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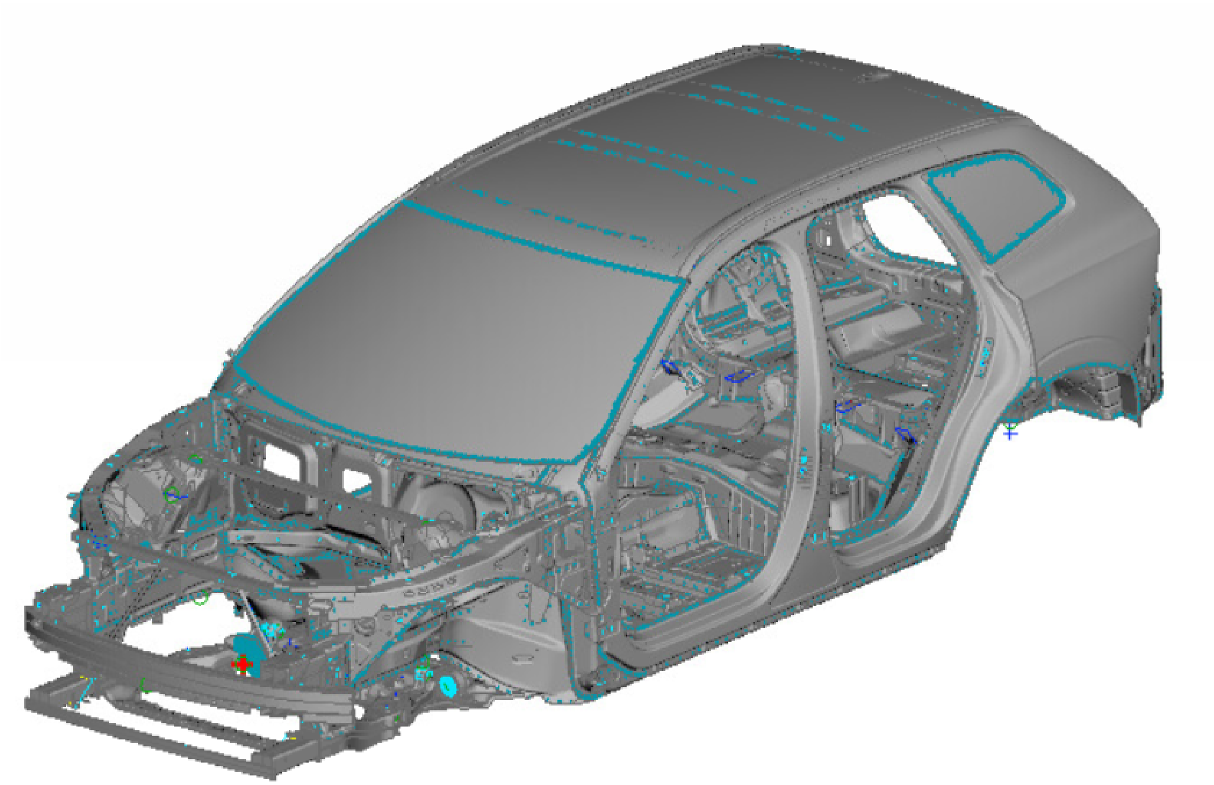

Body Torsional Stiffness - Validation & Design Optimization

Master's thesis in Applied Mechanics Master Programme

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Gothenburg, Sweden 2019

MASTER'S THESIS 2019:49

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Cover: Body Torsional Stiffness - Validation & Design Optimization.

Typeset in LaTeX
Gothenburg, Sweden 2019

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Abstract

This Thesis is a validation work, wherein, the body torsional stiffness calculated from a test-rig experiment is validated against the simulations. Earlier, the body torsional stiffness was calculated in the test-rig using a Body-In-Gray (BIG) structure (see section 2.2) and this method correlated exactly with the CAE simulation method. Hence, this test-rig experiment method was stopped and the simulations were enough for evaluating the body torsional stiffness. Later, a new test-rig experiment method (current method) was introduced, in which a full-scale car model is used to evaluate the body torsional stiffness. This method was introduced mainly to measure and compare the body torsional stiffness of the competitors' models. As this method did not correlate well with the CAE simulation, the impact of the Chassis on the BIG model is investigated. This is done to get a better correlation of the results between the simulation and the test-rig experiment. Also, to evaluate the body torsional stiffness similar to that of the test-rig method, a new simulation method is introduced during the thesis. Finally, the results from both the simulations are compared against the test-rig method to recommend the best simulation method for validating the test-rig experiment.
As a second part of the thesis, a Design Optimization study is performed to reduce the weight of the BIG structure with no change in the body torsional stiffness.

Keywords: validation, body torsional stiffness, correlation, test-rig experiment, optimization, reduced mass.

