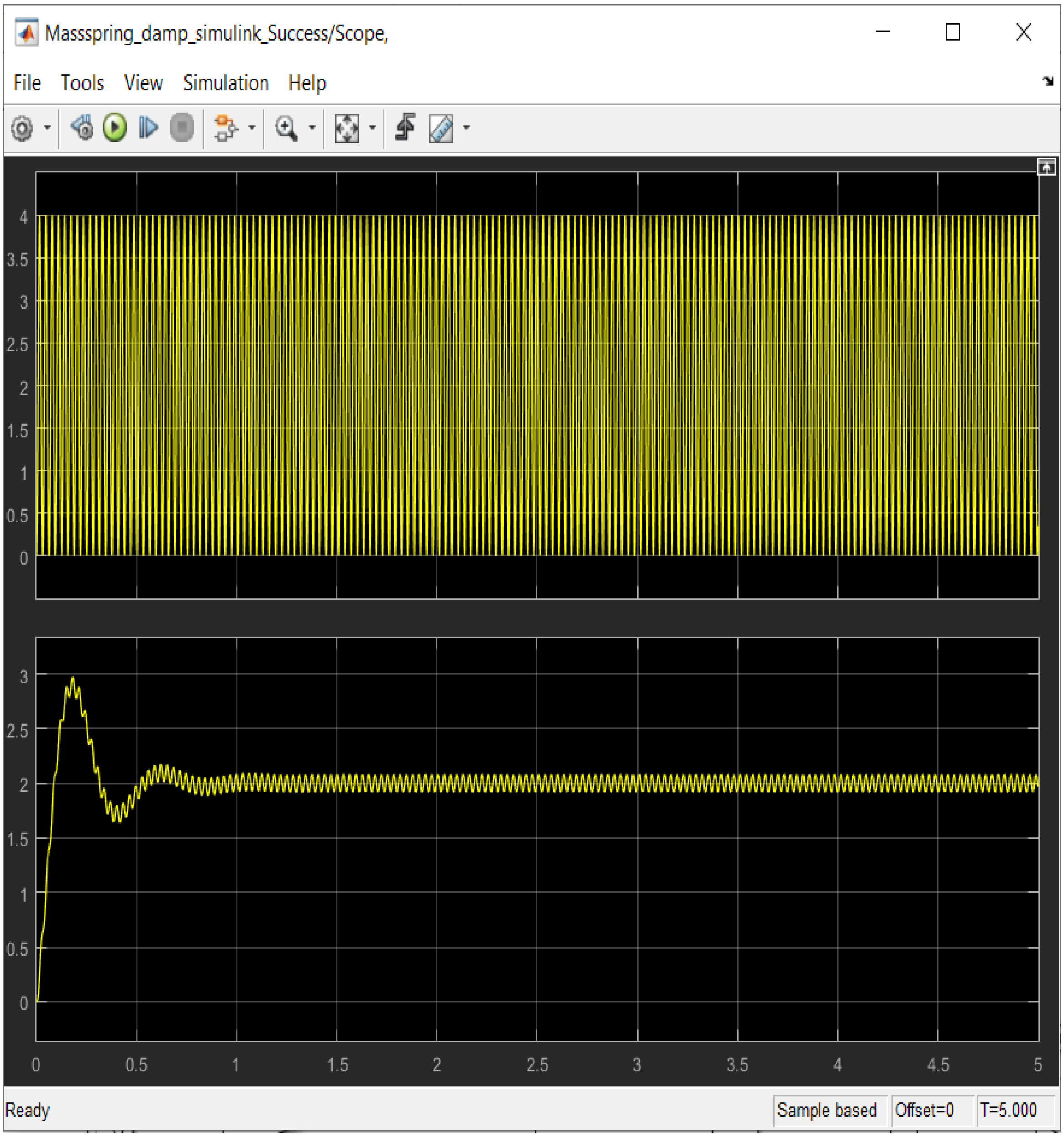


Figure 5.6: Input signal vs sprung mass displacement

Discussion: Analytical results of suspension system for ¼ car model for a rough road with a bump of 4 cm and a frequency of 200 Hz shows that the sprung mass did not overshoot and displacement was limited to 2.9 cm instead of the applied 4cm. Furthermore, after the initial bump, the suspension system settles around 2cm of displacement and oscillates around it with an amplitude of 0.1cm [8] [9]. From a comfort point of view, this indicated that the suspension system is working as intended but the result is the worst among the tested suspension. Additionally, this type of suspension system takes 0.8s to settle into a fixed frequency. The Y-axis shows displacement and the X-axis shows time.

5.5 Simulation results for Macpherson suspension

The input parameters used in Simulink to generate the input signal are as follows.

