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Design of Trailing Arm Suspension

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***Abstract*— Suspension is a term given to the system of spring, damper and linkages that connect wheel to vehicle body which allows relative motion, maintains contact between wheel and road and also damps out road shocks. Suspension system aims to improve the ride comfort, handling and safety. Now a day's most of the passenger cars use independent suspension system which allows wheel to relative to frame without affecting other wheels. Most commonly used rear suspension is trailing arm suspension system.**

This paper deals with mathematical modelling of a quarter car trailing arm suspension system. The quarter car model of two degree freedom system is solved in MATLAB and time response is plotted. The parameters such as spring stiffness, damping ratio of sprung and un-sprung masses of the quarter car model are optimized to get good comfort and vehicle handling. Design of the suspension geometry is carried out in CAD software. The optimized values of spring stiffness, damping coefficient of sprung and un-sprung masses are coupled with the geometry design and is analyzed in LOTUS SUSPENSION ANALYSER to get various values of the suspension geometry.

Keywords—traling arm, matlab, lotus suspension analysis, quarter car.

I. INTRODUCTION

Automobile chassis is mounted on the axle through a suspension system. Due to uneven road profile and variable velocity, vehicle is subjected to random excitation. It affects comfort and road handling characteristics. The main objective of suspension system is to safeguard the passenger against road shock which may be in the form of bounce, roll, pitch etc. Also suspension has to limit the body displacement to a minimum. Suspension system of a vehicle is subjected to a force due to vertical loading, rolling, braking, acceleration and cornering.

Function of suspension system are-

- To support vehicle and passenger weight.
- To safeguard the passenger against road shock and provide riding comfort.
- To maintain steering and vehicle handling characteristics, which is achieved by means of mechanical linkages and controlling relative motion between sprung and unsprung mass.
- To provide stability to vehicle during pitching or rolling, while in motion.

Spring and damper are most widely used element in vehicle. Spring reduces the force transmitted to the chassis while travelling over bump reduces the acceleration of the chassis while wheel falls on a depression or pothole. Damper reduces the tendency of the chassis to bounce up and down after the disturbance and also prevents the excessive built up of amplitude of bounce when the excitation frequency is nearer to the natural frequency the system.

Suspension system is classified into passive, active, semi-active system. Passive suspension system satisfies the requirements only at certain operating condition. Hard springing will increase acceleration force during light loading and soft springing will reduce the body height when heavily loaded. Active suspension controls the vertical movement of wheel through an on-board system. Active suspension systems use separate actuators that exert an independent force on the suspension to improve the riding characteristics. Semi-active systems can only change the viscous damping coefficients and do not impart any additional force to the suspension system. In most of passenger passive suspension system is widely used. In passive suspension system, there is always compromise between passenger comfort and handling. In passive suspension system high stiffness spring gives good handling but poor comfort. Low stiffness spring gives good comfort and less handling. The main advantage of passive suspension system is that the design is simple and also economical. Also maintenance cost is low.

Trailing arm suspension is mostly used for the rear suspension of the passenger cars. It is also called as crank axle suspension system. Control arm placed longitudinally along the driving direction and mounted on suspension sub frame or body on both the sides. Control arm is subjected to high torsional and bending stresses. It allows the wheel to move up and down only. No change in camber and toe due to lateral and vertical forces. Increased un-sprung mass of trailing arms reduces the ride quality.

In this wok quarter car model is considered for the analysis. In quarter car model, the vehicle is converted into spring and damper system which has one fourth mass of the vehicle. The equations of motion are derived for the quarter model.

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The work on the suspension system shows the comparison between active, semi active and passive suspension system. It is also stated that comfort can be increased by using low stiffness springs [1].

Another study shows the effect of vehicle velocity and wheel base on RMSAR value i.e. Root Mean Square Acceleration Response. In that it is found that RMSAR is proportional to vehicle velocity and inversely proportional to wheel base. The two degree of freedom half car model is considered and studied for the random road input values [2].

The improvement in passive suspension system is done by using controllers to change the spring stiffness and damping. Proportional Integral Derivative (PID) controller is designed for quarter car model of passenger car to improve ride comfort and handling characteristics. The sprung mass displacement and acceleration has been reduced by using active suspension system [3].

II. Objective

The objectives of the present work are as follows:

A. *Mathematical Modeling*

The physical suspension system of a vehicle is converted to quarter car model with spring mass and damper system, where in vibration response is analyzed.

