PROBLEM STATEMENT

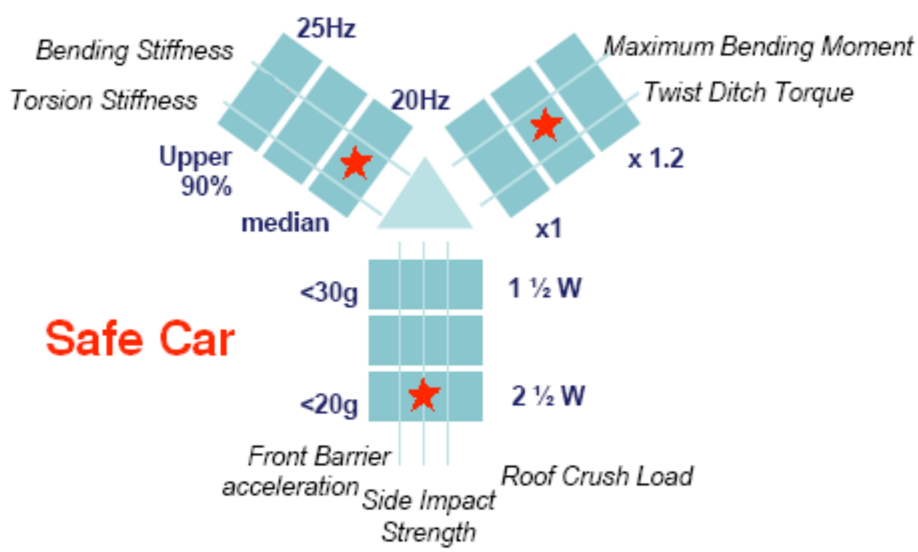
Customer Description

Our customer is a new and growing family with a limited income. The customer wants a primary transportation for the next 6 years with high degree of safety and an enough cargo space at a target price of $25000.

Step 1. Interpret the customer description

Design Description

Based on the customer requirement, our team's design focuses on the safety (energy absorbing) and durability (strength) with high fuel efficiency. Accordingly our team concentrated in developing high performance especially in the front barrier crush test, side barrier crush test, and roof crush test. The maximum number of passengers is assumed to be 5 adults each weighing 70 kg and the cargo mass is assumed to be 120 kg. The two assumptions are sufficient to consider the worst case scenario, so that our design does not incorporate any safety factor.



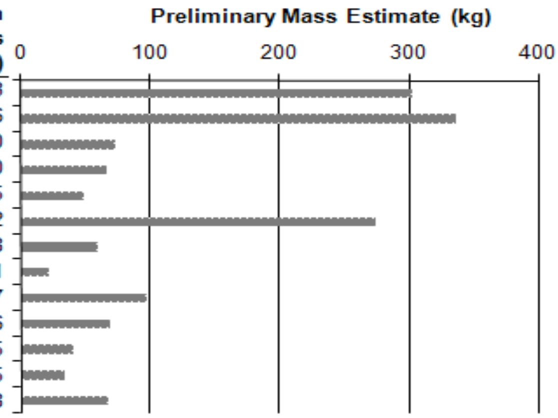
Step 2. Estimate the vehicle mass for a nominal vehicle

The length of the vehicle was measured to be 4700 mm and the width of the vehicle to be 1800 mm. Using these two dimensions with the 'MassCompRev5' provided in class, the vehicle plan view was calculated as 8.46 m^2 and the vehicle curb mass as 1479.56 kg. Along with each mass of the 5 passengers being 70kg and the cargo mass being 120 kg, the gross vehicle mass (nominal mass) was calculated to be **1950 kg**.

Table 1: Calculating Gross vehicle mass

Select Data set: Sedan

Subsystem	Subsystem Mass Fraction of Curb Mass	User Defined Mass (kg) (optional)	Preliminary Subsystem Mass (kg)
Body Non-structure	0.204		301.83
Body Structure	0.227		335.86
Front Suspension	0.049		72.50
Rear Suspension	0.044		65.10
Braking	0.032		47.35
Powertrain	0.185		273.72
Fuel & Exhaust	0.040		59.18
Steering	0.014		20.71
Tires & Wheels	0.065		96.17
Electrical	0.046		68.06
Cooling	0.027		39.95
Bumpers	0.022		32.55
Closures	0.045		66.58
Vehicle Plan View Area (m^2)		8.46	
Vehicle Curb Mass (kg)			1479.56
Number of Passengers		5	350.00
Cargo Mass (kg)			120.00
Gross Vehicle Mass			1949.56



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Step 3. Adjust the mass of subsystems

To incorporate customer's requirements into our team's design, the subsystems of a vehicle was adjusted, partially different from the nominal based needs. The Table 2 below shows the subsystems of a vehicle and adjusted subsystem.

Table 2: Adjusted subsystems based on customer requirement

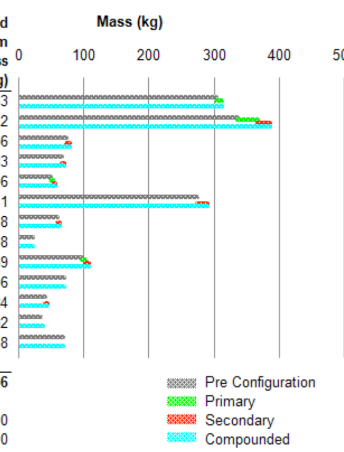
Subsystems	Component	Estimated mass change(kg)
Body Non-Structure	Safety – air bag	+10kg
Body Structure	Safety – material	+30kg
Front Suspension	·	·
Rear Suspension	·	·
Braking	Safety – material	+5kg
Power Train	·	·
Fuel & Exhaust	·	·
Steering	·	·
Tires & Wheels	Safety – size increase	+5kg
Electrical	·	·
Cooling	·	·
Bumpers	Safety – size increase	+2.5kg
Closures	·	·
Passenger Mass	·	·
Cargo Mass	·	·

Taking into account of the customer requirement resulted in the compounded gross vehicle mass as to be **2061.36 kg** as shown in the Table 3.

Table 3: Compounded vehicle mass

Select data set: Ratio

Subsystem	Influence Coefficient	User Defined Influence Coefficient (optional)	resize sub-system ?	Pre-Config. Subsystem Mass (kg)	Adjustments to Pre-Config. Mass	Compounded Subsystem Mass (kg)
Body Non-structure	0.0000			301.83	10.00	311.83
Body Structure	0.1723		☑	335.86	30.00	385.12
Front Suspension	0.0372		☑	72.50		76.66
Rear Suspension	0.0334		☑	65.10		68.83
Braking	0.0243		☑	47.35	5.00	55.06
Powertrain	0.1404		☑	273.72		289.41
Fuel & Exhaust	0.0304		☑	59.18		62.58
Steering	0.0060		☑	20.71		21.38
Tires & Wheels	0.0493		☑	96.17	5.00	106.69
Electrical	0.0000			68.06		68.06
Cooling	0.0205		☑	39.95		42.24
Bumpers	0.0167		☑	32.55	2.50	36.92
Closures	0.0000			66.58		66.58
Sum Inf. Coeff	0.5304					
Vehicle Curb Mass				1479.56		1591.36
Passenger Mass				350.00		350.00
Cargo Mass				120.00		120.00
Gross Vehicle Mass				1949.56		2061.36
Delta Mass from Pre-Config.					52.50	111.80



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Step 4. Determine numerical structural requirements