$$\theta = amp * Sin(freq * t + Phase) + Bias$$

Amp = 400
Freq = 200
t = 1s to 5s
phase = 0
Bias = 0

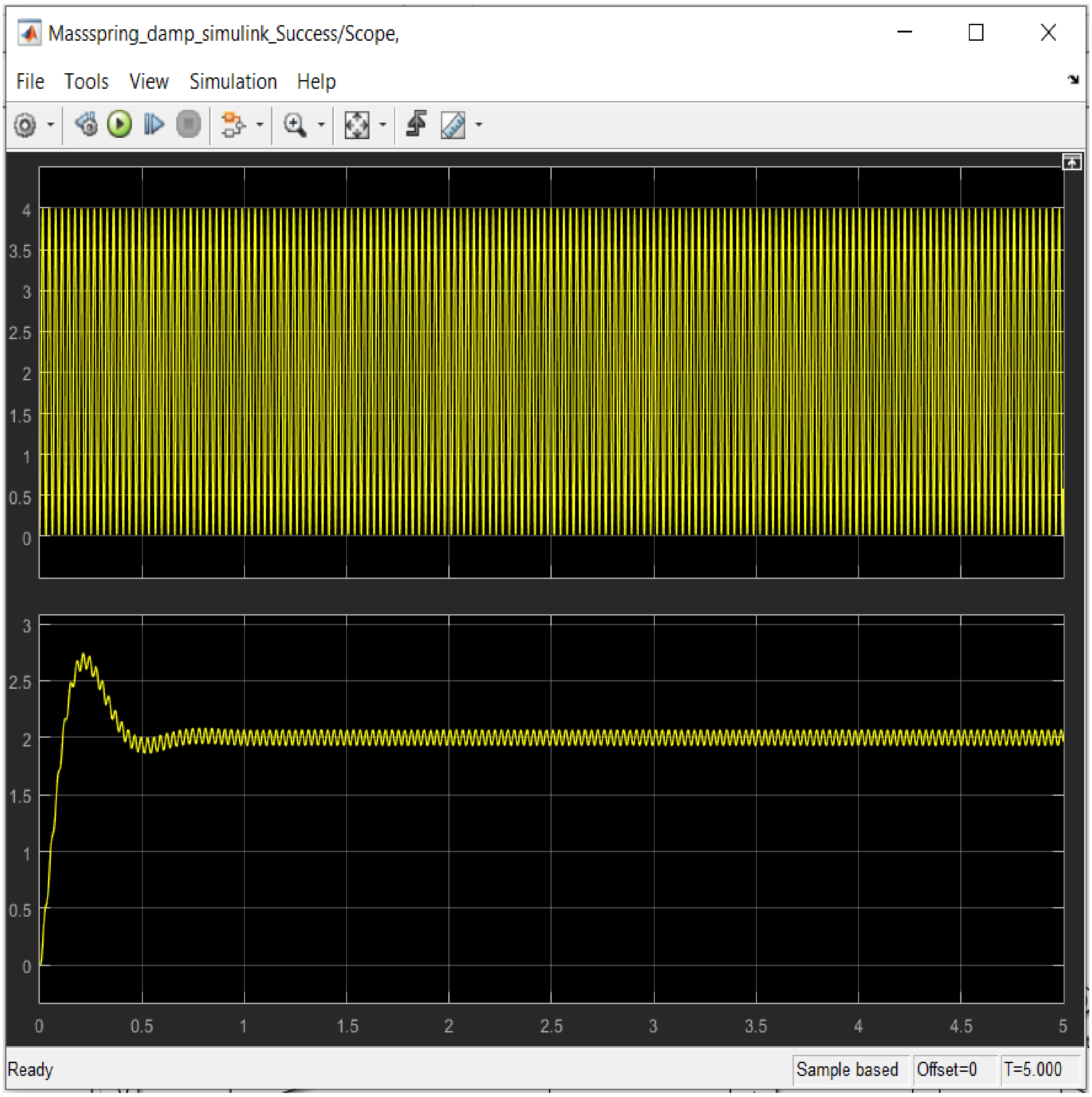


Figure 5.7: Input signal vs sprung mass displacement

Discussion: Analytical results of suspension system for ¼ car model for a rough road with a bump of 4 cm and a frequency of 200 Hz shows that the sprung mass did not overshoot, and displacement was limited to 2.7 cm instead of the applied 4cm. Furthermore, after the initial bump, the suspension system settles around 2cm of displacement and oscillates around it with an amplitude of 0.1cm [8] [9]. Since this situation represents a car driving over a rough surface at an increased speed, from a comfort point of view this is an improved result compared to the previous suspension system since this produced the lower initial bump and then settled into a lower frequency. Additionally, this type of suspension system settled into a fixed frequency after 0.5s. The Y-axis shows displacement, and the x-axis shows time.

