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Shape Optimization Of A Suspension Bellcrank Using 3d Finite Element Methods

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Abstract
The paper represents an application of an optimization procedure for a mass and stress optimization of the bellcrank of a double wishbone suspension system of SRM University's FormulaSAE vehicle. The used optimization procedure, so-called Fully Stressed Design, is based on an indirect approach utilizing optimum criteria. The aim of the optimization was to achieve the lowest possible mass of the construction taking into consideration the allowed resistance and also to investigate and analyze the structural stress distribution of bellcrank at the real time condition during damping process and the spring actuation.
Keywords— Suspension, fully stressed design, goal function, Bellcrank, optimization, damping.

I. INTRODUCTION

A Bellcrank is a mechanical device used to convert translational motion of one object into translational motion of other object operating at different angles. For its application in suspension systems, a bellcrank is used to actuate the spring-damper unit using a pushrod or a pullrod.

The dimensions of the bellcrank may be altered to change the relative motion of the spring-damper unit with respect to the pushrod/pullrod. This is beneficial as the Motion Ratio can be changed without changing the properties of the spring-damper unit. The dimensions of the bellcrank decide the Motion ratio of the Suspension setup. The motion ratio in return affects the spring rate and the damping ratio.

Reducing the weight of the structure while maintaining its original functionality, or decreasing the expected maximum values of stress fields operating in individual sections of the structure contributes significantly to improved performance properties of the selected item. The conversion of structure or of both the structure and material requires the use of most advanced materials and modern manufacturing technology, as well as close cooperation between the designer and process engineer.

The world trends are aimed at significant reduction in workload of performance parts. The use of better and better computational models enables an integration of the materials science with the design process and making a new structure, thus directing to judicious use of materials.

In the process of dimensioning elements it is necessary to ensure also the required construction reliability and safety. Requirements for the high machine reliability, demanding production process, production safety force the engineers to design every component as an optimal one utilizing optimizing procedures including the best design solution based on the set of entry conditions and required parameters of the relevant component. These conditions define the optimization conditions, e.g. a satisfactory running of the construction in some upper and lower limit of response, decrease of the mass of the component, definition of the parameters which are either constant or variable during the process of optimization.

II. DESIGN METHODOLOGY

Rocker arm optimization belongs to the first optimization tasks. Its key point is to find a construction which requires the lowest costs and fulfils all necessary conditions. In practice this task is simplified into finding of a construction requiring the smallest amount of material. We use a section optimization of the arm design. The solution of this optimization task itself assumes that the particular components are prismatic and every section represents one variable of the scheme. To reduce the number of limitations during the optimization process, we take into consideration only the critical limitations maintaining the integrity of the specifications.

Finite element analysis has been used to implement optimization and maintaining stress and deformation levels and achieving high stiffness. Reduction of weight has been one the critical aspects of any design along with reduction in deformation and stress factors, which increases the life of the product. It has a substantial impact on vehicle performance.

A problem of the optimal scheme in general can be formulated as:

$K_T = [K_1, K_2, ..., K_n]$

So that the goal function:
$S = F(K) \rightarrow \min$

Respecting the limitations:
$g_j(K) \leq 0, j = 1, ..., n$ – number of the limitations

The goal function S can stand for the mass or price of the construction expressed through the scheme variables K. In the case that the scheme variables are cross sections, it is possible to express the mass in the form:

$$S = \sum_{i=1}^{n} l_i K_i = l^T K$$

where l is a vector of the variables of the scheme. Limitations are set for the cross-sectional area of the beams, for the joint displacements and stresses in section. Then:

$$K_D \leq K \leq K_H$$
$$V_D \leq V \leq V_H$$
$$\sigma_D \leq \sigma \leq \sigma_H$$

Displacements V and stresses σ are usually implied non-linear functions of the scheme variables K. For the given values K the corresponding values of the displacements V and stresses σ can be calculated through the deformation or force method.\

Validation process is an important step in this design optimization. The optimized model's performance is compared with initial models.

