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Modeling and Simulation of Under-Seat Suspension System for Electric Vehicles

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***Abstract*—Currently there is a requirement of innovation in the field of E-Mobility in the society, For the purpose of long distance touring, trucking and public transportation units. So it is a very difficult for the designers to design an electric bus for all kind of roads in India. As the road conditions are not up to the mark in India so it is quite difficult for the designers to optimize an electric vehicle**

Safety and comfort are of the utmost importance to buses (both Combustion and electric), truck drivers, as many of them are traveling across the country in the trucks and buses they are operating. Many drivers work at a rate of pay that depends on the number of miles they travel in a given shift.

This can attribute to long periods of time behind the wheel, making it important for the driver to feel comfortable and safe at all times. This makes it pertinent for Auto-makers to design vehicles that are comfortable enough to allow extended driving sessions.

Thus, an under the seat suspension system is an essential requirement to ensure that they have a safe and sound ride while commuting over long distances. The main basis of this system is a push rod type suspension geometry which accounts for all the jerks and excruciations he has to face while driving. The push rod system increases weight by an ounce whereas it increases comfort by a mile. In addition to this, the design is highly cost-efficient and uses a single spring damper system to address the weight issue.

Keywords*— *Seat, suspension, road, dampers, pushrods, rocker-arms, spring, vibrations, vehicles, force.

I. INTRODUCTION

When the vehicle is in motion it is subjected to different vibrational impact which is in a range of 5-20 Hz. As a result of which it affects the physical health of the driver. An essential part of the improvement is the driver's seat, so research of seat suspension systems has become the focus of attention in all automotive and truck companies because the driver's body is connected to the vehicle by the seats and all of the vibration is transmitted to the driver's body through the seats.

There have various advancements in the vehicle suspension system and ergonomics. Various studies have been conducted on the seat suspension design based on the road conditions or vehicle dynamics. In [8] a review has been made on seat suspension system, where it is discussed about the development in different seat suspension with state of the art vibration control systems. Also the pros and cons of the vibration systems are also discussed. In [1] an optimization technique called Genetic algorithm has been used in passive suspension system to absorb vibrations as per ISO 2631-1:1997 standards. In [11] the health risks associated with low-frequency whole-body vibrations. In [2] an integrated active and semi-active seat suspension is designed for heavy duty vehicles. In [10] a study was conducted to discuss the effects of mounting a computer-controlled actuator in parallel with the traditional spring–damper assembly. In [9] a vibration isolation model has been designed and fabricated for improving vibration isolation effectiveness of the vehicle seat under low excitation frequencies. In [4] rotorcraft design environment is integrated with different occupant biodynamic modeling techniques and resulting differences in comfort assessment is investigated. In [5] a new type of methodology is presented for vehicles driven on long-span bridges, taking into account the traffic and environmental loads such as wind excitations. In [6] a study is conducted on the control of roll vibration and vertical vibration of a seat suspension due to uneven road under both sides of tires. In [7] a 5-degree-of-freedom driver and seating system model is designed for optimal vibration control, by measuring experimental vibration a new method is developed for identification of the driver seating system parameters. In [3] an active seat system is proposed for reducing the vibration level transmitted to the seat pan and the occupants' body under low frequency periodic excitation.

In the current paper the main requirement is to design a seat suspension system which would improve the driving experience for long distance journeys and for long driving hours on rough roads by optimizing the vehicles vibrational impact under dynamic conditions that responds to the driver's bio dynamics.

In order to lower the vertical vibration acceleration, felt by the human body, a seat suspension system is required. Automotive seats must provide drivers with a broad range of sizes with seating over relatively long periods of time and isolate vehicle vibration and shock transmitted to the drivers.

heavy-duty vehicles normally suffer from high levels of vertical vibration in the frequency range of 1–20 Hz, especially from 3 to 5 Hz and hence, passive suspension could amplify vibration in the low frequency range so that ride comfort is affected. Thus we came up with an idea of an under-seat active suspension system that would absorb medium as well as high frequency vibrations and provide the driver with the requisite comfort and smoothened driving condition. Alongside with this the system should be completely adjustable according to the requirement of the driver as well as the road conditions. The stiffness can be adjusted along with the length of the push rods.