(a) Bending moment diagrams

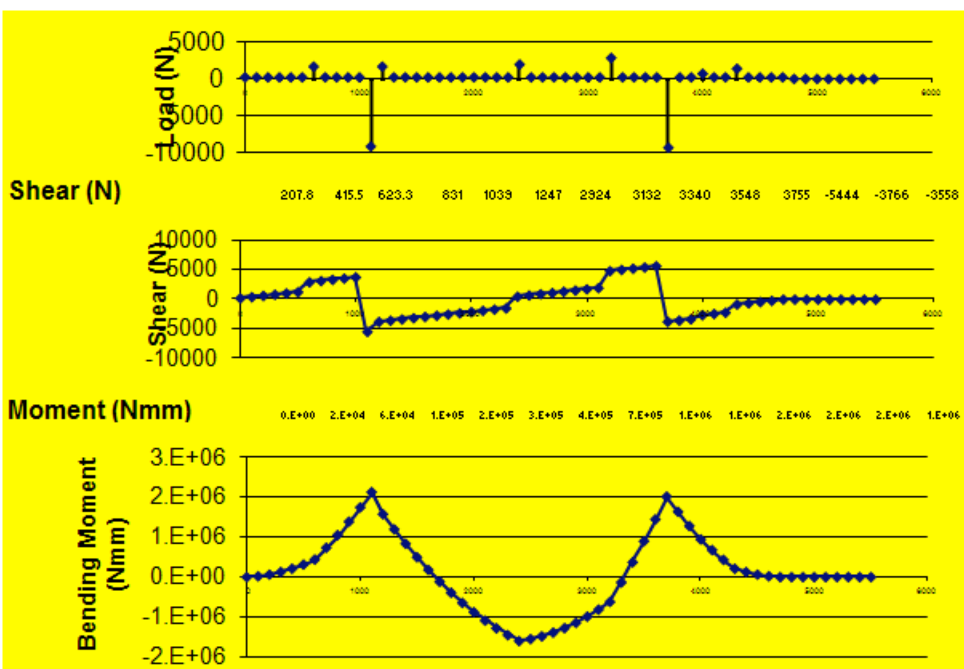
Dimensions of a vehicle were determined to calculate the bending moment and shown in the Table 4. We assumed the mass of an engine to be 300 kg, a seat to be 20 kg, and a passenger to be70 kg.

Table 4: Dimensions of a vehicle

	X(mm)	Load(kg)
Front	0	·
Front Engine Mount	600	150
Front Suspension	1100	76.66
Rear Engine Mount	1200	150
Seat and 2 Passengers	2400	180
Seat and 3 passengers	3200	270
Rear Suspension	3700	68.83
Fuel	4000	50
Cargo	4300	120
End	4700	·
Uniform Load	·	21.2kg/100mm

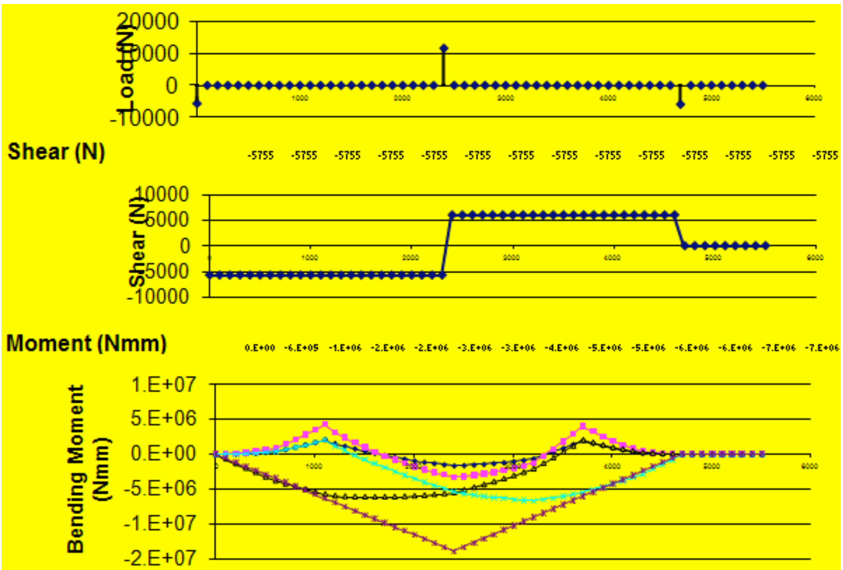
Static loading

Length Increment	100	mm			
Front End Coordinate	0				
Front Support Position	1100		Front Support Load	10158.4	N
Rear Support Position	3700		Rear Support Load	10255.9	N
End Coordinate	4700		Total Weight	20414.3	N



(B) Bending strength load for the H point bending test

Length Increment	100	mm		
Front End Coordinate	0			
Front Support Position	0		Front Support Load	5754.9 N
Rear Support Position	4700		Rear Support Load	6005.1 N
End Coordinate	4700		Total Weight	11760.0 N



Bending moment diagram of single point load 1200kg at 2400mm covers all the maximum value of various bending moment diagram (1g, 2g, front towing and rear towing). Therefore, bending strength requirement should be $1200\ kg \times 9.8\ m/s^2 = \mathbf{11760}\ N$.

(C) Twist ditch torque

Track = 1560 mm

Length Increment	100	mm		
Front End Coordinate	0			
Front Support Position	1100		Front Support Load	10158.4 N
Rear Support Position	3700		Rear Support Load	10255.9 N
End Coordinate	4700		Total Weight	20414.3 N

Front and rear support load is calculated from the spreadsheet for bending moment calculation. As we can see rear support load is heavier than front support load. We should use rear support load for calculation of twist ditch torque.

$$\text{Twist ditch torque} = \frac{1}{2} \times \text{track} \times W_{\text{axle}}$$

$$= \frac{1}{2} \times 1560\ \text{mm} \times 10255.9\ \text{N}$$

$$= 7999602\ \text{Nmm}$$

$$\approx 8000000\ \text{Nmm}$$

$= \mathbf{8000\ N}$

(D) Bending stiffness requirement

The required bending stiffness is determined by the desired bending frequency of the car and the final vehicle mass. Using a first order model of bending frequency, we find that:

$$\omega_n = 3.233\left(\frac{l}{L}\right)^{\frac{3}{2}}\sqrt{\frac{K}{M}}$$

ω_n is the natural frequency of the beam, l is the distance between the suspension attachment points, L is the total length of the car, and M is the rigidly mounted mass of the vehicle mass which is 60% of the total mass of the vehicle. Rearranging this equation, we find:

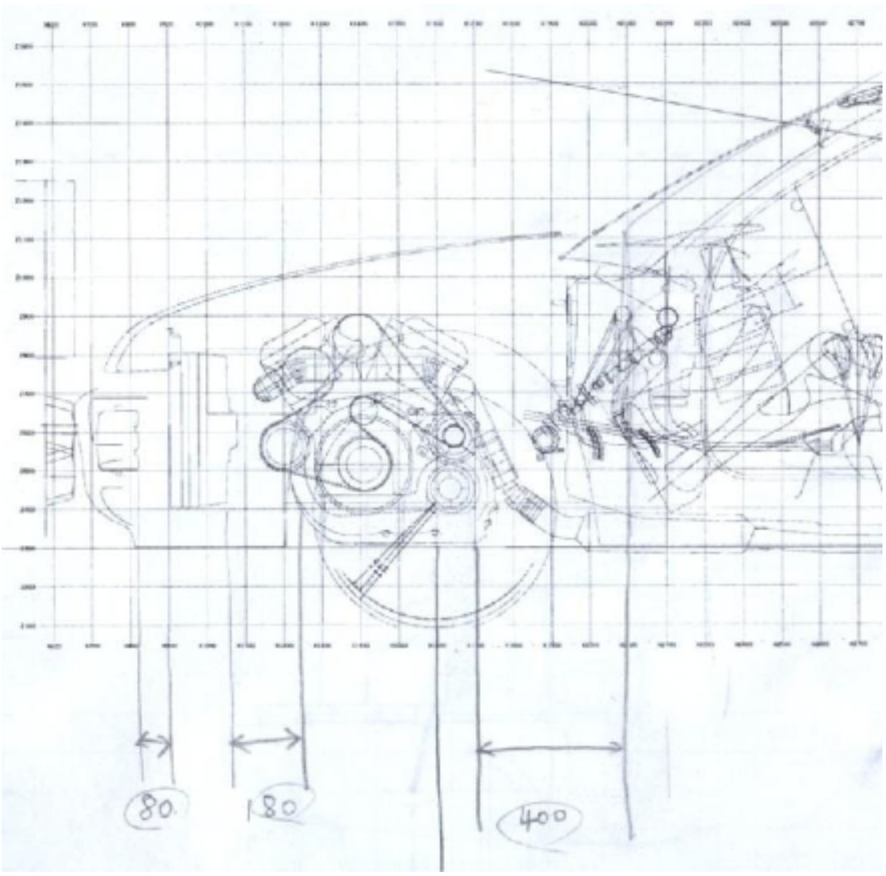
$$K = .096\omega_n^{\ 2}M\left(\frac{L}{l}\right)^3$$

From the noise and vibration mode map for sources and responders for vehicles, it was determined that the desired natural frequency lies in the range of 22-25 Hz. We choose 22 Hz as our goal since our objective does not require an extremely stiff car body. All other parameters are known and thus we are able to determine our required bending stiffness by substituting. Bending stiffness K is found to be 12644 N/mm. This value is divided by 2 to consider one side frame of the vehicle. Then **K = 6322 N/mm** is used to compare to the simulated stiffness.

(E) Torsional stiffness requirement

Customer does not want the vehicle with precise handling but one with good handling.
We set the torsional stiffness with nominal value, **12000 $Nm/^\circ$**

(F) Front crush space available



Crush space (Δ) = 660mm

(G) The average front barrier crush force required

Crush space (Δ) = 660mm

Velocity (V) = 13.3 m/sec (30 mph)

Assume a_{max} = 20g

[We determine maximum allowable cabin deceleration in order to avoid injury because the customer wants to buy safe car.]

$$a_{max} = \frac{v^2}{2} \times \frac{1}{\eta} \times \frac{1}{\Delta}$$

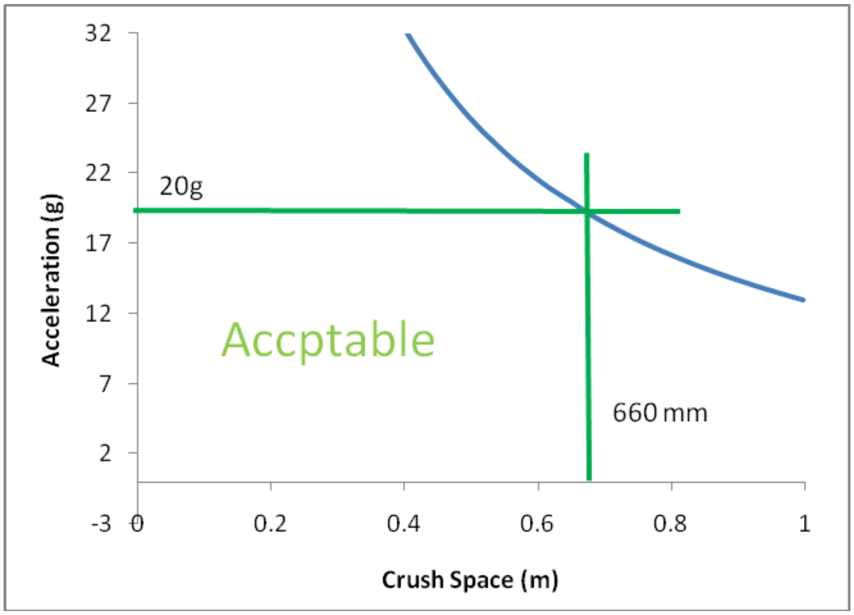
$$\eta = \frac{v^2}{2\Delta a_{max}} = \frac{(13.3\ m/s)^2}{2\ \times 0.66\,m\ \times 20\ \times 9.8\ m/s^2} \approx 0.68$$

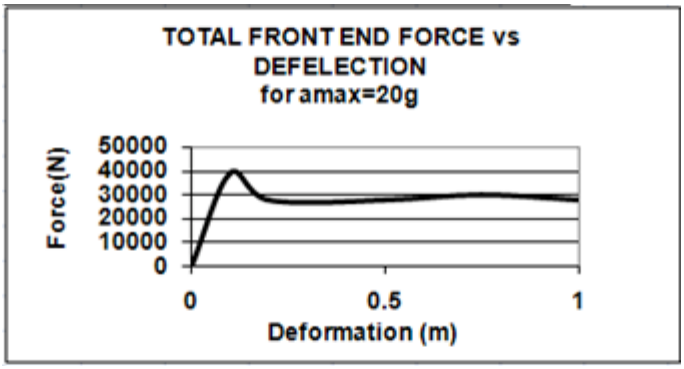
η= 0.70% which is well within achievable.

F_{max} = M x a_{max} = 2062 kg x 20 x 9.81 = 404565 N

F_{avg} = η x F_{max} = 0.70 x 404565 = 283196 N

The crush load taken by each of the front 2 crash members is 25% of F_{avg} = 70799 N (Pm)