B. *Arriving at Optimum Values of Spring and Damper Coefficient*

To get the good comfort and handling characteristics.

C. *Static Geometry for Trailing Arm*

To get the least amount of deviation in the suspension geometry at bump and rebound.

D. *Kinematic Analysis of Trailing Arm Suspension*

Changes in roll center and wheel base change can by analyzed in LOTUS Suspension Analysis [6]. Response time of the designed quarter car model is plotted in SIMULINK.

III. METHODOLOGY

To study any suspension system, it is important to derive the equations of motion related to the system and then solving them by appropriate mathematical tool. Generally for analysis of vehicle suspension system various types of suspension models are used. The most commonly used models are quarter car model, half car model and full car model. In the present work the quarter car model is considered for the study. The advantage of the quarter car model is that it is simple and easy to analyze. In this particular model, only one fourth of the vehicle is taken into consideration to develop a vehicle suspension. This model is two dimensional model because only vertical movement is taken into consideration. It is basically consists of a single wheel which is represented in the form of a spring. However in some cases the wheel can be considered as equivalent to a parallel combination of spring and a damper. The actual shock absorber is assumed to support only one fourth of the weight of the total car body mass including passenger’s weight. The quarter car model considered in this work is shown in fig 1.

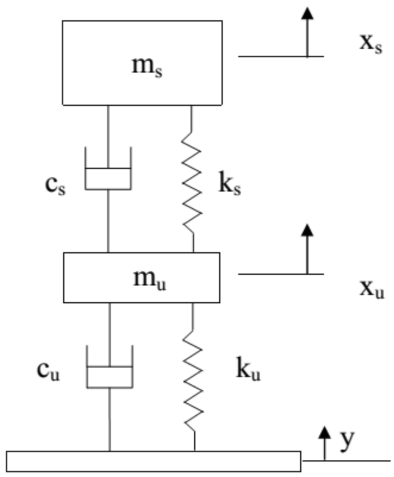

Fig. 1. Quarter car model

The equations derived from the quarter car model are as follows:

$$m_s\ddot{x}_s + k_s(x_s - x_u) + c_s(\dot{x}_s - \dot{x}_u) = 0$$
$$m_u\ddot{x}_u + k_u(x_u - y) - k_s(x_s - x_u) + c_u(\dot{x}_u - \dot{y}) - c_s(\dot{x}_s - \dot{x}_u) = 0$$

Where,
m_s = Sprung Mass (Kg)
m_{us} = Unsprung Mass (Kg)
k_s = Suspension Spring Stiffness (N/m)
k_{us} = Tire Stiffness (K/m)
c_s = Damping Coefficient (Ns/m)
c_u = Tire Damping Coefficient (Ns/m)
x_s = Sprung Mass Displacement (m)
x_{us} = Unsprung Mass Displacement (m)

Suspension parameters are taken from the literature survey.

TABLE I. Suspension Parameters

m_s	Sprung Mass	250 Kg
m_{us}	Unsprung Mass	50 Kg
k_s	Spring Stiffness	18600 N/m
k_u	Tire Stiffness	196000 N/m
c_s	Damping Coefficient	1000 Ns/m
b	Wheel track	1.7m
a	Wheel base	2.7m
	Tyre size	255/35 ZR19

In order to approach for the design of trailing arm suspension, the parameters involved must be studied. The trailing arm acts as a rocker member of the suspension where in it is mounted to the chassis by pivot points. The spring and damper are attached to the arm by pivot points, to get required relative motion during moment of the arm. The parameters involved are [5]:

Length of trailing arm
Angle of trailing arm.
Ride height.
Inclination of the spring and damper.

The pivot point of the trailing arm is subjected to forces like; thrust force, braking force and cornering forces.

Considering all these forces, the pivot point of the trailing arm is divided into two points, to share the load effectively.

Initial approximation of the geometry is considered for specified wheel track and base, and the effect of each parameter on the suspension geometry is determined by varying single parameter and keeping all other parameters constant. By this way, optimum value of the parameters are obtained. The CAD model of the trailing arm is prepared by the optimum values of the suspension geometry and is shown in the figures 2-4.