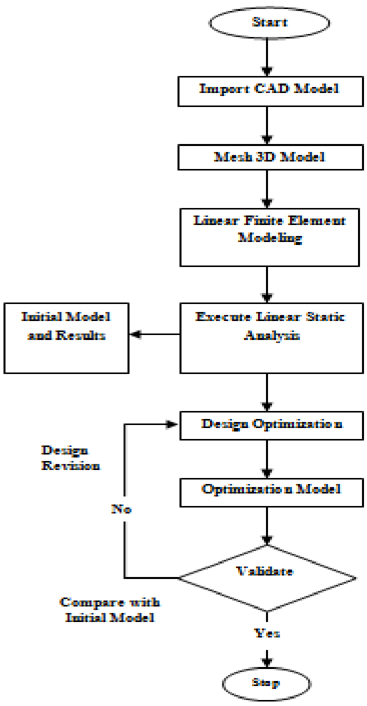

Fig.1 Design optimization process flowchart

III. MATERIAL SELECTION

Two materials Mild Steel and Aluminium 7075-T6 were compared based on their properties, and feasibility. Apart from the materials' strength to weight ratio other properties considered were Young's modulus, Poisson's ratio, hardness, ultimate tensile strength.

Table1. MATERIAL SPECIFICATION

Serial No.	Material	Density (kg/m)^3	Young's Modulus (GPa)	Poisson's Ratio	Brinell Hardness	Strength to Weigth Ratio	Ultimate Tensile Strength(MPa)
1.	Mild Steel	7870	71.7	0.33	131	47	370
2.	Aluminium 7075 -T6	2810	200	0.29	150	180	300

Both the materials have similar levels of machinability but Aluminium 7075-T6 has relatively high strength to weight ratio, Young's Modulus and Tensile Strength due to which it was chosen as the suitable material.

Gun metal bushings are used in the bellcrank to reduce the wear in the bellcrank due to the bolt, during the actuation process.

IV. DESIGN SPECIFICATION

Table2. SPECIFICATION DATASHEET

Sprung Mass of the car	300 Kgs
CG height	0.31 m
Wheelbase	1.6 m
Track width	1.2 m
Weight Distribution %(Front/ Rear)	40/ 60
Spring Stiffness	36.71 N/mm
Maximum Longitudinal Load Transfer at 1.7 G	1148N

V. DESIGN AND BOUNDARY CONDITIONS

A. Designing a CAD model

CAD model of rocker arm is developed in 3D modeling software SOLIDWORKS, it consists of bolt holes for mounting, a pivot point. Rocker design mainly depends on packaging of the components and leverage distance of travel.

The motion ratio of the bellcrank decides the relative motion of the wheel with respect to the spring-damper setup. The motion ratio can be changed by changing the relative dimensions of the bellcrank about the pivot point of actuation.

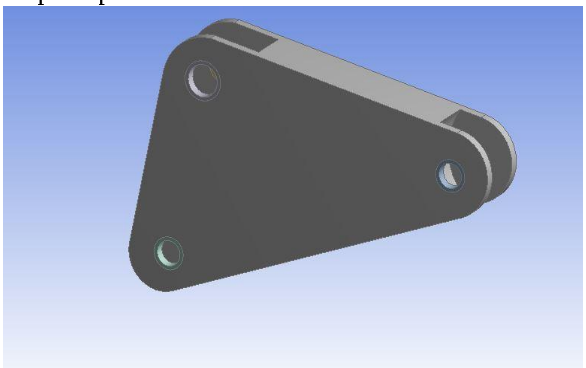

Fig2. Initial model of the rocker arm

B. *Meshing*

CAD model of rocker converted into Parasolid file. The model is imported into Ansys Workbench simulation. Geometry cleanup was performed prior to meshing of model. Finite element model was developed using Ansys Workbench Simulation. For better quality of mesh fine element size is selected.

C. *Methodology- Force distribution*

For carrying out the analysis on the bell crank used in a Double Wishbone Pushrod Actuated suspension setup, calculation of three major forces were required:

- Force on the bell crank due to Pushrod
- Force on the bell crank due to Spring/Damper setup
- Force on the bell crank at the Chassis mount

D. *Modification of Rocker*

An initial CAD model of rocker is modeled with the design constrains. Taking the initial design as reference, a modified geometry with layer of Gun Metal bushings on the pivots has been modeled for reducing wear in the Bellcrank.

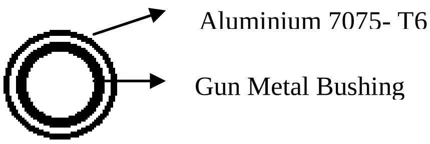

Fig3. Layer of Gun Metal in the Bolt holes

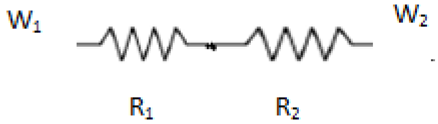

Fig4. Mechanical circuit resistance model for wear reduction on the bellcrank (R1: Gun Metal, R2: Aluminium 7075- T6 resistance)

