Optimation Front Upright Racing Car Using Finite Element Analysis

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***Abstract*−Upright is one of the car's suspension system parts that plays an important role in creating and comforting the car. Supporting driving safety, upright components must be designed to be light but strong to withstand loading which is acceleration, deceleration, and cornering load. Not only strong as a benchmark, but components must also be lightweight, so a topology optimization method has chosen. Using finite element software makes the optimization process very easy and very fast with maximum accuracy. The process is by inputting the model from CAD software, defining materials, input constraints and vector styles, meshing process, and finally the solution process. From the simulation results, it will be known the value of the solution in the form of stress, deformation and safety factor of the upright component. From several topology optimization designs, they will be compared to find out which is the best design which will be used as a design recommendation. By referring the result, 43% mass reduction is the best optimum design, its safety factor is 4.956.**

***Keywords*−Hub carrier, Optimization, Steering knuckle, Topology, Upright**

I. INTRODUCTION

Upright is one of the most important components in the suspension system. All suspension components such as wishbones, shock breakers, tie rods and push rods connected to upright. All the forces when car moves will meet because the interaction of the racetrack and the wheels will go to upright [1].

Not only acceleration and strength can support the performance of a racing car, but light weight is also one of the keys. Therefore, an optimization processing carried out the component becomes lighter. This process is called topology optimization. Some mechanical performance ability, especially a structure, have strongly related with its topology. Size and shape optimization cannot give the best structural performance, since these methods cannot change the structure's topology. Hence, topology optimization should be employed to obtain the best performance [2].

The selection of component's material and design is an important topic in industry, which produce sustainable and competitive products. Realizing strength and endurance requirements on a component level, topology and shape optimization are useful tools to predict an optimal component design in early phases of the design process [3].

Based on the results of the evaluation of the FSAE A 2020 event, the design of the Formula ITS car is still considered very heavy. The total weight of the car obtained from the design reaches 360.72 kg. Total mass is weight of the car and the driver. This compared with another teams, which a top 10 ranking in Formula events, which are very far apart.

From the design evaluation, the Formula car must start to reduce mass as much as possible. Not only cut the side geometry, but also consider the strength and safety factor of the components. Safety factors use ration from the yield strength and von-mises stress (result from the simulation). It must be stable in dynamic race. One of the optimization methods that can be used is topological optimization. It aims to reduce mass and volume by using a stress distribution approach on a material in a certain design. This optimization does not change the size and basic design of a structure but produces very complex geometric details.

II. METHOD

The research idea will be useful on designing a race car, then make the design criteria. Followed by the design process which includes supporting data from the upright such as the material to be used, the working force, the initial constraint geometry without changes which will then be carried out in the modeling process in Computer Aided Drawing (CAD) software. After that, a simulation of the static structure was carried out in Computer Aided Engineering (CAE) software. The next step is the topology optimization process. Then proceed with the final simulation using finite software elements and selecting the best optimal design. Based on the final project in this report is to optimize the strength of the upright.

A. Design Parameter

In making a design, you must first determine the design parameters. The design parameter is a design limitation so that what we design is right on target in accordance with regulations that refer to the FSAE rules [4]. Here are the design parameters:

TABLE 1
DESIGN PARAMETER OF FORMULA STUDENT CAR

No	Parameter	Front Wheel	Rear Wheel	Units
1	COG Height	410.45		mm
2	Wheel Diameter	510	510	mm
3	Wheelbase	1550		mm
4	Front track width	1220	1120	mm
5	Total Mass with a Driver	360.72		kg
6	Un-sprung Mass	81.391		kg
7	Sprung Mass	279.329		kg
8	Mass Distribution	40	60	%

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9	Sprung Weight Distribution	111.7316	167.5974	kg
10	Sprung Weight @wheel	55.865	83.798	kg
11	Total Mass Distribution	144.28	216.44	kg
12	Front / Rear	72.14kg	108.22kg	kg
13	µ road	0.6	0.6	

B. CAD

The geometry of this component will be the benchmark for making the constraint model of the upright which being model in Computer Aided Drawing (CAD) software.