5.6 Result comparison of the three types of the suspension system

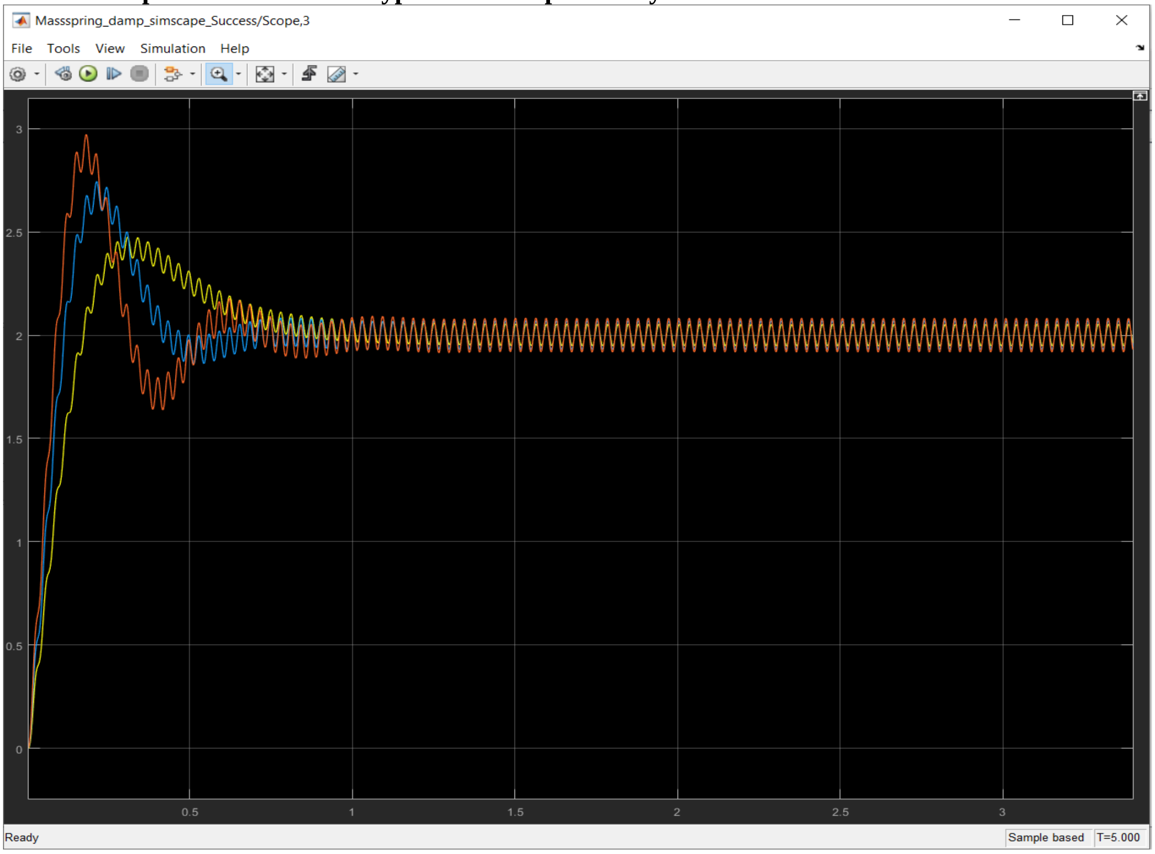


Figure 5.8: Comparison of the three Sprung mass displacement

Discussion: The red line, blue line, and yellow represent the sprung mass displacement in cm of leaf spring. The Y-axis shows displacement, and the X-axis represents time. suspension system, Macpherson suspension system, and double wishbone suspension system, respectively.

6. Conclusion:

The project is a comparison of three suspension systems. These include the Macpherson suspension system, double wishbones suspension system, and leaf spring suspension. These suspension systems were compared using Ansys for performing static structural analysis and MATLAB for performing vibrational analysis. The structural analysis shows that the leaf spring suspension system has the highest stiffness value among the suspension system which makes it ideal for heavy vehicles. The downside was that it had the worst vibrational performance. The double-wishbone suspension system had the best vibrational performance but did not possess a high stiffness value which concludes that the double-wishbone suspension is best for high-speed lightweight vehicles. The Macpherson suspension system comes in the middle in terms of stiffness and vibrational performance. It is suitable for high-speed and moderate-weight vehicles.

References

[1] A.-Q. H.A, "Al-Qureshi, H. A. "Automobile leaf springs from composite materials," *Journal of materials processing technology,* pp. 58-61, 2001.