VI. CALCULATION

A. *Force Calculation*

For finding the force on the bell crank due to the pushrod, the forces on all the members of the double wishbone suspension were found out using the Matrix method. This method consists of the principle equation:

$$AX=B$$

Using the suspension coordinates and tire contact patch forces, the matrix [A] and [B] is found out:

Table3. SUSPENSION COORDINATES (in metres)

	Upper Fore	Upper Aft	Lower Fore	Lower Aft	Tie Rod	Pushrod
Chassis End	0	0.3	0	0.3	0.2004	0.11
	-0.28	-0.28	-0.24	-0.24	-0.225	-0.33942
	0.31	0.31	0.145	0.31	0.185	0.595
Upright End	0.125	0.125	0.091	0.091	0.17	0.11
	-0.58266	-0.58266	-0.59734	-0.59734	-0.558	-0.56
	0.375	0.375	0.135	0.135	0.185	0.165

Matrix [A], a 6X6 matrix wherein the first 3 columns represent the X, Y and Z component of the unit vector of the wishbone, pushrod and tie rod coordinates along their respective directions and the remaining 3 columns represent of their attachment to the Upright.

Table4. FORMULATION OF MATRIX [A]

Matrix [A]

0.566051213	-0.64617	0.4159204	-0.56899	-0.10657	0
-1.37056848	-1.11754	-1.633242	-0.97284	-1.1674	-0.94444
0.294346631	0.240005	-0.045706	-0.47643	0	-1.8411
0.342459172	0.279235	0.2477894	0.415923	0.215968	1.186851
0.175475876	-0.27231	0.0603085	-0.03346	-0.01972	0.202521
0.15849434	-0.51619	0.0998209	-0.42841	-0.25793	-0.10389

For analysis, the forces considered are under braking and tires for the given produce a maximum longitudinal force of 1200 N (F_z) and maximum lateral force of 1100 N (F_y) N. The maximum braking torque is 750 N-m. The load on a single tire under maximum longitudinal acceleration and lateral of 1.7G's is 1148 N (F_z).

The first three values of vector [B] are the forces along the respective forces along X, Y and Z axis of the vehicle plane. The remaining forces are the moments produced about the vehicle centerline due to the forces acting on the contact patch center.

R is the distance of the vehicle from the tire contact patch center. This is equal to half the track width of the vehicle.

$$M_x = (F_z \times R) - (F_y \times R)$$
$$= (1148 \times 0.6) - (0)$$
$$= 688.8 \text{ N}$$

$$M_y = (F_x \times R) - (F_z \times R)$$
$$= (1200 \times 0.6) - (1148 \times 0.6)$$
$$= 31.5 \text{ N}$$

$$M_z = (F_y \times R) - (F_x \times R)$$
$$= (0) - (1200*0.6)$$
$$= -720 \text{ N}$$

Vector B, a column vector comprising of X, Y and Z component of forces about the tire contact patch centre and are the X, Y and Z components of the moments of the contact patch forces about the origin.

Table5. FORMULATION OF MATRIX [B]

Matrix [B]

1200
1100
1148
688.8
31.5
-720

Table6. FORMULATION OF Matrix $[A]^{-1}$

Matrix $[A]^{-1}$

-1.114	0.164	1.485	1.997	3.149	1.149
0.007	0.169	0.556	1.771	-4.013	1.021
-1.708	-1.062	-0.922	-1.484	-2.357	4.449
-5.475	-0.957	0.367	-0.739	6.033	5.513
7.217	0.811	-1.044	-1.978	-0.354	-12.15
1.282	0.322	-0.305	0.778	-1.522	-1.220

Vector X, a column vector representing the forces acting on each Suspension wishbone, steering tie rod and pushrod.

Multiplying the matrix $[A]^{-1}$ with the vector [B], we get the force values in vector [X] as

Table7. FORMULATION OF VECTOR [X]

Vector [X]

1195.827
1190.923
-8575.961
-11489.728
15734.151
2908.803

From vector [X] the force on the pushrod is calculated as 2908.8 N

The spring stiffness of 36.71 N/mm is considered for the analysis. For a maximum spring deflection of 21.8 mm the force acting on the bell crank will be 800.28 N.

B. Suspension Spring and Damper Calculation

Longitudinal Load Transfer= 1148 N
Ride Travel Allowed= 40mm
Ride Rate= 28.49 N/mm
Ride Frequency: - 3.48 Hz
Bellcrank Motion Ratio= 0.8
Percentage Sprung Mass (Front)= 40%

$$\text{Sprung Mass Front} = (40/100) \times 300$$
$$= 120 \text{ Kgs}$$

$$\text{Spring Rate} = 4(\pi)^2 \times (\text{Ride Frequency})^2 \times (\text{Motion Ratio})^2 \times (\text{Front Sprung Mass})$$

$$= 36718.03 \text{ N/m}$$
$$= 36.71 \text{ N/mm}$$

The Damper Dyno curves of Ohlins TTX 25 were studied and damping ratios for various damper settings were calculated.