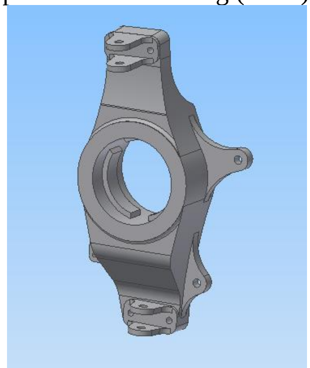

Figure 1. Upright Design CAD

C. Calculation of Loading Force

1) Calculation of Load Transfer

On any vehicle or object that moves there is a load transfer. load transfer is the transfer of vehicle weight that is accelerated from the front of the car to the back of the car or vice versa, which occurs during acceleration and braking. The total weight of the vehicle does not change, the load is only transferred from one end of the wheel to the other. There are two load transfers, namely longitudinal load transfer and lateral load transfer. The difference is that for longitudinal load transfer it works on the X axis of the car, namely when the car accelerates forward and brakes, while for lateral load transfer it works on the Y axis of the car, namely when the car turns (cornering) [5]. And here is the formula:

a) Longitudinal Load Transfer

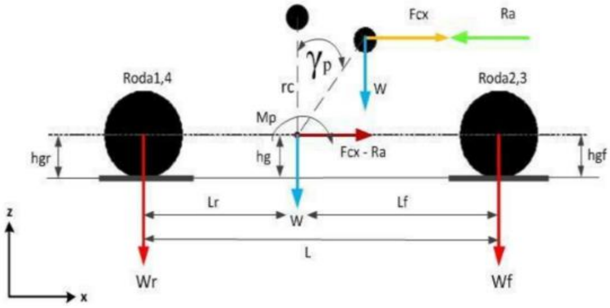

Figure 2. FBD Longitudinal Load Transfer [6]

$$Long.Acc(g)x\frac{weight(N)\ x\ CoG(m)\ x\ height(m)}{Wheelbase(m)}$$

Lateral Load Transfer

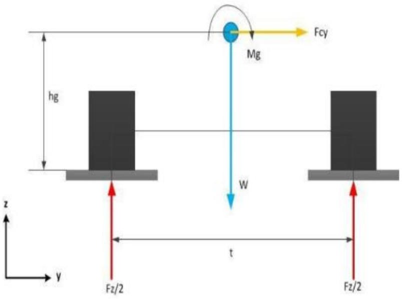

Figure 3. FBD Lateral Load Transfer [6]

$$Lat.Acc(g)x\frac{weight(N)\ x\ CoG(m)\ x\ height(m)}{Trackwidth(m)}$$

2) Force Calculation When Car Accelerate

The acceleration loading force is analysed from the FSAE car racetrack parameters during the acceleration event [4] with the following data:

TABLE 2

DATA FROM FSAE TRACK ACCELERATION EVENT

No	Data From FSAE Track Acceleration Event		
1	Maximum Velocity	100	Km/h
2	Wheelbase	1.55	m
3	Time for Maximum Velocity	4	s
4	Longitudinal Acceleration	6.9425	m/s^2
5	Longitudinal g's	0.7	g

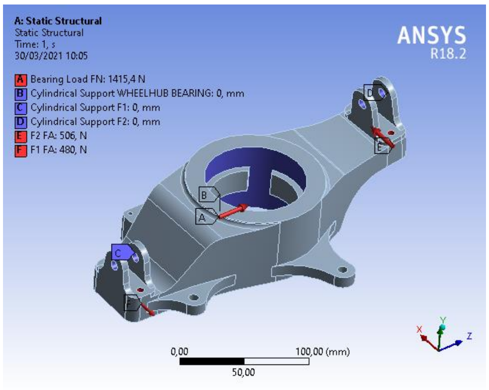

Figure 4. FBD Analyze Acceleration

Information

FN = Wheel lift / vertical force

F1 = Lower wishbone reaction force

F2 = Upper wishbone reaction force

a = Distance F1 to center Upright

b = Distance F2 to center Upright

When the wheel rotates in a straight line, the acceleration force (FA) on the tire is assumed to be at the center of the wheel so that:

