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Design and Finite Element Analysis of Suspension Systems for High-speed Vehicles

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Abstract

Suspension is one of the most important sub-systems of vehicles which plays a key role in the performance and handling of the vehicle. In this research paper, the design and optimization of the suspension system for a high-speed vehicle are discussed. The necessary parameters of geometry like roll rate, ride rate, load transfer, natural frequency, etc., are calculated based on given constraints for gaining desired performance. Kinematic analysis is carried out to find the dynamic parameters like camber gain, instantaneous center movement, toe change, etc. Computer aided design (CAD) modeling and material selection for control arms are very crucial as these are the primary structures responsible for transmitting force from ground to chassis. A case study is conducted to show the possible advantages of nanocomposites. The finite element analysis (FEA) is performed for control arms based on calculated force to attain the best strength-to-weight ratio. The factor of safety, equivalent stress, and deformation is obtained within the prescribed limit as per FEA.

Keywords

Geometric parameters, Motion geometry, Kinematic analysis, Structural analysis

Introduction

Suspension is the system that connects the un-sprung mass of the vehicle to the sprung mass. The motion of the wheel is constrained by the suspension system caused by uneven road surfaces. Any shocks or vibrations of the road are damped by the suspension and then passed to the chassis. The suspension system includes connecting linkages namely control arms, rockers, and shock absorbers or a coil over. The main aim of suspension geometry is to get maximum traction at any given condition and increase the drivability of the vehicle.

Chepkasov et al. [1] developed a mathematical model for the kinematic analysis of a double wishbone suspension system for a formula Society of Automotive Engineers car. The illustration of force transmission through a different component such as rocker's arms, and linkages were used to define the travel of the suspension and to perform kinetic analysis, considering all suspension components as rigid. The initial parameters for the calculation were taken as 2950 mm, 1530 mm, and 475 mm for wheelbase, track width, and center of gravity, respectively, whereas weight distribution for the static condition was taken as 45:55 for front to rear [2]. Levy and Poter [3] performed the dynamic analysis of the suspension geometry and found the values of dynamic parameters like camber change and toe change while bump and roll along with the instantaneous center moment. Saurabh et al. [4] showed the results of vibration analysis and roll steer analysis for the designed suspension system. The spring design method and the

FEA of the arms were also elaborated for safe design. Olsen et al. [5] described various methods to manufacture carbon fiber composite arms and performed analysis to determine the strength. Kumar et al. [6] carried out the kinematic and structural analysis on the suspension system with an anti-roll bar for better cornering performance and to manipulate the response of the vehicle mainly understeer and oversteer according to the driver feedback.

The present research paper shows the design methodology, calculation of various parameters, kinematic analysis, CAD modeling, and FEA for various suspension components of the high-speed vehicle for the formula Bharat competition. The total 50 mm wheel travel, 25 mm jounce, 25 mm rebound, and 30 mm minimum ground clearance are considered per the competition's rulebook [7]. The main objective is to get the best dynamic stability and keep the weight as minimal as possible without compromising the strength of structural members. The factors considered while designing are vehicle dynamics, overall natural frequency, ride rate, roll stiffness, kinetic behavior of the vehicle, stress and fatigue effects, and manufacturing constraints. The automobile sector can benefit by employing various nanocomposites, namely metal matrix nanocomposites, polymer matrix nanocomposites, and ceramic matrix nanocomposites. The attainment of a competitive edge for a high-speed vehicle can be facilitated by the reduction in weight of any subsystem. Therefore, the utilization of aluminum metal matrix nanocomposites (AMMCs) can yield significant advantages for automotive applications [8].

Methodology

The track width and wheelbase have been decided based on track conditions and steering geometry. The design has been carried out for vehicles weighing around 250 kg with 45 - 55 % weight distribution in front and rear, respectively. The engine is to be mounted in the rear, so the weight distribution is considered more to the rear end. For the front suspension system, a double wishbone with a direct-acting connection on coil-over is selected because of the manufacturing limitation of incorporating a push rod. In the rear, a double wishbone with a pushrod attached to coil over via the rocker's arm is selected because of the ease of design and lighter components involved [9].

Static and motion geometry

For the primary iteration, the base value of parameters like scrub radius, king pin inclination, camber, and castor is decided using past data suggested by Farrington [2]. 2D geometry for the roll center and instantaneous center is sketched followed by the construction of 3D static geometry for arms, toe link, and tie rod to look over the manufacturability and location of the inboard points i.e. mounting of the control arms towards the chassis.