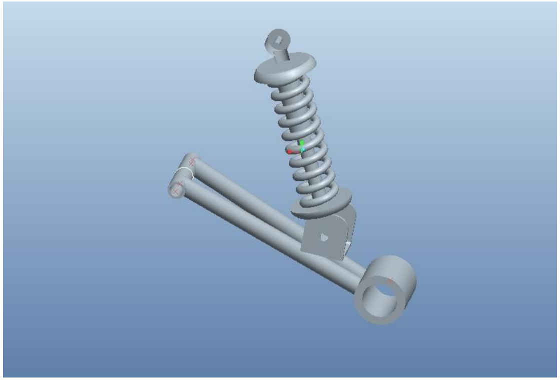

Fig. 2. Isometric view of the trailing arm suspension

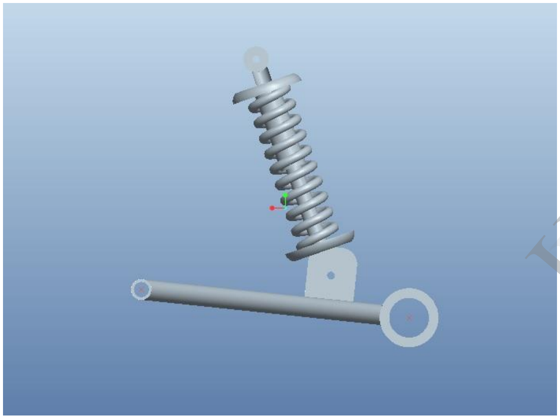

Fig. 3. Side view of the inclined trailing arm suspension

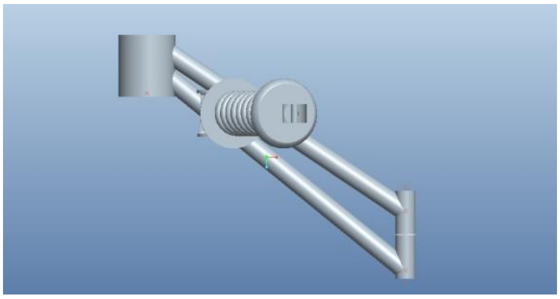

Fig. 4. Top view of the trailing arm suspension

The assembled static geometry model is imported in the LOTUS suspension analyzer by getting the co-ordinates of the pivot points of the entire assembly.

TABLE II. STATIC TRAILING ARM PARAMETERS

Parameter	Value
Camber Angle	0 deg
Toe Angle	0 deg
Roll Centre Height	-26.33 mm

TABLE III. SUSPENSION CO-ORDINATES

Component	X (mm)	Y (mm)	Z (mm)
Trailing pivot point to the frame	3026.36	-254.00	317.50
Damper lower trailing arm end	3364.69	-368.71	279.78
Damper body end	3184.40	-361.95	698.50
Wheel spindle point	3656.03	-469.90	251.33
Wheel centre point	3656.03	-622.30	251.33

The derived equations of motions are solved by using SIMULINK programming. The parameters shown in table I are used in the programming to get the sprung mass displacement, sprung mass acceleration. The base excitation given is the sine profile with amplitude of 0.1m and frequency of π rad/sec. The SIMULINK program used to solve the equations is shown in the fig. 5.

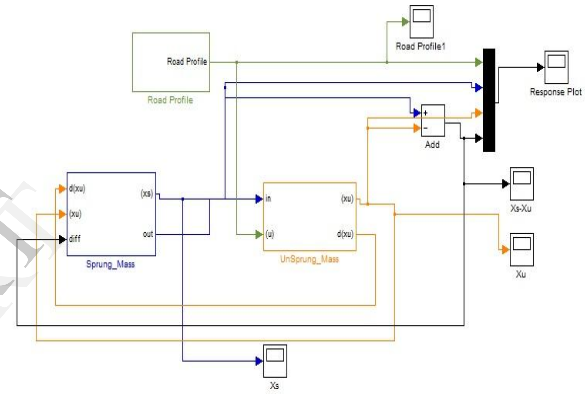

Fig. 5. Simulink model

IV. RESULTS AND DISCUSSION

After solving the equations by using the SIMULINK program following results are obtained.