Here the dampers are in the static loading condition when the driver is seated and the load transfer varies in accordance to the variation in road conditions under the influence of forces from the body of the bus under the dynamic loading conditions

A Push rod suspension system has been used, the three major components of this system are the pushrods, the bell cranks and the dampers/ shock absorbers. Here the rocker arms are placed at the highest point in the assembly. as such, the push rod is under pressure as it transfers compression force upward into the rocker arm.

II. NOTATIONS

F_p	*Load on the Pushrod*
F_s	*Force on the Spring*
D	*Wire diameter of spring*
D_m	*Mean diameter of Spring*
N	*No. of coils*
L_f	*Free length of spring*
C	*Damping Coefficient*
Cc	*Coefficient of critical damping Coefficient*
Wn	*Natural Frequency*
K	*Spring stiffness*
M	*Mass of system*
G	*Modulus of rigidity*

III. SELECTION OF GEOMETRY(FOR SEAT SUSPENSION)

From the various types of available geometry both push-rod and pull-rod types of geometry are the most convenient types of geometry used for the purpose.

- Push-rod and pull-rod suspension are similar yet distinct in design, with the main difference being the placement of the rocker arm that controls shock damping in relation to the upper control arm.
- In effect, this means that both push-rod and pull-rod systems are functionally the same design.

As such, push-rod suspension systems allow for much greater high-speed stability, much lower levels of body-roll, and a much lower center of gravity for the vehicle.

- For pull-rod suspension systems, the only difference is the orientation of the rocker arms. In a push-rod system, the rocker arms are placed at the highest point in the assembly. As such, the rod is under pressure as it transfers compression forces upwards into the rocker arms.
- In a pull-rod system however, the rocker arms are located between the upper and lower control arms, at the centre of the assembly. As such, the rod is under tension as it pulls against the rocker arms.

As a result of these factors, the push-rod layout is distinct from other suspension systems as, unlike others, it is able to be designed and assembled with components closer to, or further from, the centre of gravity of the vehicle. As a result, engineers are able to optimize the performance of their vehicle in this area as they sacrifice comfort and practicality in favour of aerodynamics, handling, and stability on track.

These are some of the factors which resulted in the selection of pushrod type geometry for the above project.

IV. SELECTION OF MATERIAL

For Bell Cranks: Aluminium 6061 T6

T6 temper 6061 has -

• Ultimate tensile strength of at least 290 MPa (42,000 psi)
• Yield strength of at least 240 MPa (35,000 psi).
• More typical values are 310 MPa (45 ksi) and 270 MPa (39 ksi), respectively.

Young's modulus (E): 68.9 GPa (9,990 ksi)
Elongation (ε) at break: 12–25%
Density (ρ): 2.70 g/cm3
Thermal conductivity (k): 151–202 W/(m•K).

PROPERTIES;

• Resistance to corrosion
• A high value of hardness
• Easy machinability
• Rigidity and toughness
• Light Weight with high yield strength
• Good weld ability

For Pushrods and Mountings: AISI 4130 Chromoly Steel 19*1 mm

Taking reference from [12]

Mechanical Properties:

PROPERTIES	METRIC
Tensile strength, ultimate	560 MPa
Tensile strength, yield	460 MPa
Modulus of elasticity	190-210 GPa
Bulk modulus (Typical for steel)	140 GPa
Shear modulus (Typical for steel)	80 GPa
Poissons ratio	0.27-0.30
Elongation at break (in 50 mm)	21.50%
Reduction of area	59.6
Hardness, Brinell	217
Hardness,Knoop (Converted from Brinell hardness)	240
Hardness, Rockwell B (Converted from Brinell hardness)	95
Hardness, Rockwell C (Converted from Brinell hardness, value below normal HRC range, for comparison purposes only.)	17
Hardness, Vickers (Converted from Brinell hardness)	228
Machinability (Annealed and cold drawn. Based on 100% machinability for AISI 1212 steel.)	70