A motion geometry is constructed which is a simulation of the actual motion of the linkages while the wheel undergoes any vertical motion caused due to uneven track surface with the help of sketched static geometry. In the motion geometry, parameters such as pushrod angle, pushrod length, rocker arms dimensions, direct mount length, motion ratio, etc., and other geometric parameters are iterated to find the geometric spring compression or actual physical spring compression. This compression is to be compared with the theoretical spring compression for that certain set of values of geometric parameters.

Utilizing force and moment balance around the pivot of the lower control arm, the front theoretical compression is computed. One end of the control arm is bolted to the chassis and therefore the arm is constrained to pivot about it. Motion ratio can be achieved depending on the ratio of the arm's length and the length from the pivot to the spring mount. The force and moment balance equation are used to determine the spring compression based on the load on one wheel i.e., using weight distribution for a specific spring angle mounted at a specific location on the arm. The theoretical values for various spring compression on a fixed set of motion ratios and spring angles are found using this concept. Iterative values for the magnitude of the force that is acting on the spring on different motion ratio at different spring angle is given in table 1.

The rear theoretical compression is calculated using moment balance about the rocker arm pivot instead of the control arms pivot as the load acting on the wheel will be

Table 1: Front initial compression calculation for various motion ratios and spring angle.

Force on the wheel due to whole mass (A)	Cos side view angle (B)	A * B	Spring angle with vertical	Geometric motion ratio/compression force in the spring direction x 10^{-3} (N/m)				
				0.8	0.85	0.9	0.95	1
60	0.9945	59.6716	0	74.59	70.20	66.30	62.81	59.67
60	0.9945	59.6716	1	75.01	70.60	66.67	63.16	60.01
60	0.9945	59.6716	2	75.45	71.02	67.07	63.54	60.36
60	0.9945	59.6716	4	76.43	71.94	67.94	64.37	61.15
60	0.9945	59.6716	5	76.97	72.44	68.42	64.82	61.58
60	0.9945	59.6716	6	77.54	72.97	68.92	65.29	62.03
60	0.9945	59.6716	7	78.13	73.54	69.45	65.80	62.51
60	0.9945	59.6716	8	78.77	74.13	70.01	66.33	63.01
60	0.9945	59.6716	9	79.43	74.76	70.61	66.89	63.55
60	0.9945	59.6716	10	80.14	75.42	71.23	67.48	64.11

transmitted via push rod to the rocker's arm. The force and moment are balanced by the spring force about the rocker arm pivot. Using the same idea, a MATLAB code has been formulated to find theoretical spring compression. The input parameters for the code include the pushrod angle, load on one wheel, spring stiffness, etc. and the output parameter is the spring compression at the particular rocker arm ratio. Thus, the theoretical compression for the rear is obtained using the code.

When the actual and theoretical values are within five percent error, the geometry is verified for any interference between the parts like control arms and rim. The parameters like roll rate, natural frequency, wheel rate, etc. are calculated and the kinematic analysis is carried out after verification of the geometry for the interference.

Calculation of kinematic parameters for the suspension system

Various kinematic parameters like ride rate, roll rate, longitudinal and lateral load transfer, and natural frequency have been obtained for the proposed geometry using assumed mass. The spring ratio, motion ratio, and wheel ratio have been calculated using equation 1 to equation 3 [9].

The calculations of roll rate, ride rate, load transfer, and natural frequency have been calculated using the formulas 1 to formula 8 [10, 11].

$$K_s = \frac{F}{l} \tag{1}$$

Where F = Force acting on spring (N), l = Spring displacement (m), K_s = spring ratio (N/m).

$$MR = \frac{Springe\ travel}{Wheel\ travel} \tag{2}$$

$$K_w = K_s(MR)^2 \tag{3}$$

Where K_s = Spring ratio (N/m), MR = Motion ratio, and K_w = Wheel ratio (N/m).

