DESIGN AND ANALYSIS OF DOUBLE WISHBONE SUSPENSION SYSTEM FOR SUV

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Abstract - *Double wishbone suspension system is an independent suspension system which consists of two control arms viz upper control arm and lower control arm. During the actual working condition, the maximum load is transferred from upper wishbone arm to the lower arm which creates possibility of failure in the arm. This paper focuses on the stress strain analysis study of lower wishbone arm to improve and modify the existing design. In this paper, the performance of conventional material and the light weight Metal Matrix Composite (AMC225xe) for SUV's is compared. The 3-D model is designed in Solidworks and the FEA analysis is performed in ANSYS. In this paper, Topology optimization is also performed using ANSYS in order to reduce material usage and reduce the weight of the lower control arm.*

Key Words: **Upper Control Arm, Metal Matrix Composite, Finite Element Analysis, Topology Optimization, SUV's.**

1.INTRODUCTION

Double wishbone suspension offers a smoother driving experience, especially on bumpy roads as it doesn't affect wheel alignment, unlike single wishbone systems, like a MacPherson strut suspension. It also gives technicians flexibility to adjust parameters like camber, caster and toe to meet the requirements of the track or road. In this paper the lower control arm is designed for SUV's using the light weight material and the performance of conventional material and the light weight Metal Matrix Composite (AMC225xe) for SUV's is compared. Topology optimization is also performed using ANSYS in order to reduce material usage and reduce the weight of the lower control arm. The weight reduction helps in increasing the efficiency of vehicle, increase payload, less dead weight, lower fuel consumption, lower emissions and easier handling.

2. RELATED WORKS

Bhushan S. Chakor et al. [1] carried out transient structural analysis of the upper control arm of double wishbone suspension, modal analysis and optimization by selecting better material for weight reduction and better strength. Testing was carried out on Universal testing machine to validate analysis results. In [2], authors have proposed a method to improve handling characteristics of the vehicle by using double wishbone suspension arms which control camber and toe angle in an adaptive manner. In this method, two telescopic arms with actuators change the camber and toe angle of the wheel dynamically in a closed feedback manner. In this mechanism, PID controller are used which trigger the actuators based on the camber and toe angle input received from sensors. The physical quarter car modeling was done in Solidworks and analysis was done in MATLAB. In [3], the authors have designed a front double A-arm push rod suspension system in context to formula student race car. The CAD modeling of the components was done in Solidworks and ANSYS workbench is used for finite element analysis for the same. The authors have performed both kinetic and dynamic analysis of the proposed model. This paper mainly focuses of design and analysis related aspects of the formula racing cars. In [4], design and analysis of double wishbone arm is discussed. The double wishbone suspension arm is designed and analyzed using CATIAV5R20 Software. The dimensions of upper and lower arm were calculated. The roll centre of ATV by geometry of double wishbone suspension system was also calculated. The results show that better performance, reduction in cost and weight of suspension assembly was achieved.

It is observed that the stress strain analysis study of lower wishbone arm is essential to improve and modify the existing design. The material of the lower wishbone can also be optimized to improve the performance of existing design. It is also concluded that the process of the topology optimization which involves the material distribution can reduce the weight of the existing industrial component. Therefore, the weight of control arm can be reduced further using the different materials.

3. OBJECTIVES & METHODOLOGY

The key objectives of this work are as follows:

- To design the lower the control arm in Solidworks.
- To perform the static analysis of lower control arm using conventional material.
- To perform the static analysis of lower control arm with metal matrix composite (AMC225xe, is a high quality aerospace grade Aluminium alloy (AA2124) reinforced with 25% by volume of ultrafine particles of silicon carbide).
- To compare the performance the both materials in terms of total deformation, von-mises strain, von-mises stress
- To perform topology optimization in ANSYS.