FA = F1+F2

FA = FN×μ_{tyre}

TABLE 3

ACCELERATION LOAD RESULTS

No	Acceleration Load Result			
	Name of Load	**Value**	**Unit**	**Direction**
1	FN	44.353	N	Axis Z Negative
2	F1	479	N	Axis X Negative
3	F2	505.621	N	Axis X Positive

3) Force Calculation When Car Decelerate

The deceleration loading force is analyzed from the parameters of the FSAE car racetrack during the braking event [4] with the following data:

TABLE 4

DATA FROM FSAE TRACK BRAKING EVENT

No	DATA FROM FSAE TRACK Braking Event		
1	*Braking Speed*	80	Km/h
2	*Braking Distance*	16.665	m
3	*Wheelbase*	1.55	m
4	*Braking Time*	1.5	s
5	*Braking Acceleration*	14.8133	m/s^2
6	*Braking g's*	1.5	g

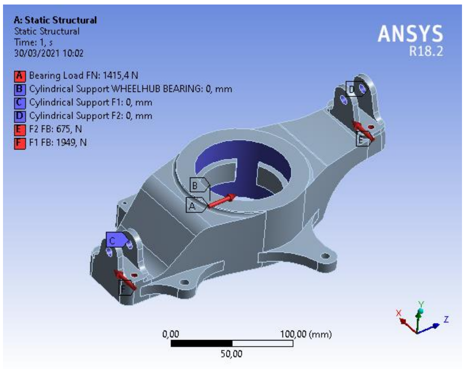

Figure 5. FBD Analyze Deceleration

Information
FN = Wheel lift / vertical force
F1 = Lower wishbone reaction force
F2 = Upper wishbone reaction force
A = Distance F1 to center Upright
B = Distance F2 to center Upright

Assuming the acceleration due to gravity is 9.81m/s2
FB = FN x μ_{tyre}
FB = F1 – F2
MB = M1 + M2 → FB × r_{roda} = F1 × a + F2×b

TABLE 5

DECELERATION LOAD RESULTS

No	Deceleration Load Result			
	Name of Load	**Value**	**Unit**	**Direction**
1	FN	2111.74	N	Axis Z negative
2	F1	1948.465	N	Axis X Positive
3	F2	674.859	N	Axis X Positive

4)Force Calculation When Car Cornering

The lateral loading force is analyzed from the parameters of the FSAE car racetrack during an autocross event [4] with the following data:

TABLE 6

DATA FROM FSAE TRACK AUTOCROSS EVENT

No	DATA FROM FSAE TRACK Autocross Event		
1	Skid pad Radius	6	m
2	Velocity during Cornering	11.11	m/s
3	Mass Distribution Front	126	kg
4	Mass Distribution Rear	189	kg
5	Lateral Acceleration	20.572	m/s^2
6	Track Width Front	1.22	m
7	Track Width Rear	1.12	m
8	Lateral g's	2.09	g

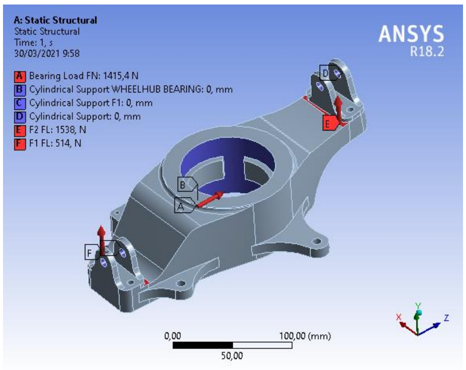

Figure 6. FBD Analyze Cornering

Information
FN = Wheel lift / vertical force
F1 = Lower wishbone reaction force
F2 = Upper wishbone reaction force
a = Distance F1 to center Upright
b = Distance F2 to center Upright

When the car is cornering, the entire weight of the car tends to shift to the outer wheels because the location of the Center of Gravity moves the outer wheels which makes the outer wheels receive a greater force from the rear.
FL = FN x μ_{tyre}
FL = F2 – F1
ML = M1 + M2 → FB × r_{roda} = F1 × a + F2 × b