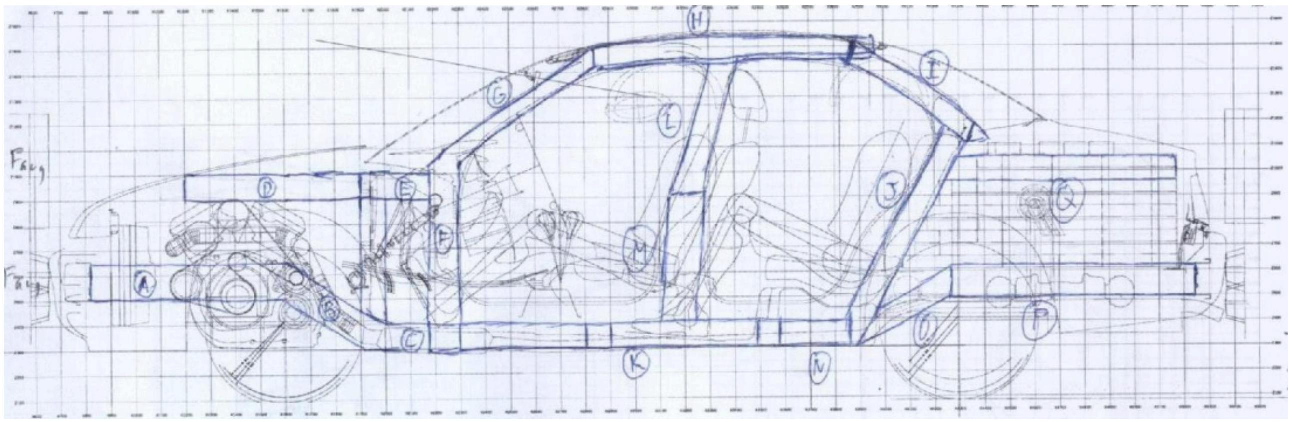
(H) Roof Crush

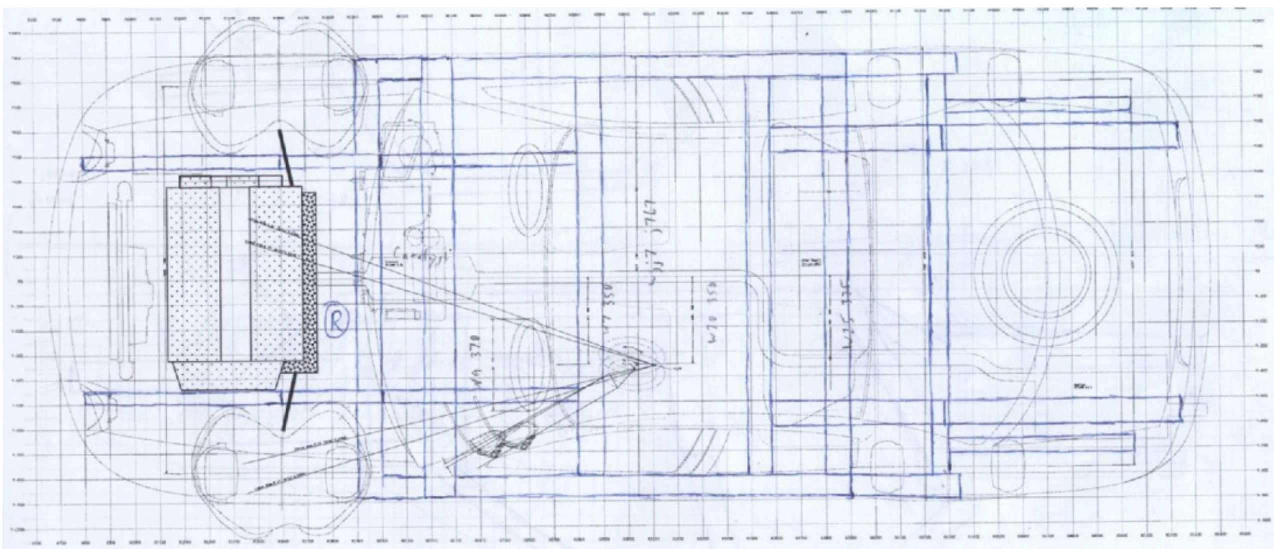
Maximum of 5 inches of deformation at static load of 2.5 times vehicle weight.

Step 5. The first order model analysis

(A) Designing process

Body Structure Concept

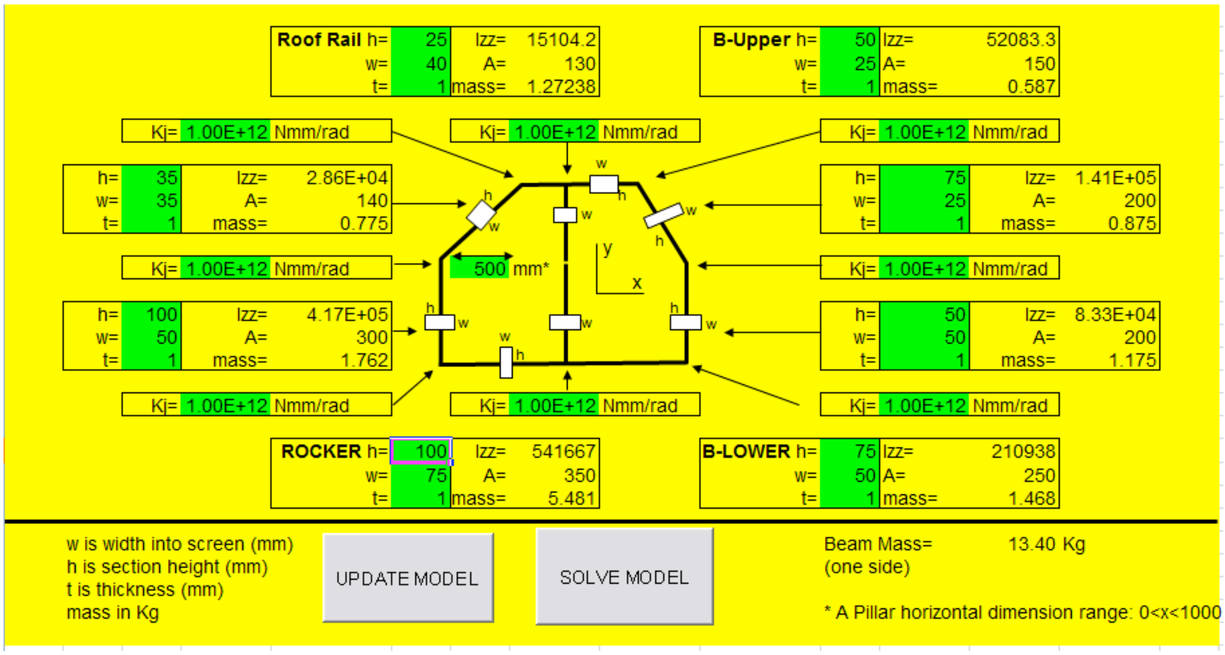


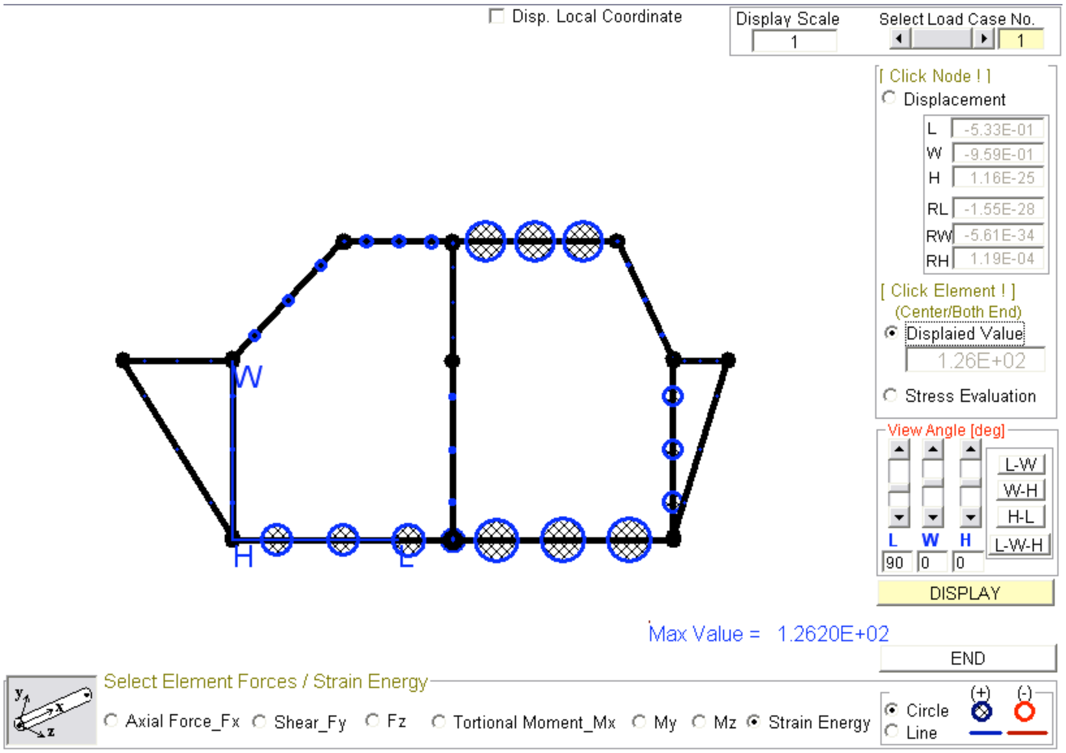


Because our primary objective was safety in our car design, the curb weight of our vehicle was slightly higher than average due to the added features. The crush area was increased to allow a lower deceleration of the entire vehicle during a crash. We also provided for the extra safety features to be added to the car such as airbags. All of this added weight thus required larger beam sections elsewhere in the car to meet the bending and torsion requirements. In the front section of the vehicle, where the driver's feet were to be located, we made changes to the structure to allow for more room. Beams were thus added in this area to support the crushable beams to ensure the driver would be safe in a crash. Our vehicle was optimized to meet all structural requirements using only rectangular cross section beams since this basic shape is one of the easiest to manufacture. However in future work, using more refined cross section shapes would create a more mass efficient design. Areas that would benefit from this would be locations such as the crush areas where a hexagonal or octagonal shape could be used to more efficiently absorb energy. Also, in our design, the joints were not optimized to be the most efficient in strength they could be. By adding fillets or blending features to the areas where beams meet one another, joints could be strengthened and the body could be made stiffer. Adding internal ribs to the joint areas would also strengthen these areas significantly. These changes would reduce the total mass of the vehicle by allowing smaller beams to be used.