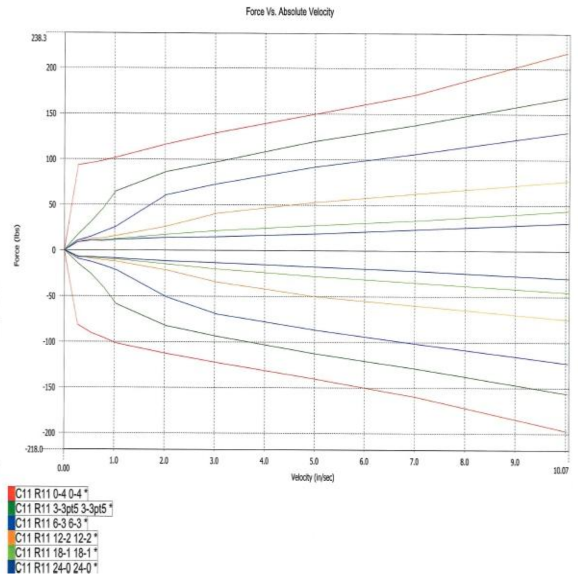

Fig5. Force-Velocity Dyno Plots of the Ohlins TTX-25 FSAE Dampers

After an iterative process, the damper setting C11 R11 6-3 6-3 was considered in the bellcrank design.

Tire Rate= 130 N/mm

Wheel Rate= (Tire Rate× Ride Rate)/ (Tire Rate- Ride Rate)
=36.83 N/ mm

Critical Damping= 2× (Wheel Rate× Front Sprung Mass)$^{0.5}$
= 4.20 N× sec/ mm

From the Damper Dyno Plot, the compression and rebound slopes of the curve was found to find the damping ratio.
For Low Speed Compression Damping,

∇ Force = 59.23N
∇ Velocity = 6.60 mm/ sec

Damping Rate= ∇ Force/ ∇ Velocity
=8.96N×sec/mm

Damping Rate (at wheel)= Damping Rate× (Bellcrank Motion ratio)^2
=8.96× 0.8^2
=5.74N×sec/mm

Damping Ratio= Critical Damping/ Damping Rate (at wheel)
=1.36

For High Speed Compression Damping,

∇ Force= 51.39N
∇ Velocity= 28.34 mm/sec

Damping Rate= 2.12 N*sec/mm

Damping Rate (at wheel) = 1.36 N*sec/mm

Damping Ratio= 0.32

For Low Speed Rebound Damping,

∇ Force= 127.09 N
∇ Velocity= 25.4 mm/sec

Damping Rate= 5.0 N× sec/ mm

Damping rate (at wheel) = 3.20 N× sec/ mm

Damping Ratio= 0.80

For High Speed Rebound Damping,

∇ Force= 47.0 N

∇ Velocity= 25.4 mm/ sec

Damping Rate= 1.85 N× sec/ mm

Damping Rate (at wheel) = 1.18 N× sec/ mm

Damping Ratio= 0.29

Based on the ride and damping calculations, the bellcrank with a motion ratio of 0.8 was designed.