Acknowledgement

The thesis work was carried out at the Solidity department, Volvo Cars Corporation in collaboration with the Division of Dynamics and the Department of Mechanics and Maritime Sciences at Chalmers University of Technology.

We would like to express our sincere gratitude to everyone who has helped us at different stages of the thesis work in making it a memorable experience.

We would to like thank **Anneli Rosell** and **Henrik Puhasmagi**, our supervisors at Volvo Cars for the support and confidence in us throughout the thesis work. We would like to thank **Rickard** who helped us provide ADAMS models for our thesis work, without which we could not have progressed with the work and got the crucial results. We would like to thank **Prof. Thomas Abrahamsson** for the continuous support and motivation at various stages of the thesis work.

We would like to thank **Mikael Holmgren** at Volvo Cars, who helped us perform the test-rig experiments. We would like to thank **Alexander Govik**, who helped us with the design optimization part. Finally, We would specially like to thank **Mohsen Bayani Khaknejad** for introducing us to this Thesis work and constantly motivating us and providing valuable technical inputs throughout all stages of the project work.

Göteborg 2019-10-20

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Nitish Malangi Sehsanna

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1 Introduction

This report is a Master's Thesis project that is a combined effort of two students in the Master's Programme at Chalmers University of Technology for the Solidity team at Volvo Cars Corporation (VCC).
The following chapter is an introduction to the project describing its background, problem formulation, objectives and scope of work and limitations.

1.1 Background

Solidity, an important attribute within **Craftsmanship and Durability Centre** at **Volvo Car's Research** & **Development**, is responsible for Squeak & Rattle (S&R) noises and Solid feeling in the car. Squeak & Rattle (S&R) are non-stationary noises that occur when adjacent parts come into contact, either impacting or sliding. One of the team's traits is to establish trustworthy methods for attribute verification in the early phases by running simulations, instead of physical, complete vehicle - testing, to meet shorter lead time.
Increasing demand drives the development of electrified and thus quieter cars. So, these S&R noises become more obvious to perception and further draws the attention of passengers. A car with annoying in-cabin S&R noises is considered to be quality deficient.
The body torsional stiffness contributes to S&R. With higher stiffness normally follows lesser S&R noises. The Solidity team works for improving the body torsional stiffness and thereby preventing S&R aging that follows from body deformation and thus the interior attachments are more seldom loose pretension, making the car noiseless for a longer period.

1.2 Problem Formulation

In previous years, the body torsional stiffness was calculated in a test-rig configuration using a Body-In-Gray (BIG) structure. The outcome of such tests correlated very well with CAE simulations. Hence, it was decided that test-rig experiments were unnecessary and the simulations were enough for evaluating the body torsional stiffness.
This computer simulation way of calculating the body torsional stiffness uses a Body-In-Grey (BIG) model, that is established at the initial project phase to check for the optimum body torsional stiffness.

Later, a new test-rig experiment method (current method) was introduced, in which a full-scale car model is used to evaluate the body torsional stiffness. This method was introduced mainly to measure and compare the body torsional stiffness of competitors' cars.
As this method did not give results that correlated equally well with the CAE simulation, a task was formulated to investigate the possibility to obtain a better correlation of results when the simulation is done with a complete model (i.e chassis and trunk-lid included on a BIG model).

1.3 Thesis Goals and Objectives

The main aim of this thesis is to find out ways for a better correlation of results between the test-rig experiment and the simulation for the body torsional stiffness. Within this main objective statement, the following objectives are identified:

- Investigate the impact of including the chassis in the BIG model simulation.
- Investigate the variation in results for different loading and boundary conditions in the BIG model simulation.
- For the purpose of correlation, a new simulation method that replicates the test-rig experiment is to be performed using **MD ADAMS** in evaluating the body torsional stiffness.
- Perform the test-rig experiment on the 4-poster rig to evaluate the body torsional stiffness.
- Compare the results of the simulations with that of the test-rig and draw up correlation conclusions.
- As a second part of the thesis, perform a design optimization study on the body torsional stiffness simulation. The objective is to achieve a reduction in weight of the BIG structure with the body torsional stiffness unchanged.