(B) Size the sections

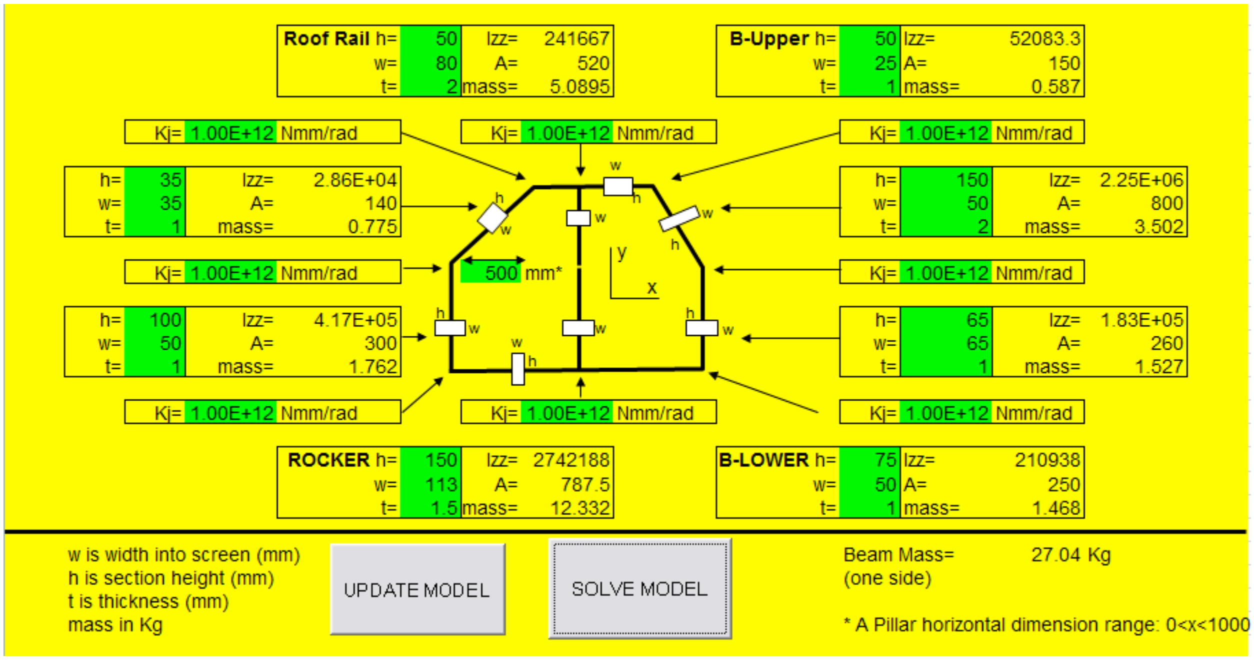
[1] Design for bending stiffness

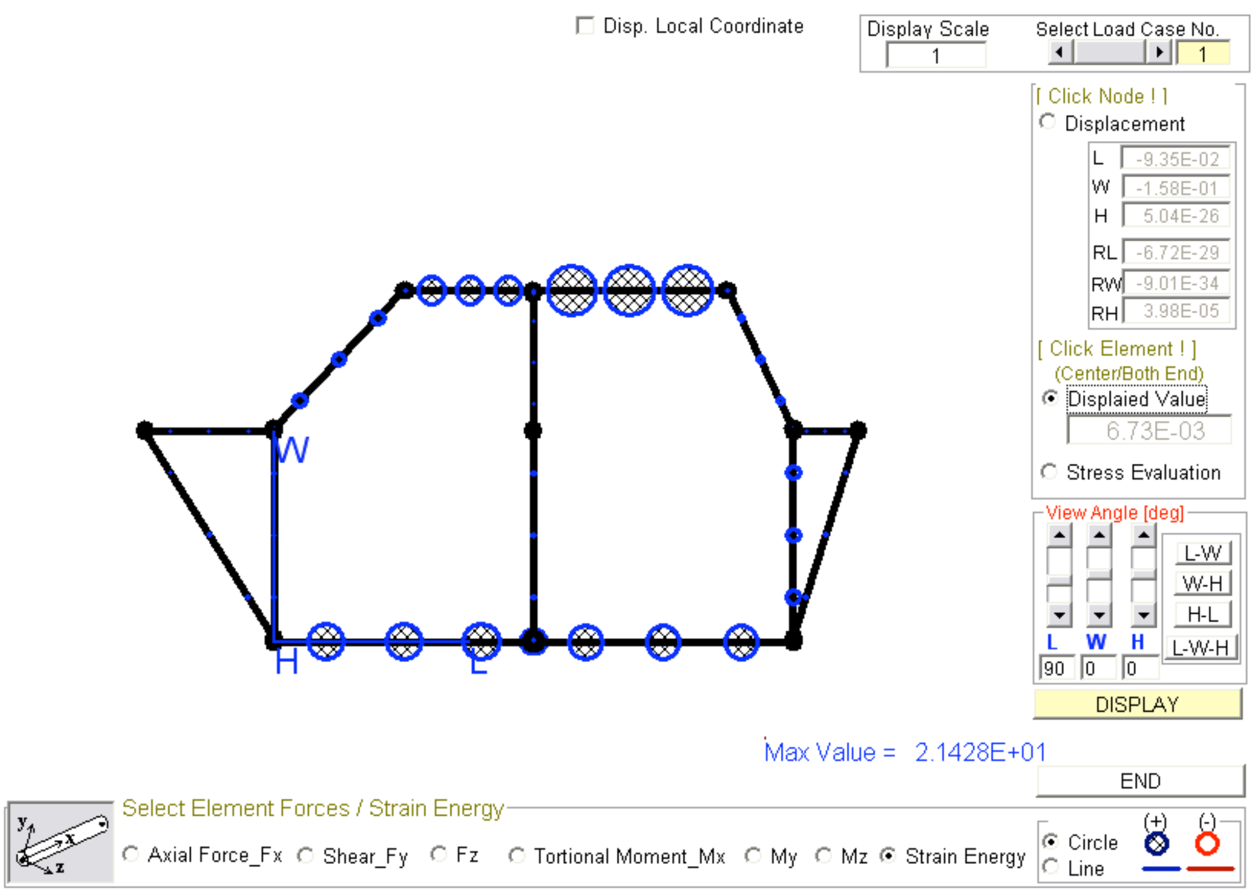




The stiffness calculated from the spread sheet with default dimensions above is 1042.75 N/mm. This simulated stiffness is less than the required bending stiffness of 6322N/mm calculated by Eq 1. To meet the stiffness requirement the beam with the highest strain energy must be resized. The right part of the rocker has the highest strain energy with a magnitude of 1.26 $\times 10^2$ N·mm. Ideally, increasing the height of a beam is the most efficient way to increase the stiffness because increasing the moment of inertia of the beam decreases the stresses In the beam. Then the beam can absorb more external force, so that the stiffness of the beam increases. However, due to space limitations, not only a moderate magnitude of the height of the rocker is increased but also that of the width and thickness are increased by 50%. After this iteration, the maximum strain energy is in the roof rail, so that the width, height, and thickness of the roof rail are increased by 100% but the stiffness still doesn't meet the requirement. After this iteration the highest strain energy is in the C-pillar upper beam, so that the width, height, and thickness of the C-pillar upper beam are increased by 100% but the stiffness still doesn't meet the requirement. In the end when we changed the height of the C-pillar by 30% and the width of the C-pillar by 30%, the simulated stiffness exceeds the requirement stiffness.