The natural frequency of the sprung mass and un-sprung mass have been calculated using equation 4 and equation 5.

$$f_s = \frac{1}{2\pi}\sqrt{\frac{(K_w K_T)/(K_w + K_T)}{M_s}} \tag{4}$$

$$f_{us} = \frac{1}{2\pi}\sqrt{\frac{K_w + K_T}{M_{us}}} \tag{5}$$

Where M_{us} = Un-sprung mass (kg), f_{us} = Un-sprung mass natural frequency (Hz), f_s = Sprung mass natural frequency (Hz), K_T = Tire spring ratio (N/m), and M_s = Sprung mass (kg).

Front roll rate and rear roll rate are calculated using equation 6 and equation 7, respectively.

$$K_{\phi F} = \frac{\pi(t_f^2)K_{LF}K_{RF}}{180(K_{LF} + K_{RF})} \tag{6}$$

$$K\phi R = \frac{\pi(t_r^2)K_{LR}K_{RR}}{180(K_{LR} + K_{RR})} \tag{7}$$

Where $K_{\phi F}$ = Front roll rate (Nm/deg), $K_{\phi F}$ = Rear roll rate (Nm/deg), t_f = Front track width (m), t_r = Rear track width (m), K_{LF} = Left front wheel rate (N/m), K_{RF} = Right front wheel rate (N/m), K_{LR} = Left rear wheel rate (N/m), and K_{RR} = Right rear wheel rate (N/m).

The roll gradient is calculated using equation 8.

$$\frac{\phi_r}{A_y} = \frac{M * H}{K_{\phi F} + K_{\phi R}} \tag{8}$$

Where A_y = Lateral acceleration (g), H = Distance from roll center to center of gravity (m), M = Mass of the car (kg), and $\frac{\phi_r}{A_y}$ = Roll gradient (deg/g).

The optimum value for the stability of the vehicle and driver comfort has been achieved using several iterations in the kinematic analysis calculation. Optimum values for ride rate, roll rate, load transfer, wheel rate, etc., are achieved by modifying stiffness or motion ratio for the static geometry. The kinematic analysis is performed on these suspension parameters to obtain dynamic values of parameters like dynamic camber, toe change, and roll center migration. These iterations have been performed by revising geometry for a different set of combinations of roll center, caster, camber, motion ratio, Instantaneous center values, etc. The iteration for front and rear suspension geometry has been shown in figure 1 and figure 2, respectively. Here, a_1 and a_2 are compressed spring lengths, b_1 and b_2 are direct attachment lengths, c_1 and c_2 are upper control link lengths, d_1 is spring mount distance from pivot whereas d_2 is lower control link length, e1 and e_2 are kingpins or steering axis, and f_2 is rocker arm length.

Kinematic analysis of geometry

The kinematic analysis of the geometry is carried out in Lotus Shark suspension analysis software. The outboard and inboard points of the 3D geometry of suspension along with the coil over parameters like spring rate, the height of chamber gain, and other input parameters are fed in the software. Dynamic loading conditions and dynamic suspension parameters like camber gain, toe change, roll, and instantaneous center movement have been obtained and further analyzed. Figure 3 to figure 7 have been obtained after carrying out the kinematic analysis of the geometry.

Results and Discussion

The final suspension parameters shown in table 2 and the inboard points have been finalized after carrying out several iterations in Lotus Shark. The values of camber change and toe change in figure 3 to figure 7 have been compared to the suspension system given by Levy and Potter [3]. The dynamic parameters are within the permissible range that is one-degree change per one-inch travel in bump and one-degree change per degree of the roll while in roll and a bump steer value within 0.27 degrees along with all the other desired parameters, hence the geometry is finalized.