Thermal Properties:
Thermal conductivity (100°C) - 42.7 W/mk

Properties

• Easy machinability
• Good atmospheric corrosion resistance
• Reasonable strength
• High Strength to weight ratio

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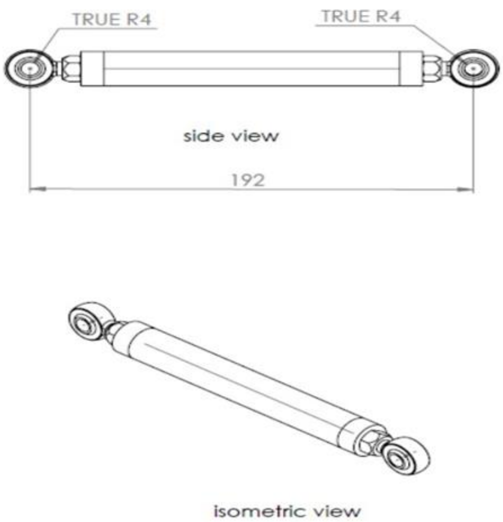
The spring damper system is an OEM part from DNM springs. The model used is Burner RCP- 2S with a stiffness(k)=140 pounds/inch.

V. DESIGN PROCEDURE:

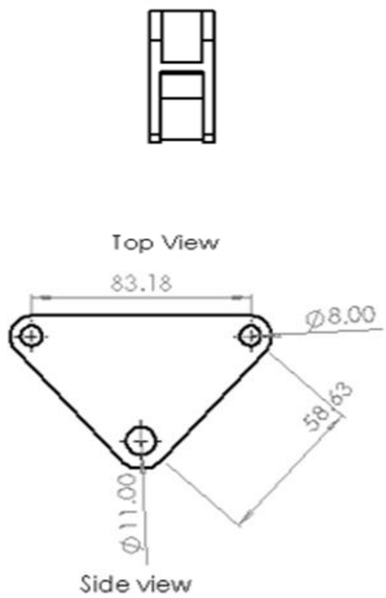
- A standard seat model was taken.
- The suspension geometry was decided.
- A spring-damper system was selected instead of regular springs to dampen the active vibrations.
- All the calculations for the forces were done.

A. DESIGN OF COMPONENTS AND ASSEMBLY



dimensions are in mm

Fig. 1: Pushrod



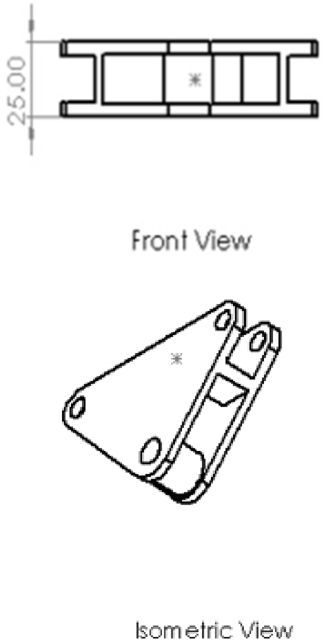
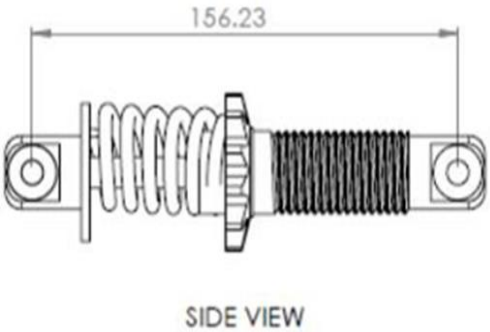