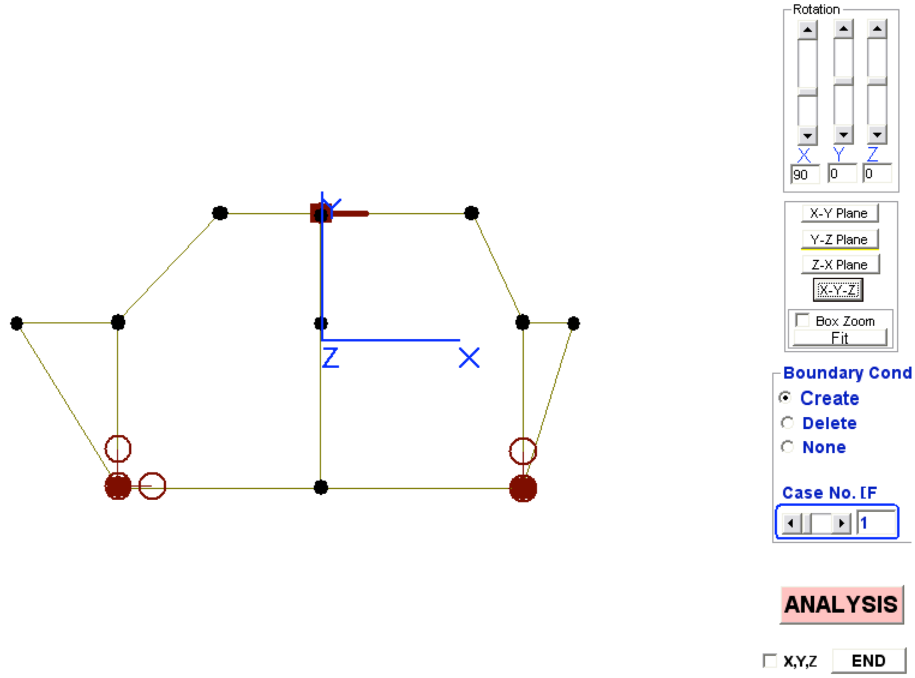
Rocker	h,w,t => 50 % increase
Roof rail	h,w,t => 100% increase
C-pillar upper	h,w,t =>100% increase
C-pillar lower	h,w => 30% increase, t =>0% increase
Simulated Stiffness	6329 N/mm
Required Stiffness	6322 N/mm

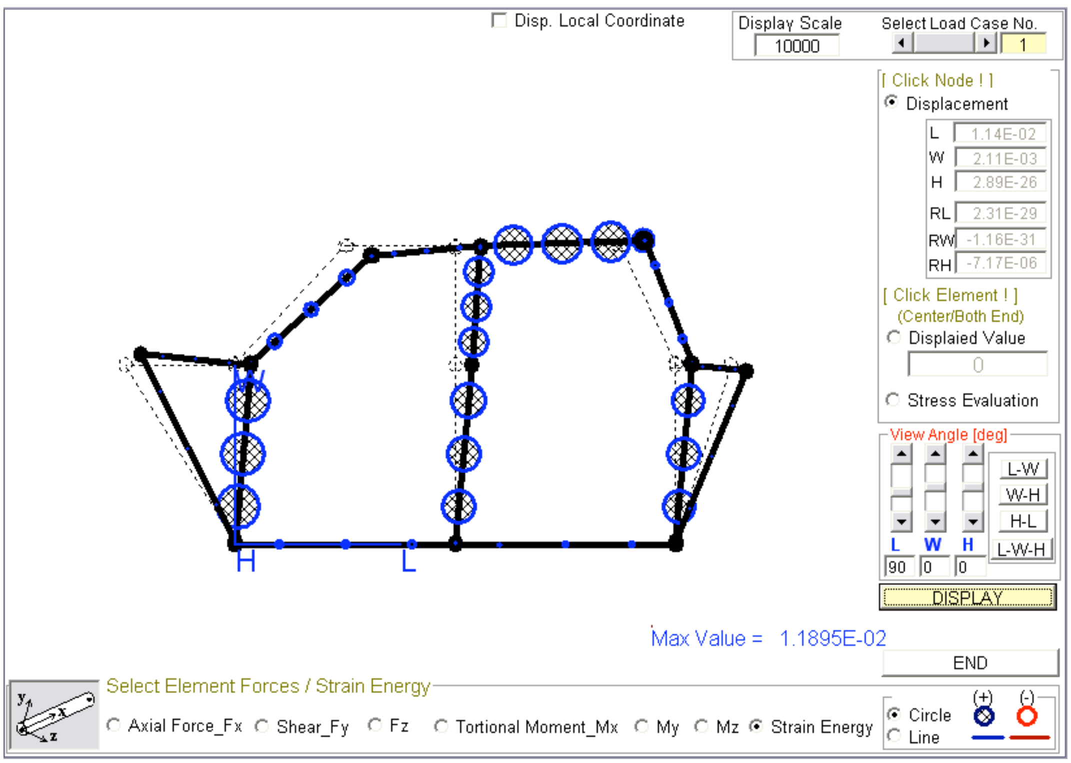

Roof Rail h= 50 Izz= 241667
w= 80 A= 520
t= 2 mass= 5.0895
B-Upper h= 50 Izz= 52083.3
w= 25 A= 150
t= 1 mass= 0.587
Kj= 1.00E+12 Nmm/rad
h= 35 Izz= 2.86E+04
w= 35 A= 140
t= 1 mass= 0.775
h= 150 Izz= 2.25E+06
w= 50 A= 800
t= 2 mass= 3.502
500 mm*
h= 100 Izz= 4.17E+05
w= 50 A= 300
t= 1 mass= 1.762
h= 65 Izz= 1.83E+05
w= 65 A= 260
t= 1 mass= 1.527
ROCKER h= 150 Izz= 2742188
w= 113 A= 787.5
t= 1.5 mass= 12.332
B-LOWER h= 75 Izz= 210938
w= 50 A= 250
t= 1 mass= 1.468
w is width into screen (mm)
h is section height (mm)
t is thickness (mm)
mass in Kg
UPDATE MODEL
SOLVE MODEL
Beam Mass= 27.04 Kg
(one side)
* A Pillar horizontal dimension range: 0<x<1000


Disp. Local Coordinate
Display Scale 1
Select Load Case No. 1
[Click Node !]
Displacement
L -9.35E-02
W -1.58E-01
H 5.04E-26
RL -6.72E-29
RW -9.01E-34
RH 3.98E-05
[Click Element !]
(Center/Both End)
Displaied Value
6.73E-03
Stress Evaluation
View Angle [deg]
L-W
W-H
H-L
L-W-H
L W H
90 0 0
DISPLAY
Max Value = 2.1428E+01
END
Select Element Forces / Strain Energy
Axial Force_Fx Shear_Fy Fz Tortional Moment_Mx My Mz Strain Energy
Circle
Line
(+) (-)