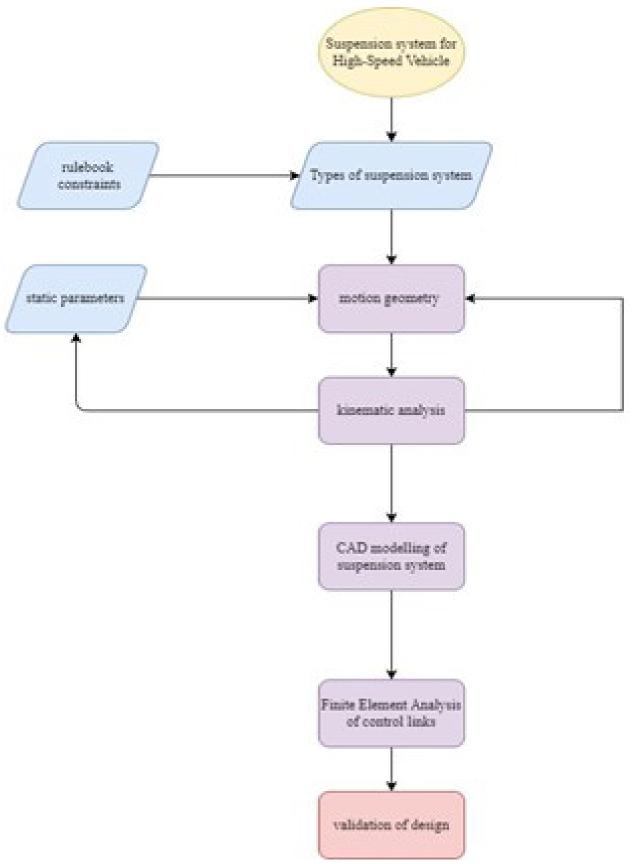

Figure 1: Process flowchart.

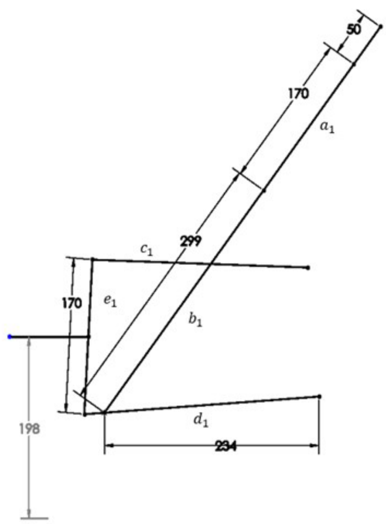

Figure 2: Front iteration motion ratio 0.9.

CAD modeling of arms

The endpoints of control links have been obtained from the finalized suspension geometry. For assembling the rod ends for the front and rear arms at the inboard side, a bush is used having 8 mm metric threading. For the front, the arms bush is used at the outboard side, and for the rear 5 mm plate with a ball joint pressed in it is used. The CAD model for the same is prepared in SolidWorks 3D CAD software and the main dimensions for the front upper arms are 48 mm length of bush (outboard side), 25 mm length of bush (inboard side), 173- and 137-mm pipe lengths whereas the main dimensions for the front lower arms are 60 mm length of bush (outboard side), 25 mm length of bush (inboard side), 169 and 150 mm pipe lengths. The main dimensions for the rear upper arms are 25 mm length of bush (inboard side) and 236- and 239-mm pipe lengths whereas for the rear upper arms are 25 mm length of bush (inboard side) and 247- and 255-mm pipe lengths (Figure 8 and figure 9).

Table 2: Final suspension parameters.

Description	Front	Rear
Curb weight	86 kg	106 kg
Scrub eadius	25 mm	35 mm
Camber	2	0
Castor	7	7
King pin inclination	3	0
Roll center	80 mm	90 mm
Camber gain (in the roll) deg/deg	0.99	0.92
Camber gain (in bump) deg/in	0.98	0.78
Motion ratio	0.9	0.92
Roll rate	8198.28 Nm/rad	7426.12 Nm/rad

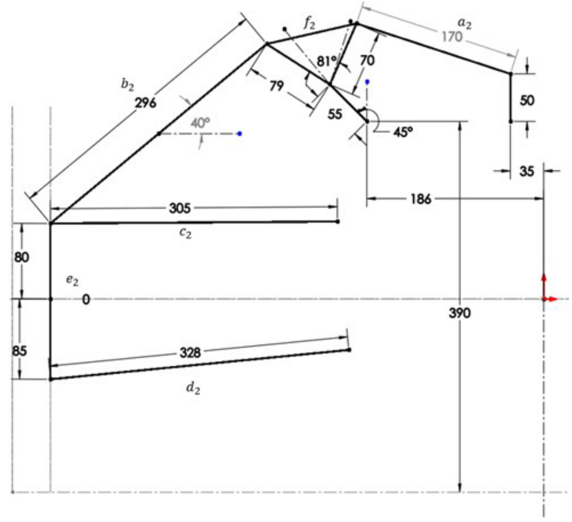

Figure 3: Rear iteration with motion ratio 0.92 and rocker angle of 81°.