TABLE 7

PARAMETERS DISPLAY

No	Cornering Load Result			
	Name of Load	**Value**	**Unit**	**Direction**
1	FN	1.706.33	N	Axis Z Positif
2	F1	513.938N	N	Axis Y Positif
3	F2	1.537.736	N	Axis Y Negatif

2.4. Mechanical Properties of Material

To be able to maximize the performance of the car when maneuvering, light-weight materials are needed. The optional materials are Al 7075 T6 and Al 6061. Those materials are less mass and middle yield strength. The material also applying in aircraft manufacturing material and another lightweight structure. The specification shows in Table. 8.

TABLE 8

CORNERING LOAD RESULTS

No	Mechanical Properties	Al 6061	Al 7075 T6	Unit
1	Density	0.0027	0.00281	g/mm^3
2	Young Modulus	68.9	71.7	GPa
3	Poison's Ratio	0.33	0.33	
4	Ultimate Tensile Strength	310	572	MPa
5	Yield Strength	276	503	MPa

III. RESULTS AND DISCUSSION

A. *100% Initial Design Simulation Results*

Analysis of the initial design results with 100% mass retain without topology optimization. Comparison of the analysis of the simulation results using Al 7075 T6 and Al 6061 materials. Acceleration, deceleration, and cornering load applied in that geometry.

TABLE 9

RESULT OF SIMULATION ANALYSIS OF 100% INITIAL DESIGN ACCELERATION LOADING

No	Upright Design Analysis Results When the Car Accelerates			
1	Material	*Von-mises Stress* (MPa)	Total Deformation (mm)	Safety Factor
2	Al7075-T6	75.324	0.0030755	15
3	Al6061	75.324	0.0032005	15

TABLE 10

RESULT OF SIMULATION ANALYSIS OF 100% INITIAL DESIGN DECELERATION LOADING

No	Upright Design Analysis Results When the Car Decelerates			
1	Material	*Von-mises Stress* (MPa)	Total Deformation (mm)	Safety Factor
2	Al7075-T6	24.928	0.0071188	15
3	Al6061	24.928	0.0074081	11.072

TABLE 11

RESULT OF SIMULATION ANALYSIS OF 100% INITIAL DESIGN CORNERING LOADING

No	Upright Design Analysis Results When the Car *Cornering*			
1	Material	*Von-mises Stress* (MPa)	Total Deformation (mm)	Safety Factor
2	Al7075-T6	13.942	0.004134	15
3	Al6061	13.942	0.0043021	15

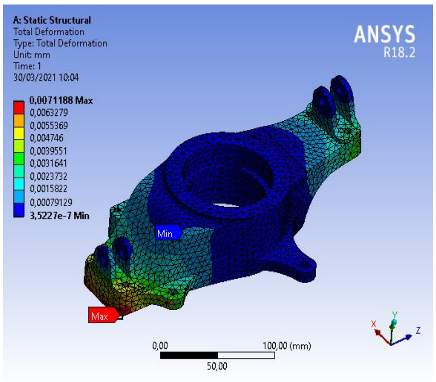

Figure 7. Simulation Result of Deformation 100% Initial Design with Material Al 7075 T6 When the Car Deceleration

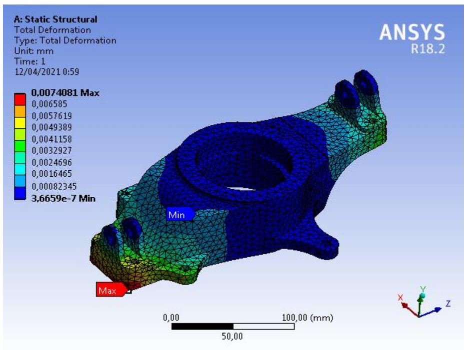

Figure 8. Simulation Result of Deformation 100% Initial Design with Material Al 6061 When the Car Deceleration