The model was first designed in Solidworks, and imported in ANSYS. Then, the static analysis was performed using the conventional material (structural steel) and metal matrix composite. Force acting on the control arm was calculated, the boundary conditions were applied and meshing was done in order to perform the static analysis. At last, topology optimization was carried out in ANSYS. The vehicle specification [5] used are given in table below:

Table 1: Vehicle Specification

Description	Symbol	Value
Total weight of vehicle (N)	W	19620
Width of the vehicle (mm)	b	1855
Wheel base (mm)	L	2450
Height of the vehicle (mm)	h	1844
Front axle track width(mm)	B	1520
Vehicle mass (Kg)	M	2000
Centre of gravity height (mm)	H	922

3.1. Material properties

We have used two materials and the performance of these two materials is also compared. One material is the conventional material (structural material) and metal matrix composite (aluminium alloy with SiC particulates).

Table 2: Material Properties

Mechanical properties	Structural steel	AMC225xe
Young's Modulus (Gpa)	212	115
Density (kg/m^3)	7850	2880
Poisson's Ratio	0.3	0.3
Yield Stress (Mpa)	420	480

3.2. Force Calculation

Weight division for front axle and rear axle is 0.52 and 0.48 respectively.

$G_R = 0.48*G$ where G_R is force acting on rear axle

$G_R = 9417.6$ N

$G_F = 0.52*G$ where G_F is force acting on front axle

$G_F = 10202.4$ N

Calculating the distance of front and rear axle from the center of gravity.

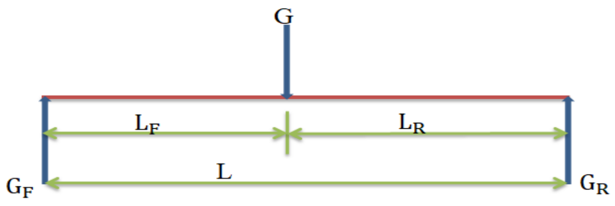

Figure 1: Typical forces acting on vehicle under static loading [1]

Taking moment about G_F

$G*L_F - G_R*L = 0$

$L_F = 1176$ mm

$L_R = L - L_F$

$L_R = 1274$ mm

Table 3: Assumptions made for calculation [6]

Retardation(m/s^2)	a	2
Aerodynamics drag coefficient	C_d	0.32
Air density(kg/m^3)	ρ	1.228
Velocity(kmph)	V	80
Radius of bend(m)	R	100
Coefficient of friction	μ	0.6
Stiffness of the spring(N/mm)	K	50

3.2.1 Longitudinal Force (G_{FA})[7]

$G_{FA} = \frac{\mu}{L}(G*L_R + M*a*H - F_D*H)$,

$F_D = C_d*A*\frac{\rho}{2}*V^2$, where F_D is the aerodynamic drag

A= 0.8*b*h

$F_D = 336.04$ N

$G_{FA} = 6948.75$ N

$G_{FAY} = \frac{G_{FA}}{2}$, where G_{FAY} is the force acting on one wheel of the front axle

$G_{FAY} = 3474.375$ N

3.2.2 Lateral Force (G_{FL})[7]

$G_{FL} = \frac{W}{B}[\frac{v^2}{g.R}*H + \frac{B}{2}]$

$G_{FL} = 15800.9$

$G_{FLX} = G_{FL} - \frac{G}{2}$

$G_{FLX} = 5990.9$ N, where G_{FLX} is the force acting on the outer wheel

3.2.3 Bump Force

Maximum compression x = 50 mm
Bump force acting at the hinge point is

$F_X = k*x$

$F_X = 2500$ N

Torque about the hinge point F = $F_X * L_1$, where L_1 is the perpendicular distance