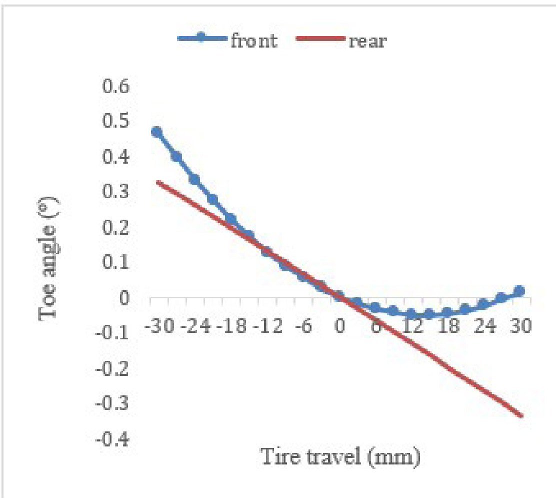

Figure 4: Toe change in bump.

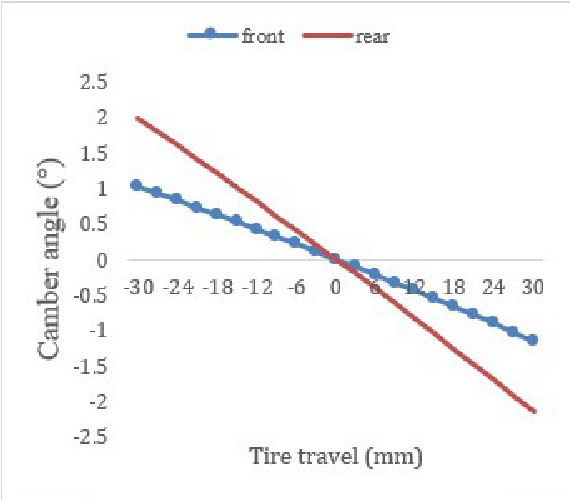

Figure 5: Camber change in bump.

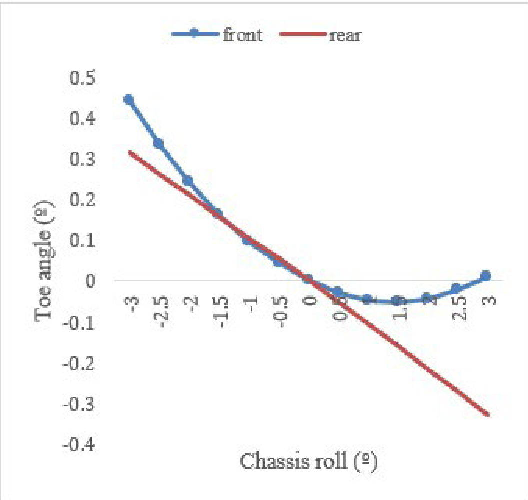

Figure 6: Toe change in roll.

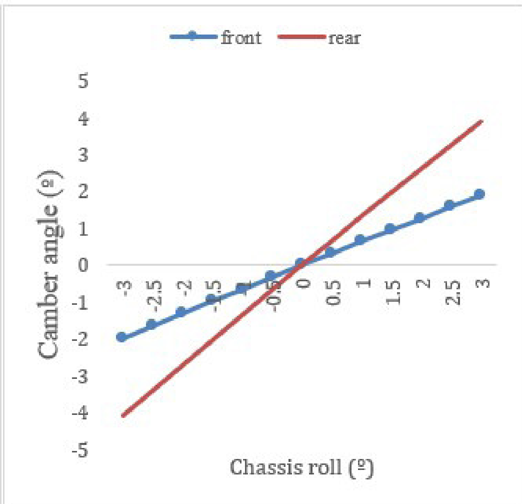

Figure 7: Camber change in roll.

FEA of the system

The FEA has been carried out for structural members like front and rear A-arms and rocker arms to obtain the best combination of weight and strength using the ANSYS workbench. For the control arms, AISI 4130 material was chosen due to its situated ultimate tensile strength and favorable strength-to-weight ratio. Similarly, aluminum 7075 T6 was selected for the rocker arm. To demonstrate the potential advantages of nanoparticles in aluminum for enhanced weight and strength, an ANSYS simulation was conducted. This simulation considered the mechanical properties as recommended in earlier studies.