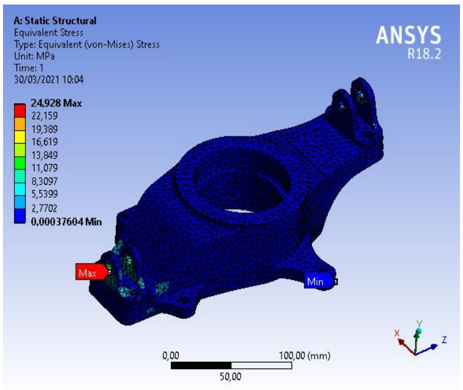

Figure 9. Simulation Result of von Mises Stress 100% Initial Design When the Car Deceleration

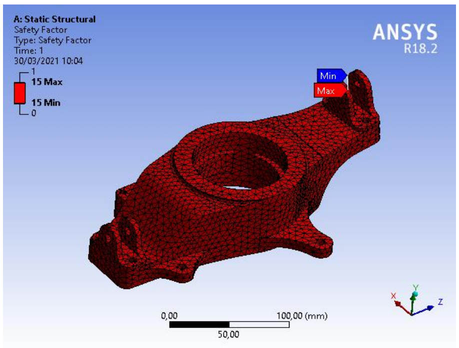

Figure 10. Simulation Result of Safety Factor 100% Initial Design with Material Al 7075 T6 When the Car Deceleration

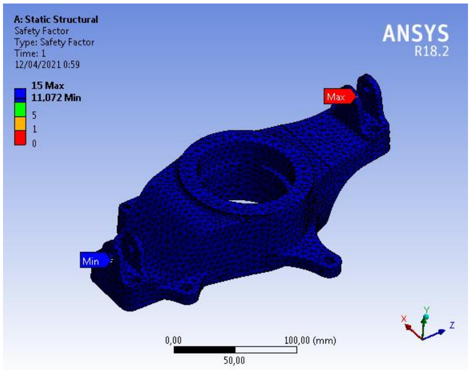

Figure 11. Simulation Result of Safety Factor 100% Initial Design with Material Al 6061 When the Car Deceleration

B. *Mass Retain 60% Design Simulation Results*

Analysis of the initial design results with 60% mass retain without topology optimization. Comparison of the analysis of the simulation results using Al 7075 T6 and Al 6061 materials. Acceleration, deceleration, and cornering load applied in that geometry.

TABLE 12

RESULT OF SIMULATION ANALYSIS OF DESIGN MASS RETAIN 60% ACCELERATION LOADING

No	Upright Design Analysis Results When the Car Accelerates			
1	Material	Von-mises Stress (MPa)	Total Deformation (mm)	Safety Factor
2	Al7075-T6	10.291	0.0060314	15
3	Al6061	10.291	0.0062766	15

TABLE 13

RESULT OF SIMULATION ANALYSIS OF DESIGN MASS RETAIN 60% DECELERATION LOADING

No	Upright Design Analysis Results When the Car Decelerates			
1	Material	*Von-mises Stress* (MPa)	Total Deformation (mm)	Safety Factor
2	Al7075-T6	39.224	0.01595	12.824
3	Al6061	39.224	0.016598	70.365

TABLE 14.

RESULT OF SIMULATION ANALYSIS OF DESIGN MASS RETAIN 60% CORNERING LOADING

No	Upright Design Analysis Results When the Car *Cornering*			
1	Material	*Von-mises Stress* (MPa)	Total Deformation (mm)	Safety Factor
2	Al7075-T6	16.091	0.0051007	15
3	Al6061	16.091	0.005308	15

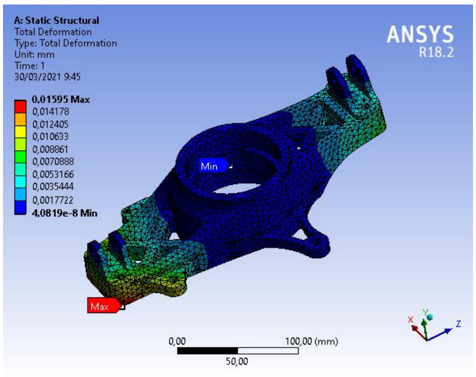

Figure 12. Simulation Result of Deformation 60% Mass Retain Design with Material Al 7075 T6 When the Car Deceleration