Fig. 2: Bell Crank



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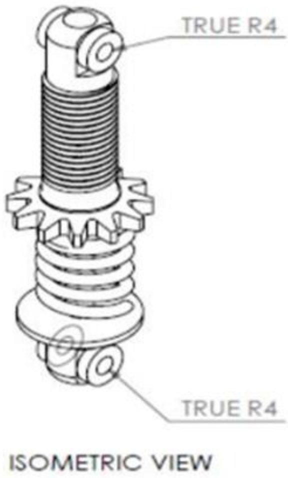

Fig. 3: Shock Absorber

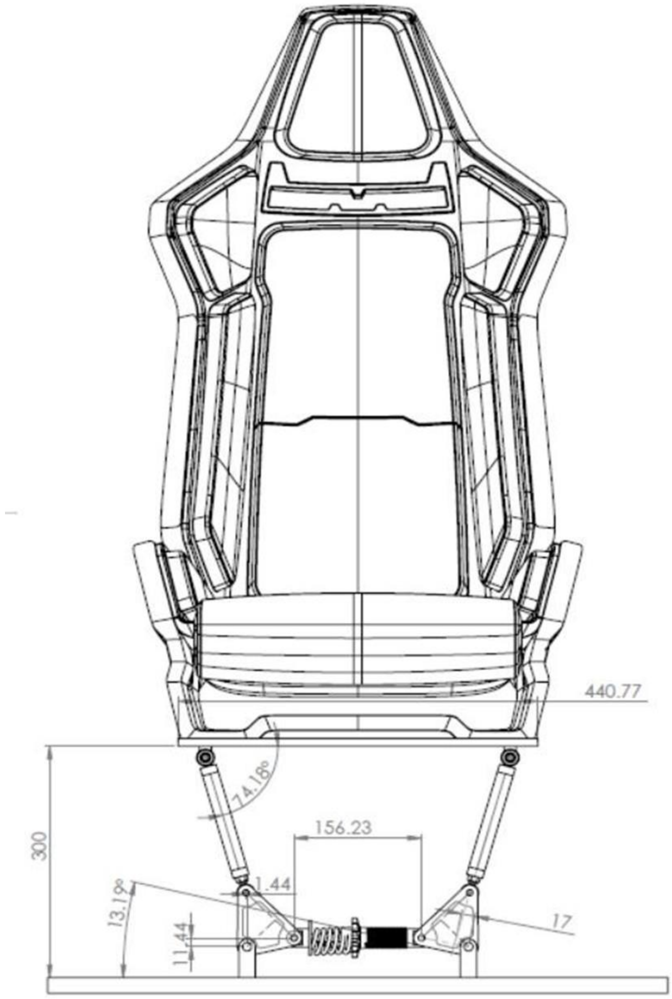

Fig. 4 : Seat suspension arrangement

Fig. 5: Isometric View

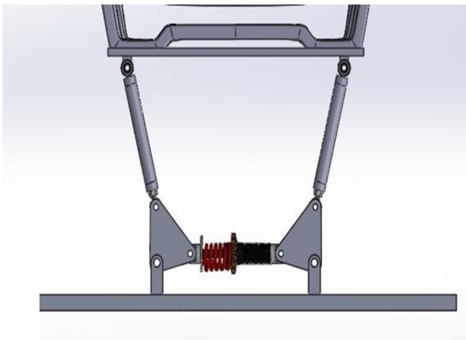

Fig. 6: Base set View

Fig. 7: Front View (full)

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VI. RESULTS AND DISSCUSSIONS

ANALYSIS REPORT:

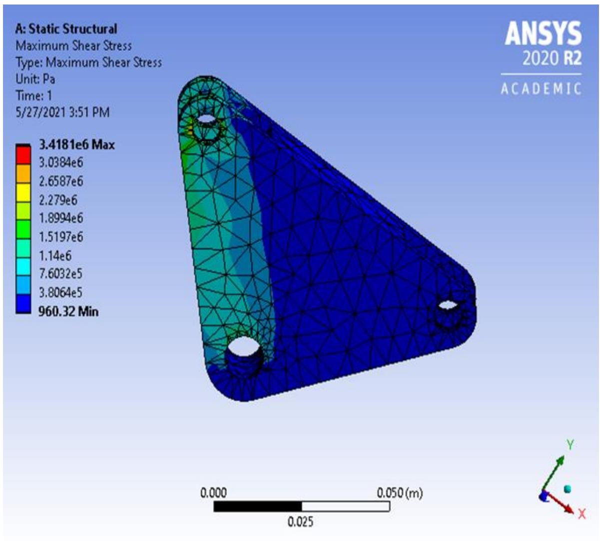

Fig. 8: Maximum Shear stress

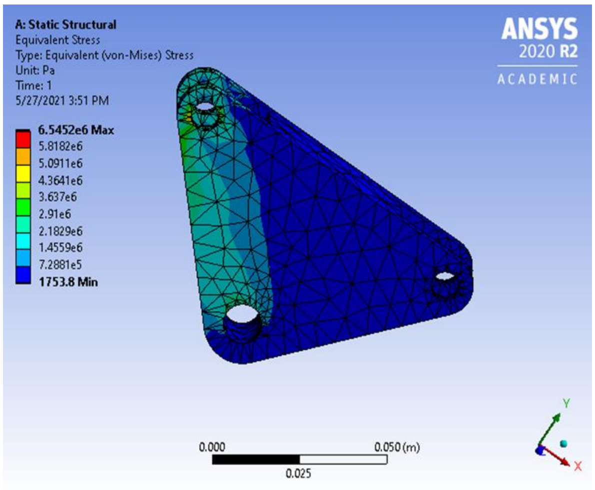

Fig. 9: Equivalent stress

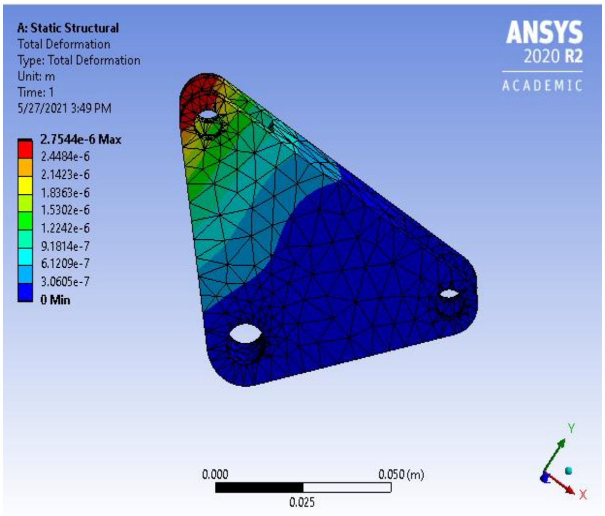

Fig. 10: Total Deformation

VII. CALCULATIONS

Weight of an average Driver = 78 kg
Weight of the cushioned seat = 2 kg
Total Weight = 80 kg
In a seating position the total weight of the driver is taken as 73% of the total weight,
Driver Weight = 57 kg
Now, total weight acting on the setup = 59 kg
Vertical Load (P1) = (59*9.8)/2 = 289.1 N
Load on Pushrod (Fp) = 289.1* Cos (15.82) = 278.14 N,

Now to find the spring force (Fs) we have to balance the moments about the pivot point of the bell crank,

Fs * 11.44 = Fp * 17,

Fs = (278.14*17)/11.44,

Fs = 413.31 N

K = 140Psi = 25 N/m

Wn = √ (K/m)

=√ (25/80)

=0.55

Cc = 2mWn

=88 Ns/mm

We have to keep the damping condition as C < Cc, for Under damping condition.

K = Gd4/8D3

[K = 25N/mm (given), G = 2.1*105 MPa, C = d/D = 6 (assume), n = 6 (assume)]

25 = 2.1*105d4/8*216*6

- d = 5mm
- D=30mm

Free length of spring, Lf = nd+ (n-1)
= 6*5 + 5 = 35mm

X_{st} = Compression in the spring in static condition

= Fs/K

=413.31/25= 16.53 mm

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