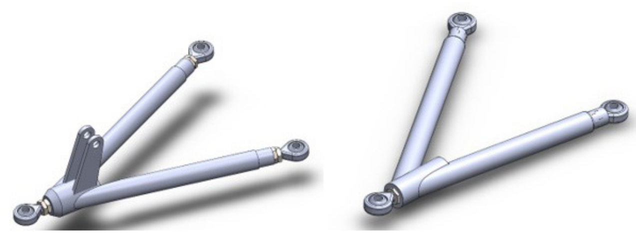

Figure 8: Front control links.

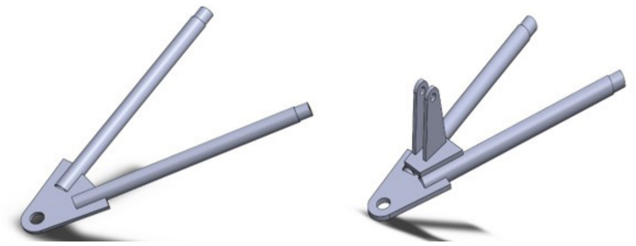

Figure 9: Rear control links.

The Young's modulus, yield strength, and ultimate tensile strength of metal matrix composite with 1.5 wt.% carbon nanotubes were 75.9 GPa, 608 MPa, and 674 MPa, respectively; the corresponding values for pure aluminum were 72.9 GPa, 394 MPa, and 420 Mpa, respectively [12]. AISI 4130 material has a density of 7850 kg/m^3 and aluminum has an approximate density of 2710 kg/m^3 thus can reduce the weight of the subsystem to one-third.

Meshing, forces, and boundary condition

Mesh generation, providing appropriate supports like revolute or fixed, magnitude and direction of the forces acting are the important aspects for FEA. An accurate quadratic mesh with an element size of 1 mm has been preferred for analysis. As the linkages are tested in extreme conditions, fixed support is given on the inner side of inboard pivot points which is the condition when the suspension is in fully rebound condition and the force is applied on it. The forces acting on each component in a different direction have been obtained from force balance at maximum spring compression conditions as given in table 3. For the rocker arm the extreme condition of fixing the pivot and full spring compression force that is 2000 N is applied and results are obtained.

After carrying out the analysis the values of equivalent stress, factor of safety, and deformation have been obtained (Table 4 and table 5) and the thickness of 1.65 mm is finalized. Any FOS under 2 is considered safe as the values of the force are known (Figure 10 to figure 14) [6].

Table 3: Forces on different points in the suspension system.

	Bump force (N)	Braking force (N)	Axial force (N)	The force from spring (N)
Front upper A-arms	2220	1200	2500	-
Front lower A-arms	2220	1341	2700	2561
Rear upper A-arms	2220	1131	2900	2023
Rear lower A-arms	2220	1321	3200	-

Table 4: Results table of FEA of AISI 4130.

Particulars	FOS	Equivalent stress (MPa)	Deformation (mm)
Front upper arms	1.48	505.21	0.46
Front lower arms	1.56	481.4	0.65
Rear upper arms	1.23	605.14	0.87
Rear lower arms	1.25	557.59	0.91
Rocker arm	2.4	240.73	0.9

Table 5: Results table of FEA for AMMCs.

Particulars	FOS in AMMCs	Equivalent stress (MPa)	Deformation (mm)
Front upper arms	1.21	505.21	0.46
Front lower arms	1.23	481.3	0.65
Rear upper arms	1.01	605.14	0.87
Rear lower arms	1.10	557.59	0.91

Conclusion

The suspension parameters obtained through iterative design in SolidWorks have been successfully analyzed using Lotus Software for kinematic analysis of the geometry. Furthermore, FEA of the control arms has provided insights into the structural integrity of the suspension system. Modern CAD software and simulation tools have enabled us to optimize the suspension system for performance and safety while reducing the need for physical prototypes. However, it is important to note that there are limitations to simulation tools and physical testing is still necessary to verify the accuracy of the simulation results. Overall, the use of CAD, simulation, and FEA has greatly improved the design process for suspension systems in formula student racing and other high-performance applications. The future progression in the manufacturability of nanocomposites is expected to contribute to a significant reduction, estimated at around 66.67%, in the weight of suspension systems.