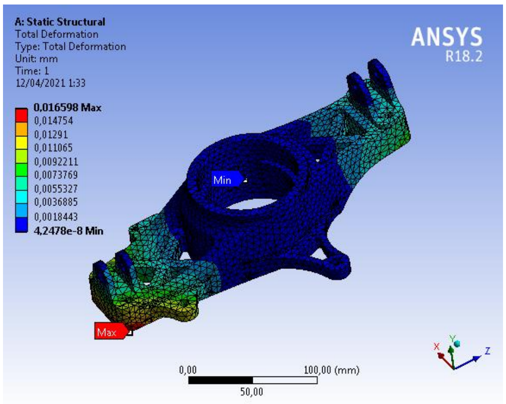

Figure 13. Simulation Result of Deformation 60% Mass Retain Design with Material Al 6061 When the Car Deceleration

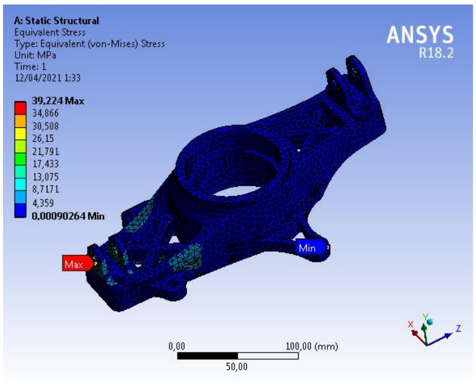

Figure 14. Simulation Result of von Mises Stress 60% Mass Retain Design When the Car Deceleration

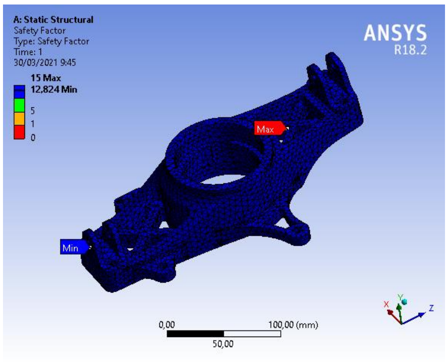

Figure 15. Simulation Result of Safety Factor 60% Mass Retain Design with Material Al 7075 T6 When the Car Deceleration

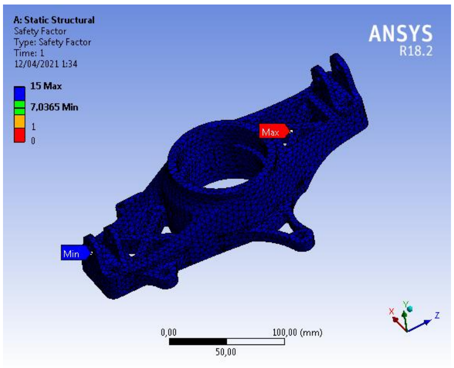

Figure 16. Simulation Result of Safety Factor 60% Mass Retain Design with Material Al 6061 When the Car Deceleration

3.3. Mass Retain 43% Design Simulation Results

Analysis of the initial design results with mass retain 43% without topology optimization. Comparison of the analysis of the simulation results using Al 7075 T6 and Al 6061 materials. Acceleration, deceleration, and cornering load applied in that geometry.

TABLE 15

RESULT OF SIMULATION ANALYSIS OF DESIGN MASS RETAIN 43% ACCELERATION LOADING

No	Upright Design Analysis Results When the Car Accelerates			
1	Material	Von-mises Stress (MPa)	Total Deformation (mm)	Safety Factor
2	Al7075-T6	12.846	0.0089706	15
3	Al6061	12.846	0.0093351	15

TABLE 16

RESULT OF SIMULATION ANALYSIS OF DESIGN MASS RETAIN 43% DECELERATION LOADING

No	Upright Design Analysis Results When the Car Decelerates			
1	Material	*Von-mises Stress* (MPa)	Total Deformation (mm)	Safety Factor
2	Al7075-T6	50.222	0.022761	10.016
3	Al6061	50.222	0.023686	4.956