[2] Design for torsional stiffness

- Shear stiffness of side frames





$$(Gt_{eff}) = \frac{F\,H}{\delta\,L} = \frac{10\ N}{0.0114\ mm}\frac{1125\ mm}{2000\ mm} = 493.42\,{}^{N}/_{mm}$$

Stiffness of shear panel vehicle

Panel	Area of Panel (mm2)	Effective Shear Stiffness Gt(N/mm)	Shear Flexibility Contribution (mm3/N)	Pecent Contribution to Shear Flexibility	
Dash	1260000	80000	15.8	0.2%	
Winshield	1323490	80000	16.5	0.2%	
Roof	1800000	80000	22.5	0.3%	
Back Light	927699	80000	11.6	0.1%	
Seat Back	1440000	80000	18.0	0.2%	
Floor	3600000	80000	45.0	0.5%	
Side Frame-Left	2057500	493.42	4169.9	49.2%	
Side Frame-Right	2057500	493.42	4169.9	49.2%	
		sum	8469.1	100.0%	
	angle of twist (rad)		4.94E-03	rad	Under T= 8.00E+06 Nmm
	torsional stiffness		1.62E+09	Nmm/rad	
	torsional stiffness		28264	Nm/deg	

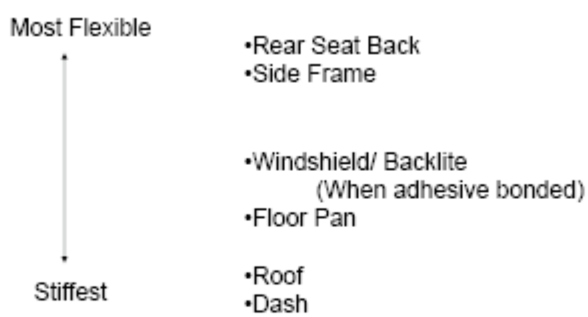
Torsioinal stiffness of the vehicle with side frame is 28264 $Nm/°$ which much higher than requirement 12000 $Nm/°$. We need to change the design of the panels to make effective shear stiffness Gt lower. For example, we can add sun roof to vehicle which makes roof panel more flexible. Or we can design car with folded rear seat that we can put larger cargo into cabin area through seat back. Also, glass will be put into windshield and back light. Those panels are not flat panel anymore. We can assume them with rectangular beam structure.

Stiffness of shear panel vehicle

Panel	Area of Panel (mm2)	Effective Shear Stiffness Gt(N/mm)	Shear Flexibility Contribution (mm3/N)	Pecent Contribution to Shear Flexibility
Dash	1260000	80000	15.8	0.1%
Winshield	1323490	500	2647.0	13.4%
Roof	1800000	5000	360.0	1.8%
Back Light	927699	500	1855.4	9.4%
Seat Back	1440000	300	4800.0	24.2%
Floor	3600000	2000	1800.0	9.1%
Side Frame-Left	2057500	493.42	4169.9	21.0%
Side Frame-Right	2057500	493.42	4169.9	21.0%
		sum	19817.9	100.0%

angle of twist (rad)	1.16E-02	rad	Under T= 8.00E+06 Nmm
torsional stiffness	6.92E+08	Nmm/rad	
torsional stiffness	12078	Nm/deg	

If we can design each panel with above value of effective shear stiffness Gt, we can meet the vehicle torsional stiffness 12000 $Nm/^{\circ}$. Ranking of the above values seem to be reasonable according to typical ranking of panel shear stiffness from the lecture note.



[3] Design for bending and torsional strength

[4] Crashworthiness

- Front barrier crush test

• Motor compartment side rails

From the GA drawing, maximum height and width of mid-rail we can get are 100mm and 50mm.

$$\mathrm{P_m} = 386 \times t^{1.86} \times b^{0.14} \times \sigma_y{}^{0.57}$$

$$t = \left(\frac{P_m}{386 \times b^{0.14} \times \sigma_y{}^{0.57}}\right)^{\frac{1}{1.86}}$$

$$t = \left(\frac{70800}{386 \times 60^{0.14} \times 207^{0.57}}\right)^{\frac{1}{1.86}}$$

$$t \approx 2.37\ mm$$

• Reaction member

$$\mathrm{F_{limit}}\,\delta = \mathrm{M_P}\big(\theta_1 + (\theta_1 + \theta_2)\big)$$

$$\mathrm{F_{limit}}\,\delta = \mathrm{M_P}\left(\frac{\delta}{\mathrm{L_1}\sin\varphi} + \left(\frac{\delta}{\mathrm{L_1}\sin\varphi} + \frac{\delta}{\mathrm{L_2}\tan\varphi}\right)\right)$$

$$\mathrm{F_{limit}} = \mathrm{M_P}\left(\frac{2}{\mathrm{L_1}\sin\varphi} + \frac{1}{\mathrm{L_2}\tan\varphi}\right)$$

From the previous calculation,

$$\mathrm{F_{limit}} \geq \frac{\mathrm{F_{max}}}{4} = \frac{404565\ N}{4} = 101142\ N$$

From the safety view, we'll set $\mathrm{F_{limit}}$ to be the value of 105000N.

$$\mathrm{M_P} = \frac{\mathrm{F_{limit}}}{\left(\frac{2}{\mathrm{L_1}\sin\varphi} + \frac{1}{\mathrm{L_2}\tan\varphi}\right)}$$

$$\mathrm{M_P} = \frac{105000\ N}{\left(\frac{2}{416mm \times \sin 32.7°} + \frac{1}{115mm \times \tan 32.7°}\right)} \approx 4.7 \times 10^6 Nmm$$

$$\mathrm{M_P} = \frac{3}{2}\sigma_y b^2 t$$

$$t = \frac{2}{3}\frac{M_p}{\sigma_y b^2} = \frac{2 \times 4.7 \times 10^6\ Nmm}{3 \times 207\ N/mm^2 \times (100mm)^2} \approx 1.51mm \approx 1.6mm$$

- Side barrier crush test

- Roof crush test

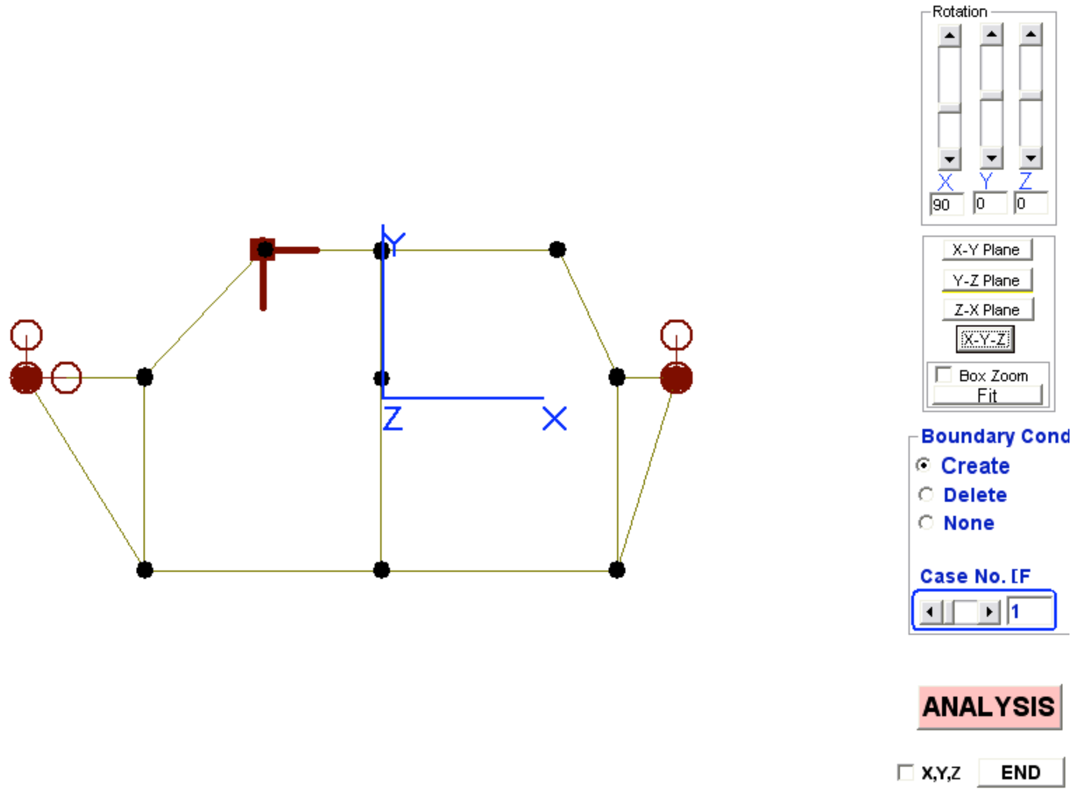
We set the roof crush load to be 2.5 time the vehicle weight for more safety.

