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Compliance Minimization of Laminated Composite Chassis Frame by Optimizing Size and Rotation

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***Abstract*--This study investigates the improvement of stiffness for a laminated composite chassis frame, along with obtaining the least minimized mass by structural optimization. The laminate thickness and orientation are the design variables under certain constraints. The optimal material distribution was computed for the micro structural elements of the finite element model. A static analysis was performed and then optimized with minimized parameter values. The overall mass was reduced by 56.19% and with the displacement achieved within 3.442mm. The convergence and the analysis results were evaluated using the HyperWorks (OptiStruct & RADIOSS). The model was designed in CATIA-V5R16 and Pro Engineer.**

I. INTRODUCTION

Structural optimization is an approach that optimizes material distribution over a given design domain, under subjection to a set of mechanical loads and boundary conditions.

We consider the minimum compliance topology design problem with a volume constraint and discrete design variables, as shown by [1]. Topology optimization has been extensively applied to a variety of structural optimization problems such as the stiffness maximization problem [2], vibration problems [3], and optimum design problems for compliant mechanisms [4].

Structural optimization can be performed for various types of materials. The density method together with this penalization is often called the SIMP method (Solid Isotropic Microstructures with Penalization) is used in case of isotropic materials, as shown by [5]. On the other hand as shown by [6] homogenization is an intrinsic part of topology design together with the area of material science which is concerned with bounds on the properties of composites. The microstructure is a composite material with an infinite number of infinitely small voids [7]. This method also considers the angular orientation along with material thickness (full) or void, with which the elasticity of the elements can be computed with a finite element method.

II. OBJECTIVE OF THE STUDY

It is known that, a material which is denser is likely to be more stiff (k) and vice-versa and therefore the displacement due to an external loading will also vary respectively. Hence the elements that have more displacement value under external loading are less stiff and therefore needs material filling (1), similarly the elements that are less subjected to or no displacement can take a null or void of material (0). These material parameters such hole size or rotation are the deciding factors for the improvement of stiffness. The model shown in Figure2.1, taken for the study is an under-bone type chassis frame of a scooter which is made up of steel, with all the welded extra panel brackets weighing 17Kg on a hanging spring scale, which is heavier than the geared diamond type chassis frame bikes.

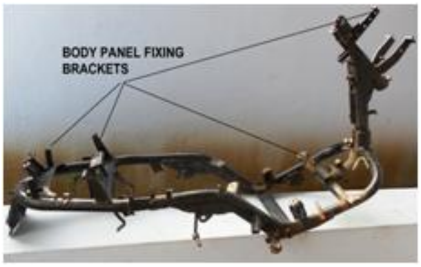

Figure2.1. Side view of the actual scooter chassis frame, mass=17Kg with extra body panel supporting brackets.

The composite material chosen for optimization is a unidirectional High Modulus Carbon Fibre which is much stiff, least in weight but costs more among other laminated fibres. The choice of material is not the issue here, as a reason to say more of the future hybrid vehicles are going on for carbon fibre chassis, namely the BMW i3 and Ferrari's Enzo, which obviously have much effect on the cost. However the upcoming conceptual products that are yet to come can be designed optimally in a cost-effective way to minimize these issues which can also bring out concepts of alternative manufacturing techniques that are economically much compatible towards the design pattern of these modern structures.

III. STATIC ANALYSIS AS REFERENCE

The initial wireframe model is shown in Figure 3.1 with sweep line profiles randomly in space from which the developed shell model is shown in Figure 3.2, the structure consist of three main sections the front tube section (F.T.S) with O.D 40mm and I.D 30mm, middle neck section (M.N.S) with O.D 48mm and I.D 42 mm and the main body section (M.B.S) with O.D 38mm and I.D 34 mm. There are three mounting brackets, engine fixed to two middle brackets and shock absorber to the rear bracket, also the front tube interior wall is hinged with the handle steering shaft. The remaining part of the engine is supported by the rear tyre along with the CVT transmission system.

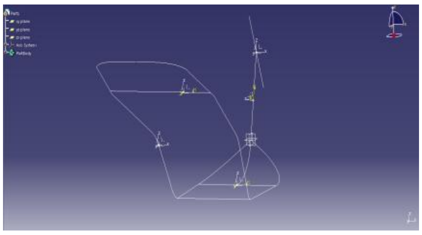

Figure3.1. Wireframe model

The magnitude of the load acting is calculated from the kerb weight of the entire vehicle (103Kg) and with standard weight of one operator (70 Kg) which is overall about 1700 N force. The finite element shown in Figure3.3 was developed using 2D mesh elements considering the model as shell structure.

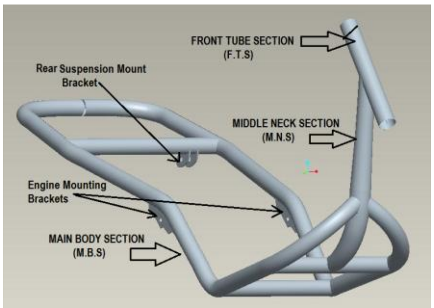

Figure3.2. Shell structural model

For isotropic materials the property adopted was pshell, and for orthotropic materials pcompp, these special card images shown in [8], features facilitate altering thickness for large structures thereby minimizing mesh time and design efforts.