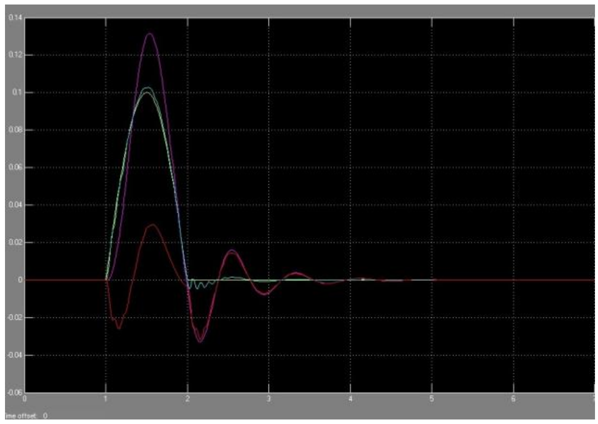

Fig. 6. Response of system for sine bump input in SIMULINK

The above response plot shows the sprung mass deflection, unsprung mass deflection and suspension working space with respect to time for the sine bump road input. From the response plot maximum sprung mass displacement and acceleration are calculated with constant value of damping. By varying the values of Kt/Ks, the displacement and acceleration obtained and are plotted in the following graphs.

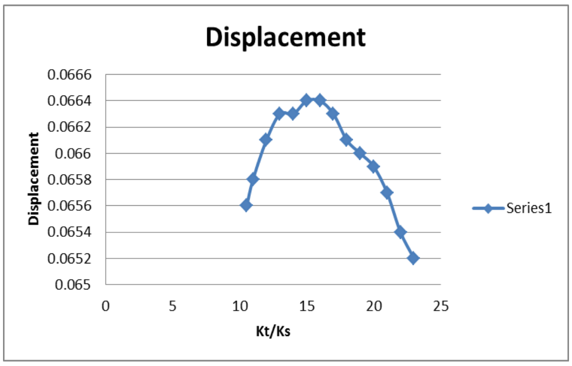

Fig. 7. Displacement vs Kt/Ks

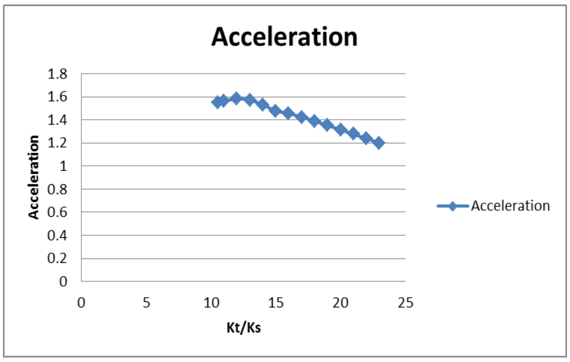

Fig. 8. Acceleration vs Kt/Ks

From the above two graphs optimum value of Kt/Ks is determined. When Kt/Ks is 23, the sprung mass displacement and acceleration obtained are minimum so Kt/Ks value is taken as 23. The list of optimized parameters obtained from the mathematical modeling are shown in table IV.

To obtain the optimum value for damping coefficient, Kt/Ks is taken as 23 and by changing the damping values sprung mass deflection and acceleration are obtained. The graphs for deflection and acceleration at constant Kt/Ks value are shown below.

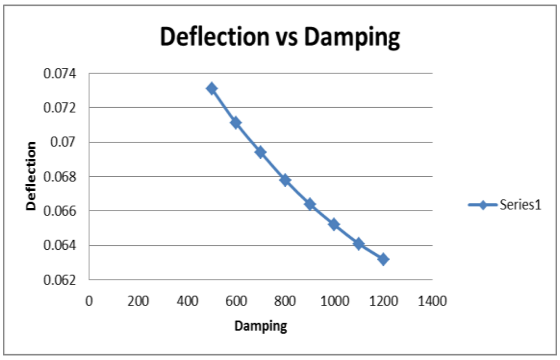

Fig. 9. Deflection vs Damping

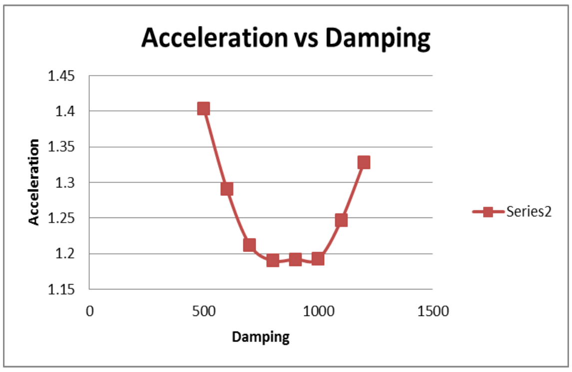

Fig. 10. Acceleration vs Damping

The above graphs are plotted for the damping values from 500Ns/m to 1200Ns/m. From the graphs it is observed that as damping increases deflection decreases. But in the acceleration, up to 800Ns/m damping value it decreases and beyond 800Ns/m its value starts increasing. So 800 Ns/m is selected as the optimum damping value.