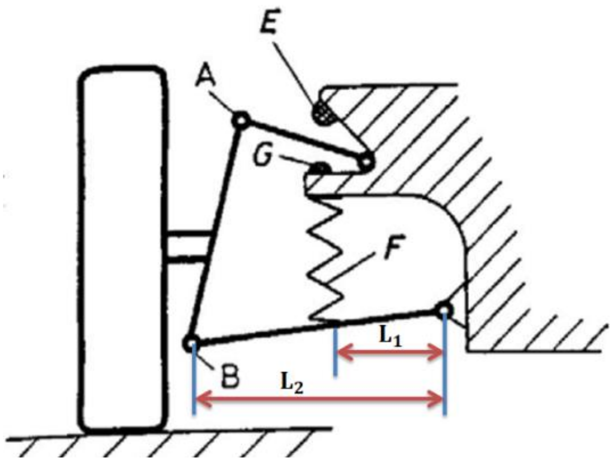

Figure 2: Suspension 2-D geometry [6]

Torque = 2500 * 0.12 = 300 N-m
Vertical bump force acting about point B,

F_Y = Torque / L_2

F_Y = 1666.7 N

3.2.4 Forces acting on lower control arm

A_X and A_Y are the forces acting on upper control arm.
B_X and B_Y are the forces acting on lower control arm

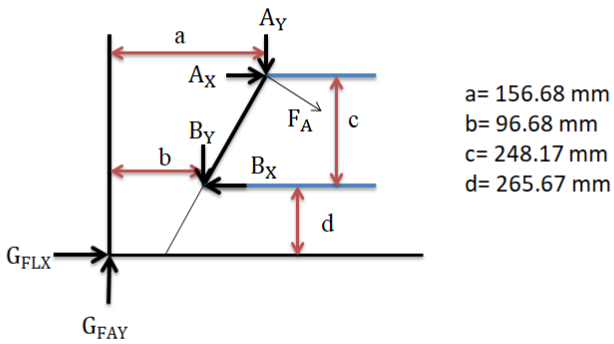

Figure 3: Forces acting on upper and lower control arm

The forces acting on the upper and lower control arms are calculated by the force and moment equilibrium.

Table 4: Forces acting on control arms

Velocity(kmph)	$A_{X(N)}$	$A_{Y(N)}$	$B_{X(N)}$	$B_{Y(N)}$
80	1335.98	211.598	3685.973	4654.92

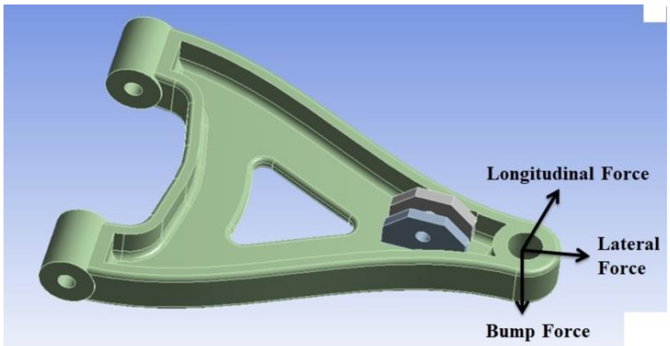

Figure 4: Typical forces acting on lower control arm

4. RESULTS & DISCUSSION

The 3-D modelling was done in Solidworks and imported in ANSYS. Then, the boundary conditions were applied to it for the static analysis. Geometry of the control arm is shown in figure 5.