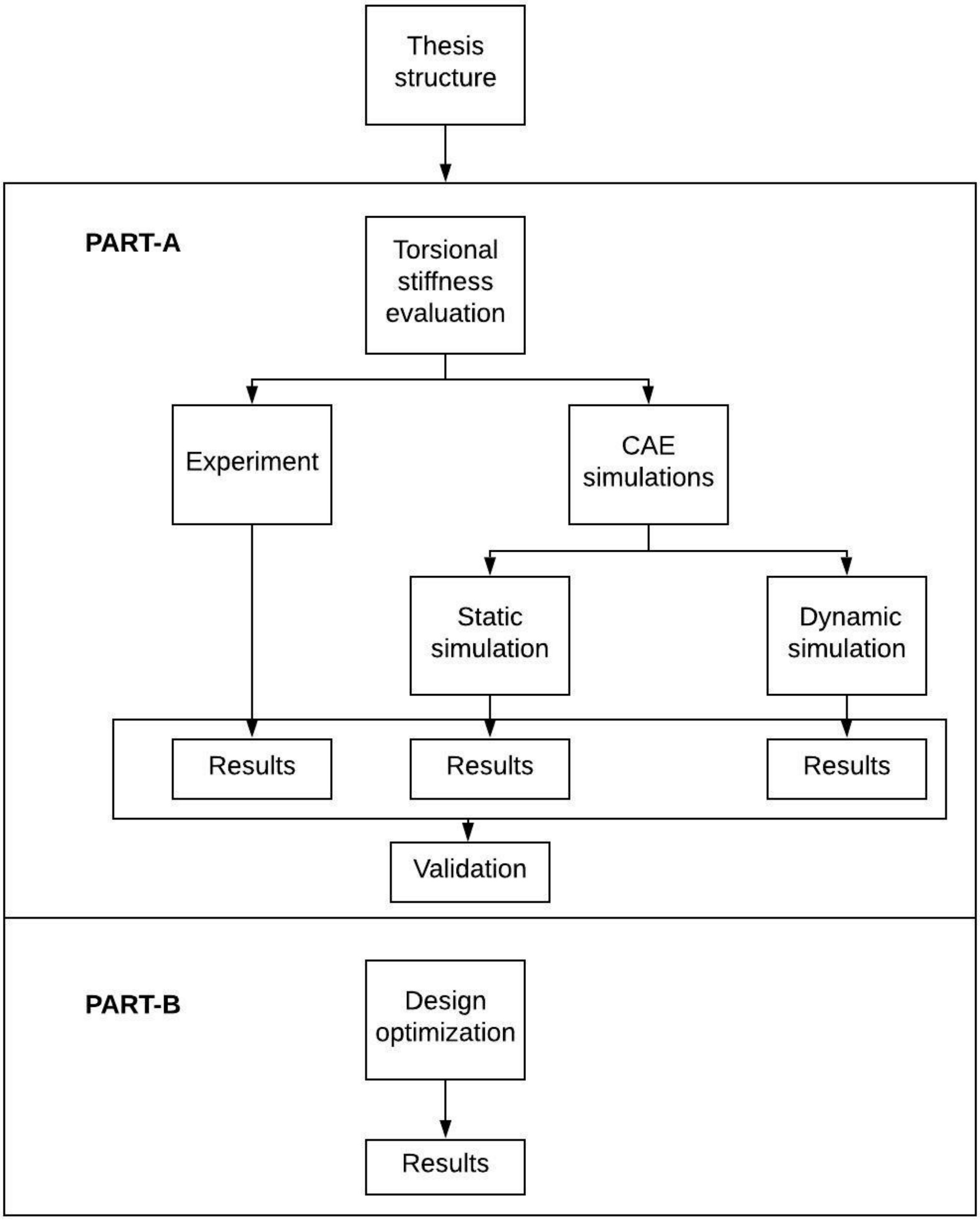
1.4 Limitations and Assumptions of the Thesis Work

The master thesis is approximately 20 weeks of work. The main limitation of the thesis is the available time frame. Other limitations identified are as follow:

- No FE-models are built, but provided by VCC and only the boundary conditions and loading cases are changed.
- The CAE tools used are industry specific and not applicable in all areas.
- Only one or two specific model's results are showed in this report but the number of models investigated/simulated is higher.
- For the reasons of confidentiality, the model names will not be specified but mentioned as MODEL-A, MODEL-B and so on.

1.5 Thesis Overview

The flowchart below shows the structure of the thesis work.



2
Theory

In this chapter the theory used in this Master Thesis is presented.

2.1 Body Torsional Stiffness

Body Torsional Stiffness is the resistance to twist that the car body has when a torsional load or a twisting load is applied to the BIG. At Volvo Cars Corporation, this body torsional stiffness is calculated as 2 values - KT_S and KT_G. The brief explanation is as follows.

2.1.1 KTS

KT_S refers to the Torsional Stiffness at the Support location. It is the ratio of the torque applied to the BIG structure to the support-adjusted twist angle between the front and rear support locations. Support location deflections are measured at the attachment points, and the deflection curves are “support-adjusted” such that the measured twist at the rear support location is zero. Support-adjusted torsional stiffness is used to study the stiffness of the entire structure including local compliance at the support locations which can significantly decrease full-vehicle stiffness. The support-adjustment also eliminates the effects of free-play in the supports and hem joints. The displacement measurements are taken at the front strut towers.