Therefore, from the SIMULINK program the optimum values obtained for this suspension system are Kt/Ks = 23 and damping coefficient is 800 Ns/m.

TABLE IV. OPTIMUM PARAMETERS

Optimum Parameters	
Sprung Mass (Kg)	250
Unsprung Mass (Kg)	50
Kt/Ks	23
Tire Stiffness (N/m)	196000
Damping Coefficeint (Ns/m)	800

The obtained values of the damping coefficient and spring stiffness from the mathematical modelling is coupled with the static geometry design of the trailing arm suspension and is analyzed in LOTUS suspension analyzer. Figures 11-13, shows the model in generated in Lotus suspension Analyzer.

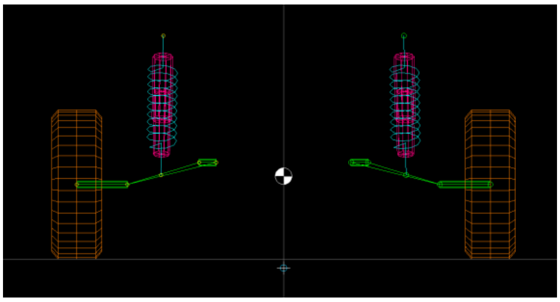

Fig. 11. Front view of the geometry in Lotus analyzer

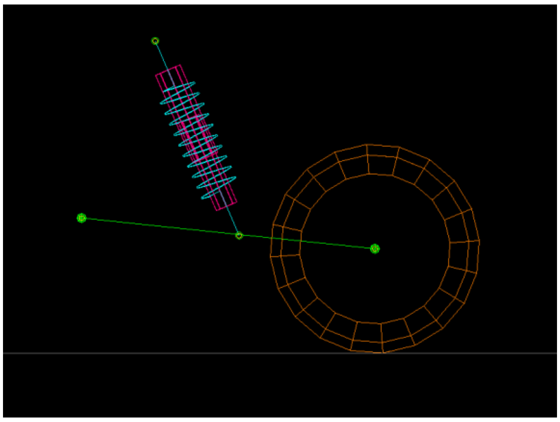

Fig. 12. Side view of the geometry in Lotus analyzer

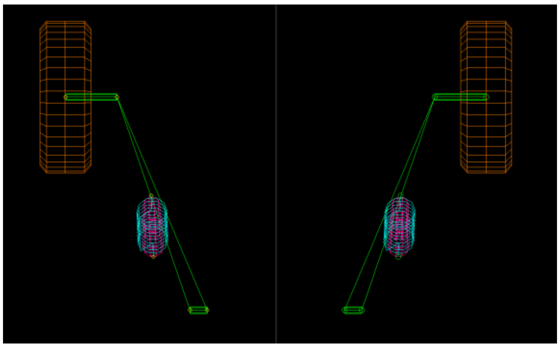

Fig. 13. Top view of the geometry in the Lotus analyzer

The generated model was run in LOUTS with the input parameters and the following results were obtained.

TABLE V. INCREMENTAL SUSPENSION PARAMETER VALUES

Bump Travel (mm)	**Roll Center Height (mm)**	**Wheel base Change (mm)**
60.00	0.03	3.44
40.00	-66.33	2.93
20.00	-46.33	1.78
0	-26.33	0
-20.00	-6.33	9.36
-40.00	-0.94	18.65
-60.00	-1.88	27.87

- Camber Angle change: 0 deg.
- Toe angle change: 0 deg.

V. CONCLUSION

In the present work optimum values for the suspension stiffness and damping are obtained for good comfort and handling conditions. From results it is observed that soft spring gives good comfort i.e. less sprung mass displacement and acceleration. Very less and very high damping values gives more acceleration. So optimum value is selected which gives minimum sprung mass acceleration.

All the obtained suspension parameters are checked by the LOTUS suspension analyzer for kinematic analysis. For the calculated values and it is observed that there is no change in the camber and toe angle values, and even change in wheel base is also within the desired limit.

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