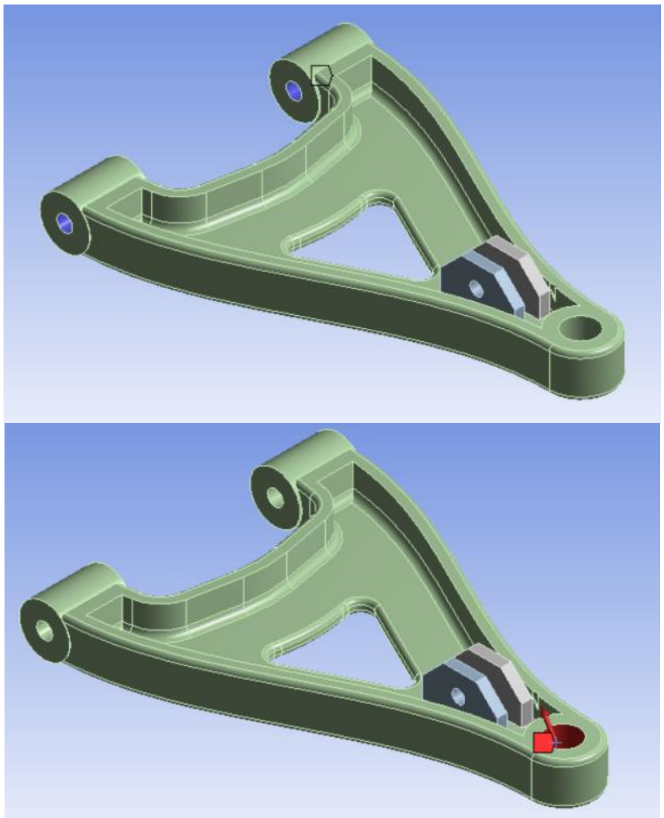

Figure 5: Geometry of the control arm with boundary condition

We have used hexagonal meshing as it helps in finding more accurate results. Element size of order 0.001 is created. The meshed structure of the geometry is shown in figure 6.
During analysis in ANSYS, it is observed the maximum stress developed in the control arm is 291.63 pa for structural steel and 291.88 pa for metal matrix composite (refer Figure 7) which is well below the maximum allowable stress of both materials. Hence, the design is safe and acceptable. It is also

observed the total deformation is 0.83 mm for the structural steel (Figure 8) and 1.54 mm for the metal matrix composite (Figure 8) which is well below the thickness of the control arm and the deformation limit of the materials. At last, the topology optimization was done using ANSYS (Figure 9). The optimized shape of the control arm is analysis and result showed the von mises stress (303.6 Mpa) acting on it is well below the yield stress of the material. It results in weight reduction by 3.8%. The static analysis results are summarized in table 3.

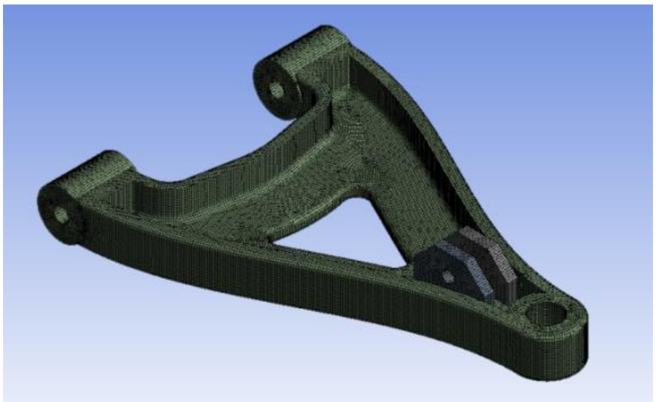

Figure 6: Hexagonal meshed geometry of control arm

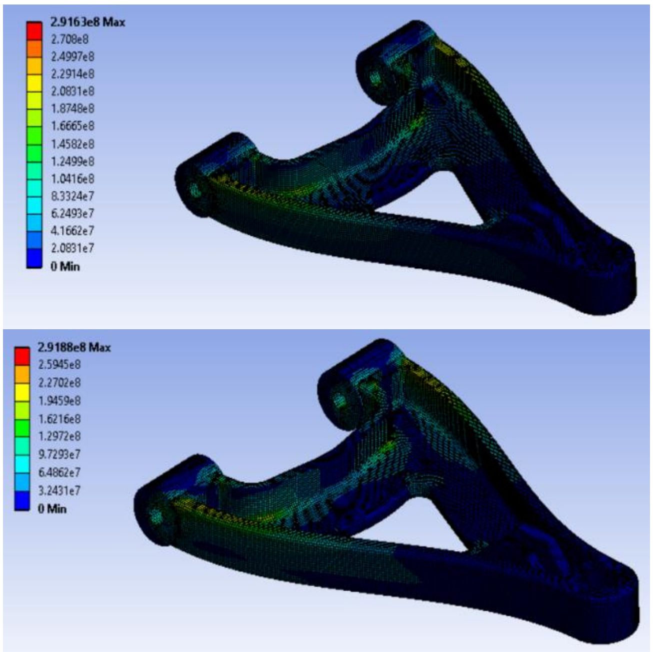

Figure 7: Von misses stress for Structural steel and AMC225xe

Table 5: Analysed Results

Design Parameters	Structural Steel	AMC225xe
Max. Stress (Mpa)	291.63	291.88
Max. Displacement (mm)	0.83	1.54
Yield Strength (Mpa)	420	480
Factor of Safety	1.44	1.64