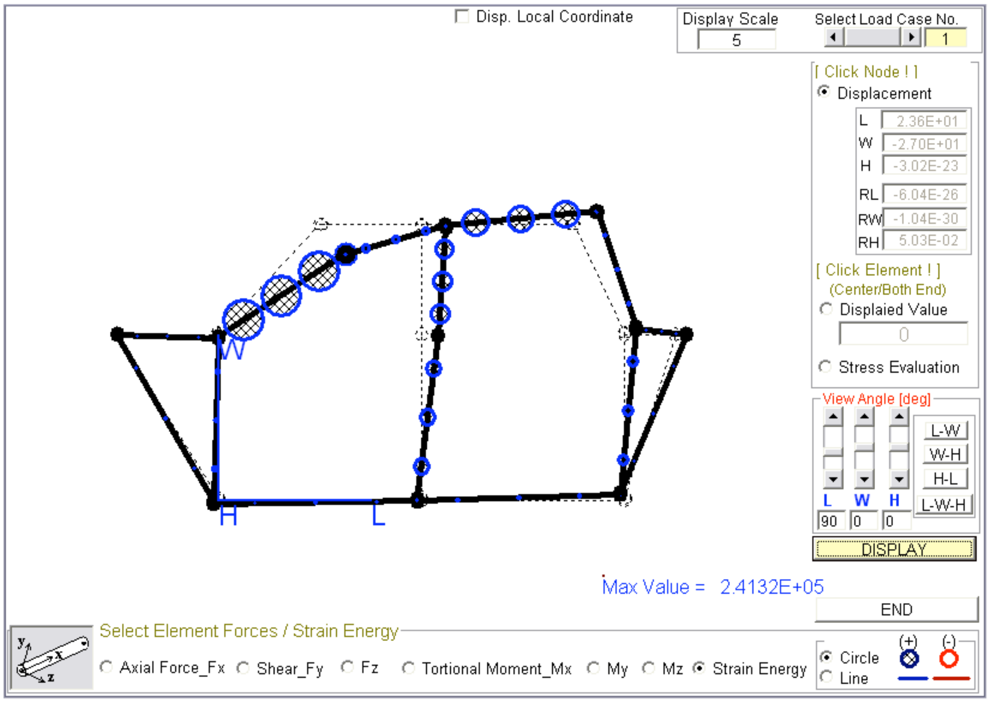
$F = 2.5 \times W = 2.5 \times 2062\ kg \times 9.8\ m/s^2 = 50519\ N$

We just did 2-dimensional analysis with FEA software as we can't do the 3-dimensional analysis. The load will be applied to the joint between the roof rail and the A pillar. The angle between the load and the vertical to the roof rail is 5°. Applied load is divided into X component and Y component.

$F_x = F \sin\theta = 50519\ N \times \sin 5° = 4403\ N$

$F_y = F \cos\theta = 50519\ N \times \cos 5° = 50327\ N$





Combined deformation at the joint between the roof rail and the A pillar is

$$\delta = \sqrt{(23.6\ mm)^2 + (27.0\ mm)^2} = 35.86\ mm$$

, which is much lower than 5 inches (127mm).

(C) Select the specific materials for the major structural elements

Step. 6,7 Packaging and manufacturing constraints

In sizing our beams, we paid careful attention to other vehicle subsystems. We sized our mid rail so it would be small enough to fit between the front wheels and the engine but large enough to be as efficiently strong as possible. The horizontal beams that support the mid rail were carefully positioned under the seats so they would allow the most room in the passenger compartment as possible while still meeting structural requirements. In the rear part of the vehicle, the mid rail beams as well as the upper crush beams were placed in positions that would allow the greatest amount of luggage space possible. In the rest of our design, we tried to remain within the space allowed to us in the parts location drawings. We ensured that the cross section of our beams met all space requirements so they would not contact any other parts. We also sized our beams in the side frame so they would fit to the car drawing dimensions within 50mm. At the same time, we made all of the beams straight to ensure creation of the parts is as easy as possible to manufacture. Our design intention is to have all the beams formed by welding two c beams together. This allows quick and reliable manufacturing of parts. The weld joints are also in easily accessible areas making manufacturing of the parts possible.

After FEA analysis, it was determined that the required roof rail beam size was smaller than the original size allowed. So we inspected the possibility of roof crush due to small height but our calculations done in Step 5 meet the requirement of the roof crush test. Therefore, the roof rail did not need to be modified.

On the other hand, the required rocker beam size was determined to be larger than the original size allowed. There are two options in increasing the size of the rocker beam. First, the rocker beam can be increased entirely downward, which will increase the possibility of damaging the bottom part of the vehicle due to small clearance from the ground. Therefore, this case was not considered for our design and took the other method that increases the rocker beam size entirely upward. This increased rocker beam size would decrease the door frame space making it less comfortable for passengers to enter and exit the vehicle. To resolve this problem, high strength steel could have been used. However, in calculations, it was determined that the required roof rail beam size was smaller than the allowed size. This extra space allows the rocker to be increased in height without decreasing the overall ingress/egress space. Thus normal strength steel can be used in the rocker beam.

The upper C-pillar was slightly increased beyond the original allowed space for structural reasons. This decreases the driver's field of view but not significantly. However, if we had been required to fit within the original allotted space, high strength steel could have been used to meet the requirements. Ribs and/or foam could also have been added in the upper C-pillar beam to make it stronger without changing the allowed original size of the beam.

The lower C-pillar size was determined to fit within the original allowed space.

The required mid-rail size exceeded the space between the mid-rail and the engine. This problem can be resolved by changing the section shape of the mid-rail from a square to a rectangular. However, this change in the section shape will increase the thickness of the mid-rail. Therefore, the total mass of the mid-rail will increase which will decrease the fuel efficiency of the vehicle.

Step.8 Estimate the structure mass

Element	length [mm]	b [mm]	t [mm]	density [kg/mm^3]	mass [kg]
Side frame					27.04
Motor compartment mid-rail - front	660	100	2.37	7.86E-06	4.92
Reacti on member -front	1231	100	1.6	7.86E-06	6.19
Motor compartment mid-rail - rear	500	100	2.37	7.86E-06	3.73
Reacti on member -rear	1247	100	1.6	7.86E-06	6.27
				Sum	48.15