[2] Z. D. J. Y. F. a. L. L. M. Liu J, "Optimized design for a Macpherson strut," *International Journal of Automotive Technology,* pp. 29-35, 2008.

[3] A. G. C. A. S. G. a. R. B. Vivekanandan N, "Design analysis and simulation of Double wishbone suspension system," *International Journal of Mechanical Engineering,* pp. 1-7, 2014.

[4] T. X. a. W. L. Shi Ying, "A model of the 1 / 4 of a simple suspension model simulation and analysis," *IEEE International Conference on Educational and Information Technology,* vol. 1, pp. 92-94, 2010.

[5] T. T. a. R. M. Claudiu Valentin Suciu, "Modeling and Simulation of a Vehicle," *Journal of Telecommunications,* pp. 83-89, 2012.

[6] A. S. a. K. K.Kamalakannan, "Performance Analysisand Behaviour Characteristics of CVD (Semi Active) in Quarter Car Model," *Jordan Journal ofMechanical and Industrial Engineering,* vol. 5, no. 3, pp. 261-265, 2011.

[7] O. O. Oluwole, "Matlab and Simulink Use in Response Analysis of Automobile Suspension System in," *International Journal of Traffic and Transportation Engineering,,* vol. 1, no. 2, pp. 19-31, 2012.

[8] C. A. B. a. L. P. P. Marius Constantin Popescu, "Road Vehicle Suspension Modelling," *Proceedings of the 8th WSEAS International Conference on Education and Educational Technology,* pp. 113-120, 2009.

[9] A. K. a. A. Stribrsky, "A full-car model for active suspension - some practical aspects," *IEEE International Conference on Mechatronics (ICM),* pp. 41-45, 2004.