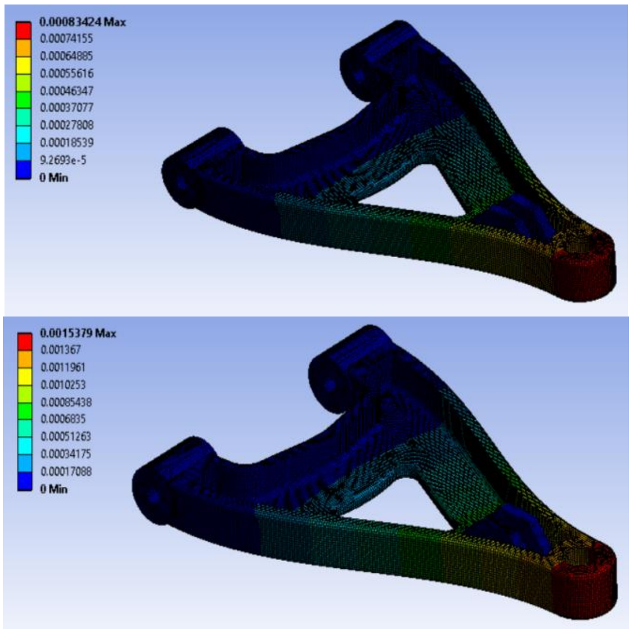

Figure 8: Total deformation for Structural steel and AMC225xe

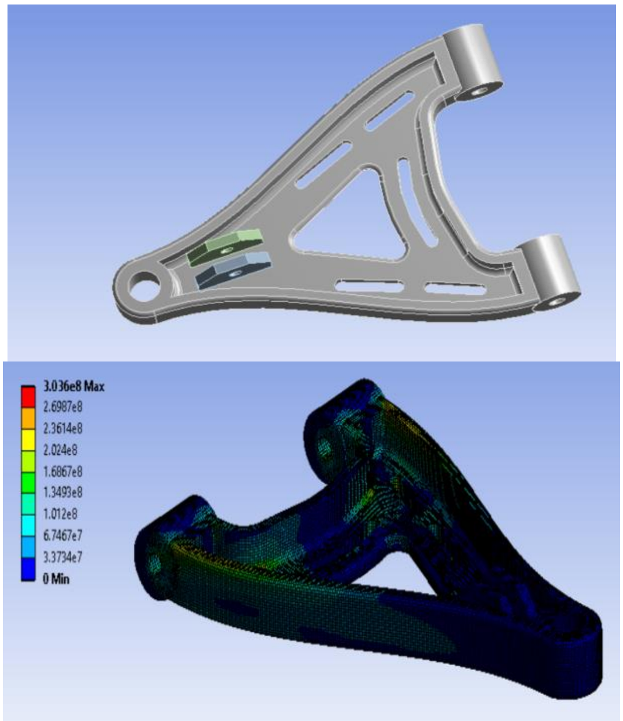

Figure 9: Topology optimization of the control arm

5. CONCLUSIONS & FUTURE SCOPE

It can be concluded that the metal matrix composite can replace the conventional material as the stress generated is within the allowable maximum stress and is in the safe limits. The use of metal matrix material helps in reduction of the weight of the control arm by 63.38%. Topology optimization results in further weight reduction by 3.8% which in turn reduces the fuel consumption and hence, increases the efficiency of the vehicle. This work can be further extended by applying dynamic forces due to lateral load transfer on banking, the vehicle at the instant of breaking on a downhill grade and experimental analysis.

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