VII. RESULTS

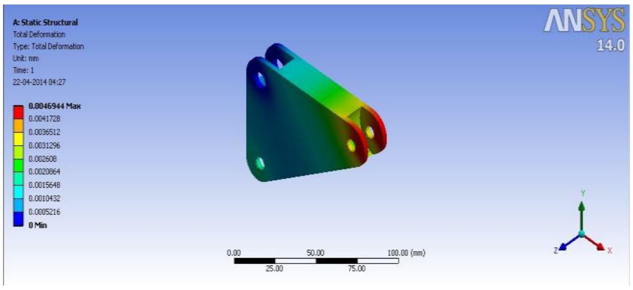

Fig6. Total deformation for the rocker without ceramic insert

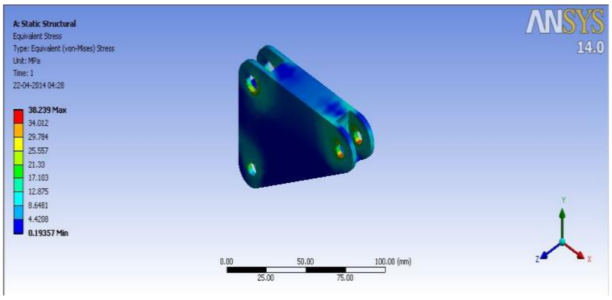

Fig7. Von Mises stress for the rocker without ceramic insert

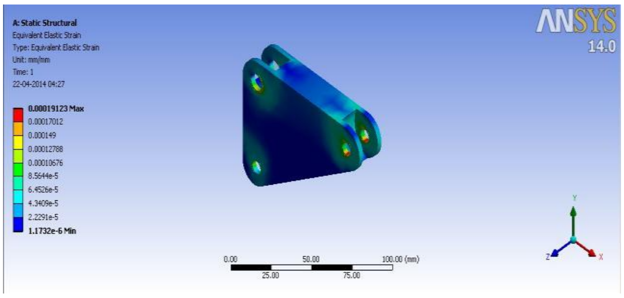

Fig8. Equivalent elastic strain for the initial Bellcrank design

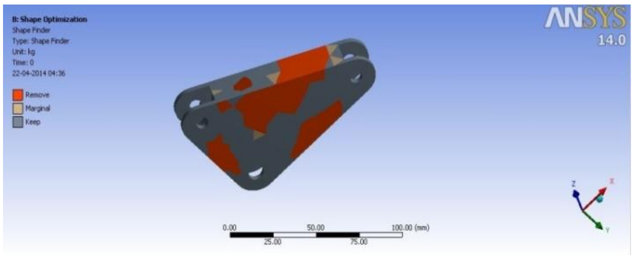

Fig9. Shape optimization for material removal

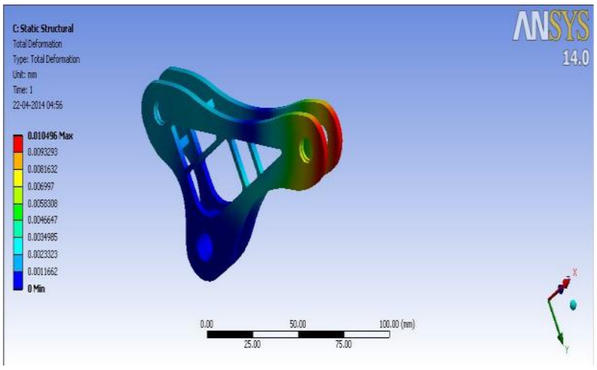

Fig10. Total deformation for the final bellcrank design

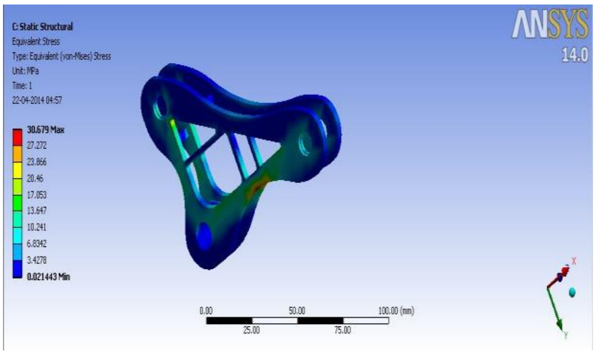

Fig11. Von Mises stress for the final bellcrank design

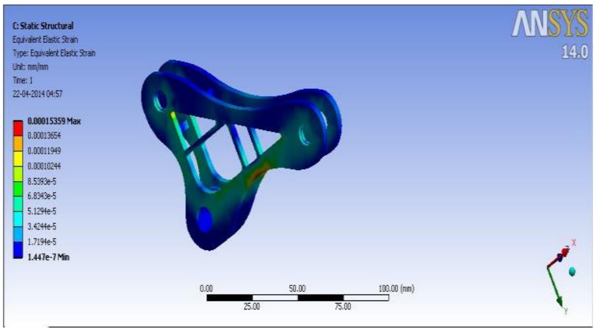

Fig12. Equivalent elastic strain for the final bellcrank design

VIII. CONCLUSION

Optimization method used in this study in reducing the mass of rocker arm. Validation is done through finite element solver with the initial model and checked that maximum stress and displacement are within control. This optimization process also gives small change on the stress. The stress has significantly reduced in the new optimized design proving to provide better life of the product. Therefore, the overall weight of the bellcrank can be reduced by 22.34% and stress by 19.77%.

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Table8. ANALYSIS RESULT

Serial No.	Material	Deformation (mm)	Von Mises Stress (MPa)	Equivalent Strain	Weight (grams)
1.	Bellcrank (Initial Model)	0.0046	38.239	0.000191	217.36
2.	Bellcrank (Final Model)	0.0104	30.679	0.000153	168.81