TABLE 17

RESULT OF SIMULATION ANALYSIS OF DESIGN MASS RETAIN 43% CORNERING LOADING

No	Upright Design Analysis Results When the Car Cornering			
1	Material	*Von-mises Stress* (MPa)	Total Deformation (mm)	Safety Factor
2	Al7075-T6	25.037	0.073447	15
3	Al6061	25.037	0.0076431	11.024

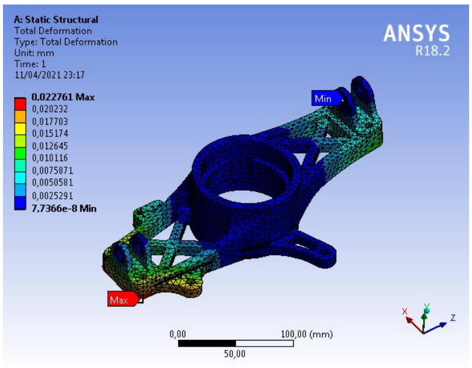

Figure 17. Simulation Result of Deformation 43% Mass Retain Design with Material Al 7075 T6 When the Car Deceleration

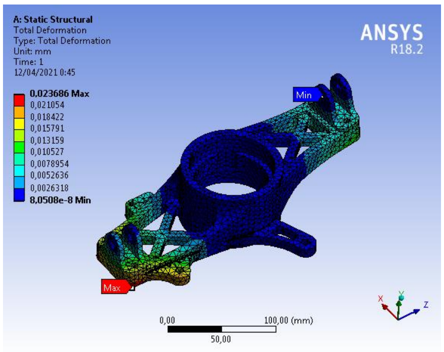

Figure 18. Simulation Result of Deformation 43% Mass Retain Design with Material Al 6061 When the Car Deceleration

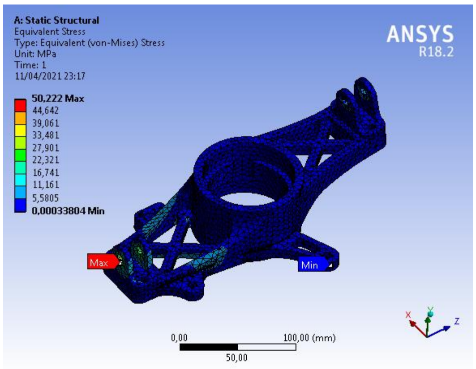

Figure 19. Simulation Result of von Mises Stress 43% Mass Retain Design When the Car Deceleration

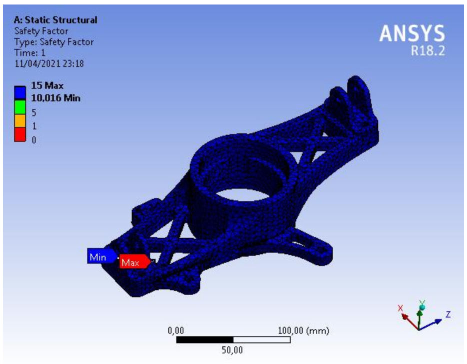

Figure 20. Simulation Result of Safety Factor 43% Mass Retain Design with Material Al 7075 T6 When the Car Deceleration

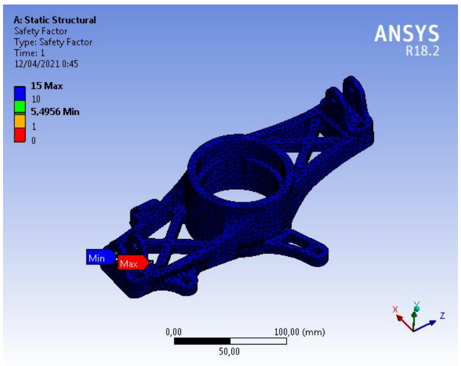

Figure 21. Simulation Result of Safety Factor 43% Mass Retain Design with Material Al 6061 When the Car Deceleration