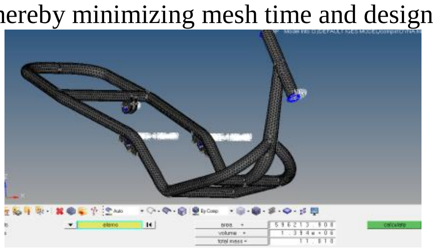

Figure3.3. Steel finite element model with mass=11.01Kg

The element mass computed excluding the extra body panel bracket design is 11.01Kg with a difference from the actual model about 6Kg. For this study let us consider the mass 11.01Kg which must be further reduced by optimization. The static displacement values were computed considering the model as a simply support beam structure one end fixed at the rear bracket and the other end hinged at the front tube section, with point loads acting at the two middle brackets. The Figure3.4, shows the steel structural displacement value of max=3.442mm.

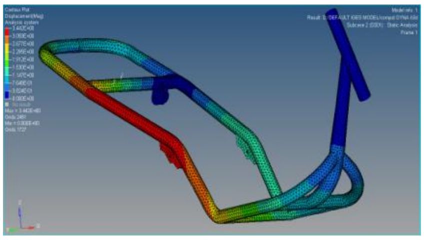

Figure3.4. Displacement of steel model, 3.442mm

This shows that the deflection has more effect at the mid region of the chassis where there are no supports or any other fixations.

The above value is a reference for the carbon fibre model which is specified within those ranges. A similar static analysis was performed with increased tube sectional diameters for the carbon fibre model. The analysis was done without optimizing its topology with resulting mass of 4.881Kg shown in Figure3.4 and displacement value of 2.089mm as shown in Figure3.5.

Figure3.5. Carbon fibre finite element model without optimized mass=4.881Kg.

The material properties [9], used in the problem is shown the below table5.1, and its notations are described below,

E– Modulus of Elasticity
E1–Longitudinal Modulus
E2– Transverse Modulus
G– Axial Shear Modulus
Nu– Poisson Ratio
Xt– Longitudinal Tension
Xc– Longitudinal Compression
Yt–Transverse Tension
Yc– Transverse Compression
S– Shear Strength
ρ– Density

Table5.1. Material properties

Material / Notations	Steel	H.M.C.F
E	210GPa	-
E1	-	175GPa
E2	-	8GPa
G	-	5GPa
Nu	0.33	0.30
Xt	-	1000MPa
Yt	-	40MPa
Xc	-	850MPa
Yc	-	200MPa
S	-	60MPa
ρ	7.9g/cm^3	1.60g/cm^3

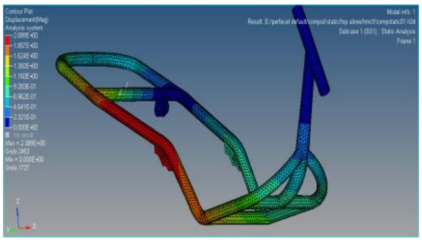

Figure3.6. Displacement of carbon fibre model, 2.089mm

Though the value is less than steel displaced values the mass is not cost effective therefore it can be further reduced in the following design modifications made by varying the size of laminate thickness and angular orientation which in turn also results in displacement result which is less than 3.442mm.

IV. Optimization of Size and Orientation

The objective is to minimize the mass with design variables such as size, rotation under the constraints such displacement, modal frequencies in case of dynamic analysis. This section shows the optimized structure that attains a mass further lesser than 4.881Kg. The optimized thickness ranging from 1mm to 5mm is shown in the following Figure 4.1. The maximum thickness is at regions where there is more displacement and least where there is lesser amount of displacement.

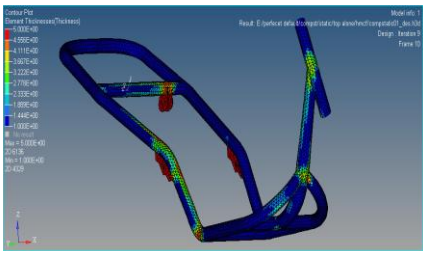

Figure4.1 Optimized laminate size distribution

The orientation thickness varies for each angles 0, 45, -45 and 90 of the plies ranging from values, 0.2549mm to 2.726mm, are shown in Figure 4.2, 4.3, 4.4 and 4.5. A balance in the angle ranging for 45 and -45 is done for better orientation strength.