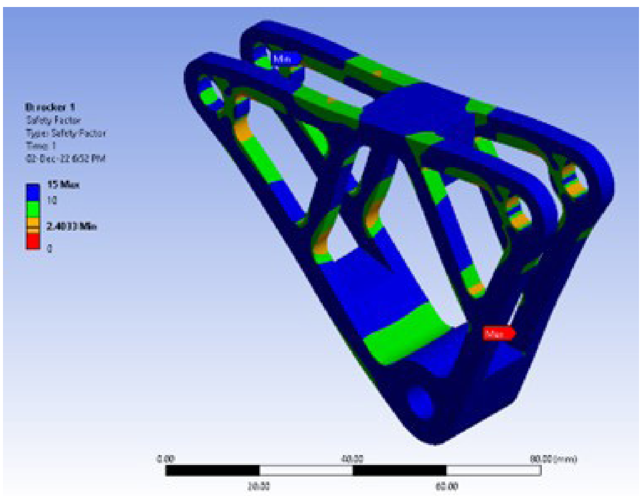

Figure 10: Static structural analysis of rocker arm.

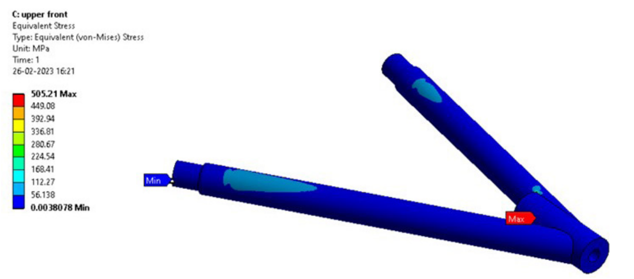

Figure 11: Static structural analysis of front upper A-arm.

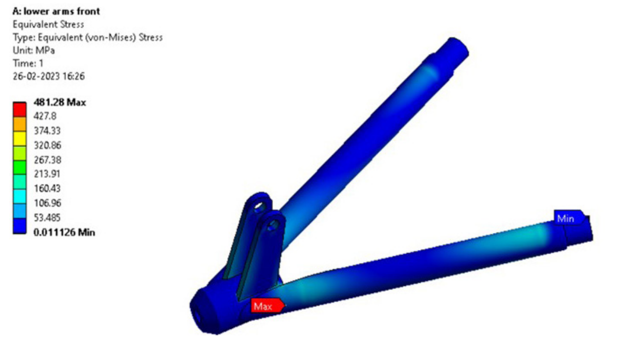

Figure 12: Static structural analysis of front lower A-arm.

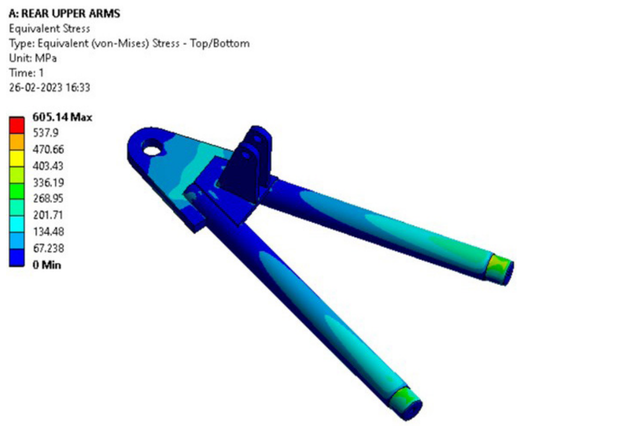

Figure 13: Static structural analysis of rear upper A-arm.

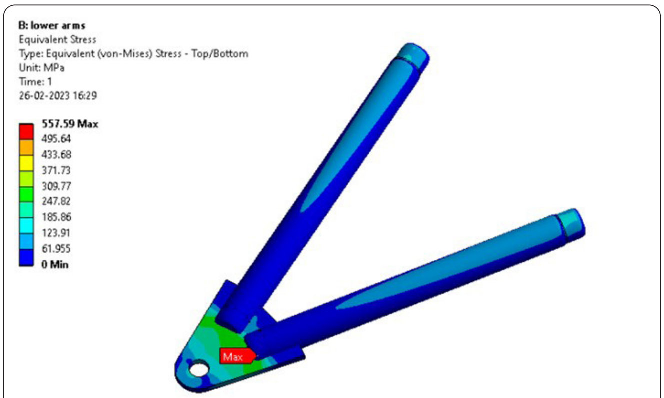

Figure 14: Static structural analysis of rear lower A-arm.

Acknowledgments

None.

Conflict of Interest

None.

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