D. *Comparison Simulation Results*

a. Von Mises Stress Comparison

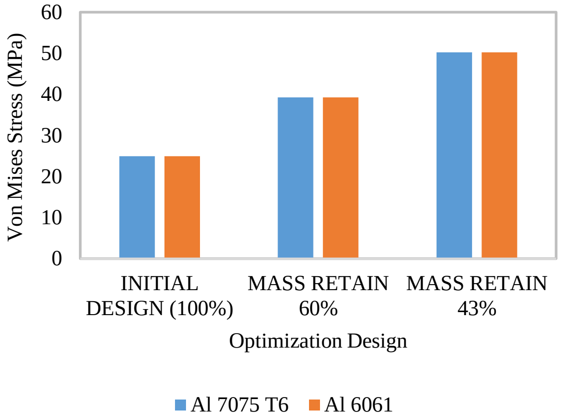

Figure 22. Comparison Simulation Result of von Mises Stress

Based on the graph of the von mises stress comparison, the upright initial design data shows the von mises stress only around 20 MPa. After upright reduce in 60% of its mass, von mises stress around 40 MPa. The last reducing mass, in 43%, von mises value shows around 50 MPa.

The graph shows that upright using mass retain 43% the maximum stress. Based on von mises stress, this value still lower than the yield stress. The 43% mass retain design can approve.

b. Deformation Total Comparison

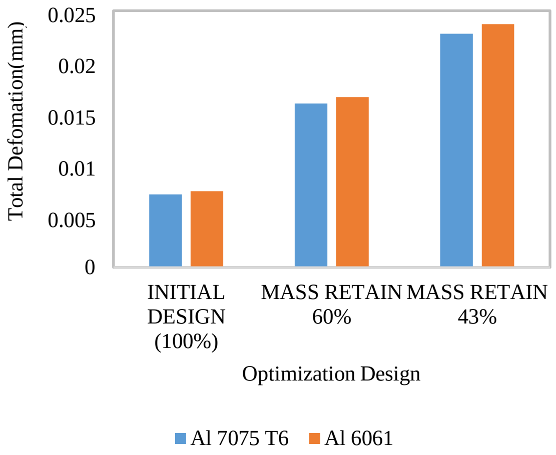

Figure 23. Comparison Simulation Result of Total Deformation

The graph shows total deformation comparation each other, the upright initial design data shows the maximum total deformation only 0.005 mm. After upright reduces in 60% of its mass, total deformation has growth up around 0.016 mm. The last reducing mass. in 43%, total deformation value still 0.21 mm.

The graph shows that upright using mass retain 43% has the maximum total deformation. The total deformation under 0.05 mm at all, it means the deformation of this component can’t use as a parameter design.

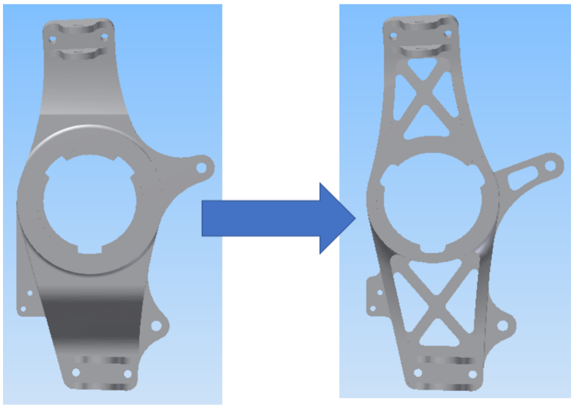
MASS: 2.1701 kg MASS: 0.95707 kg

Figure 24. Design Comparison Conclusion

The upright geometry has changed significantly, shows from Figure 24. In this research only propose the use case in topology optimization, not for manufacturing parameter. After the topology geometry lookup, continuing this study into manufacturing method or step.

IV. CONCLUSION

Based on this research, the design recommendations obtained are in design with 43% mass retain optimization. Obtained a mass reduction of 43% from 2.1701 kg become 0.95707 kg and von-mises stress value of 50.222 MPa. The geometry in this research needs more research for the manufacturing process and method.

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