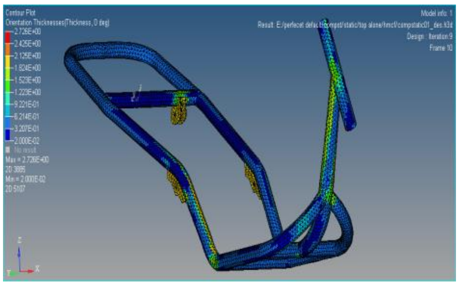

Figure4.2. Orientation thickness for 0 degree plies

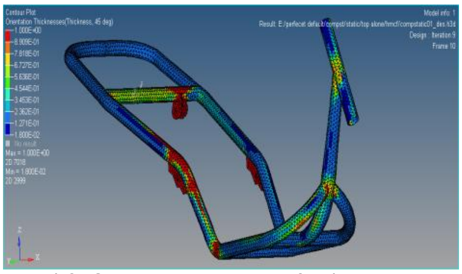

Figure4.3. Orientation thickness for 45 degree plies

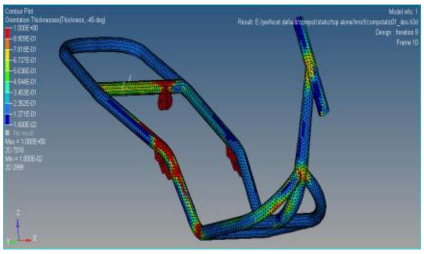

Figure4.4. Orientation thickness for -45 degree plies

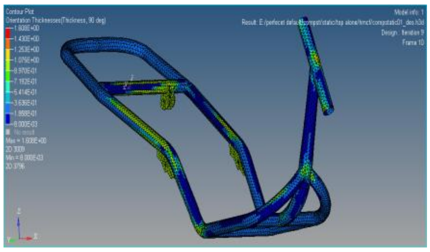

Figure4.5. Orientation thickness for 90 degree plies

From the above orientation thickness the balance angles 45 and -45 has the maximum range of thickness and promises to provide much strength by keeping the displacement within range.

V. Result and Discussion

The optimal design here gives an understanding of how we can minimize the material cost with respect to its volume fractions. The resin volume is not included hence the mass may increase to a few kilos while fabricating the model as a prototype.
The following graphs show the final optimal value for mass, displacement and constraint violations in Figure 5.1, 5.2 and 5.3. The final optimized weight of the model is about 1.44616Kg without considering extra brackets. The extra panel brackets (6Kg) can be replaced by S2-glass fibre

composites which could be much more economical. The table 5.1 represents the optimized values for overall mass and displacement.

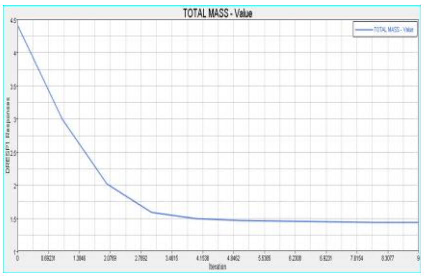

Figure5.1. Mass reduced from 4.881Kg to 1.44616Kg

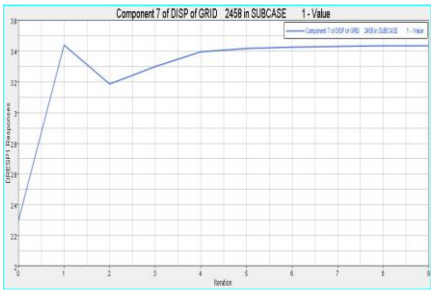

Figure5.2. Displacement at grid 2458 with max value 3.439mm

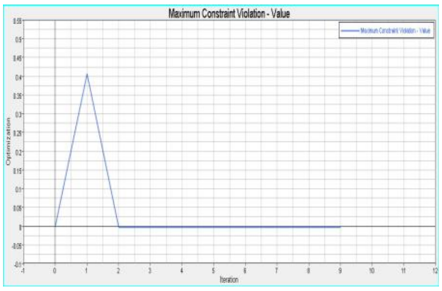

Figure5.3. Constraints violated is 0%

Table5.1. Overall values of mass and displacement

<table>
<tr><th>Material</th><th>Steel</th><th>H.m.c.f</th></tr>
<tr><td colspan="3">Before Topology Optimization</td></tr>
<tr><td colspan="3">Mass</td></tr>
<tr><td>Excluding body panel brackets (Computer design model)</td><td colspan="2">Including body panel brackets (Actual real scale model, adding 6Kg)</td></tr>
<tr><td>Steel-11.01Kg</td><td colspan="2">Steel-17Kg</td></tr>
<tr><td>H.M.C.F-4.881Kg</td><td colspan="2">H.M.C.F-10.881Kg</td></tr>
<tr><td colspan="3">Displacement</td></tr>
<tr><td>Steel- 3.442mm</td><td colspan="2">Steel-Nil</td></tr>
<tr><td>H.M.C.F-2.089mm</td><td colspan="2">H.M.C.F-Nil</td></tr>
<tr><td colspan="3">After Topology Optimization</td></tr>
<tr><td colspan="3">Mass</td></tr>
<tr><td>Steel-Nil</td><td colspan="2">Steel-Nil</td></tr>
<tr><td>H.M.C.F-1.44616Kg</td><td colspan="2">H.M.C.F-7.44616Kg</td></tr>
<tr><td colspan="3">Displacement</td></tr>
<tr><td>Steel-Nil</td><td colspan="2">Steel-Nil</td></tr>
<tr><td>H.M.C.F-3.439mm</td><td colspan="2">H.M.C.F-Nil</td></tr>
</table>

From the tabulated values above, considering the extra panel fixing brackets the mass reduced after topology optimization is about 56.19%.

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