2.1.2 KTG

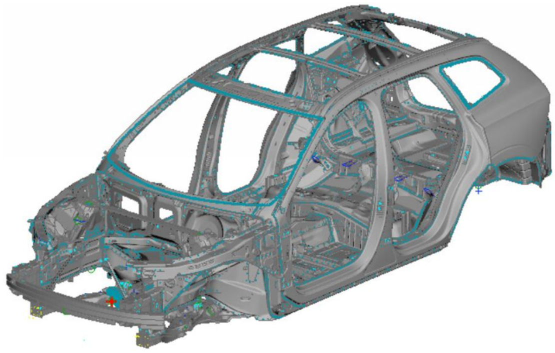
KT_G refers to the Torsional Stiffness at Global level. It is the ratio of the torque applied to the BIG structure to the globally adjusted twist angle between the front and rear support locations. The support location rotations are interpolated based on inner rail rotations measured fore and aft of the supports such that the twist at the rear interpolated support location is zero. Global torsional stiffness is used to study the stiffness of the entire structure excluding local compliance at the support locations. The measurements are taken at the front and rear side members.

2.2 Body-In-Gray (BIG)

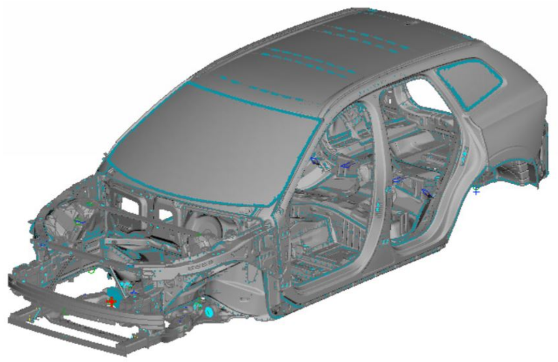
In the automotive world, a Body-In-White (BIW) structure is a common term. This refers to the stage in automobile manufacturing in which the car body sheet metal (including doors, hoods, bumpers and trunks) has been assembled but without the other major components (Engine, Chassis, interior and exterior trims and trim parts) [1]. Similar to the BIW, the Body-In-Gray (herein referred to as 'BIG') structure is a combination of BIW with windshield (Glass), bolted components and rally parts.

2.3 Four Poster Test rig

The four poster rig is used to carry out the body torsional stiffness experiment. It consists of four position controlled hydraulic rams with the 'load plates' mounted at the ends. The rig is controlled from a control room and the car is positioned freely on the load plates. The figure 2.2 shows the test rig and figure 2.3 shows the vehicle positioned on the rig.



(a) BIW Structure



(b) BIG Structure

Figure 2.1: Car body structures

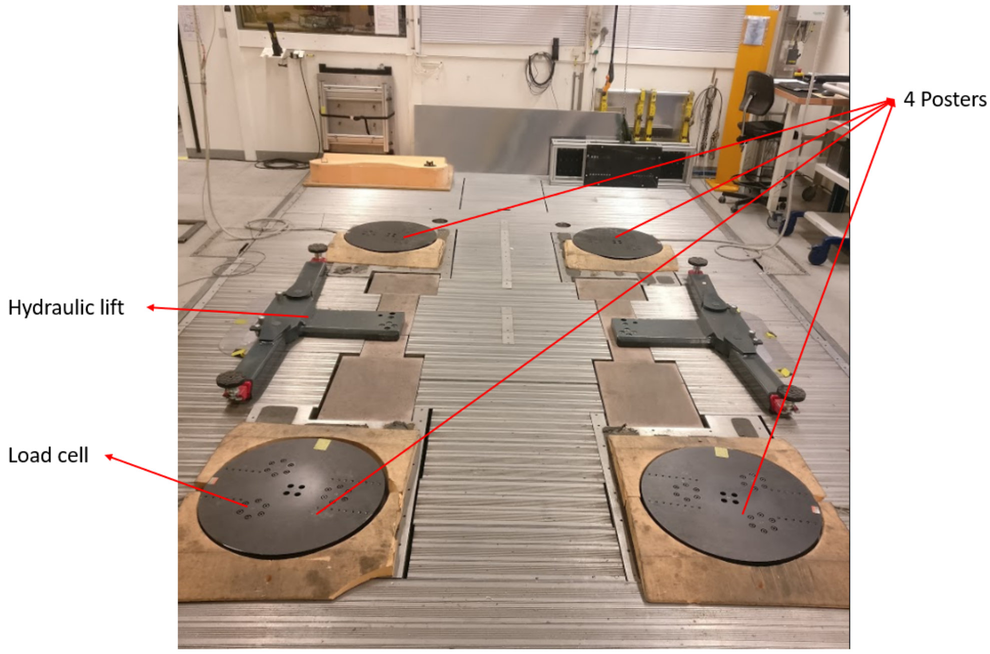


Figure 2.2: Four poster test rig

Figure 2.3: Vehicle positioned on the test rig

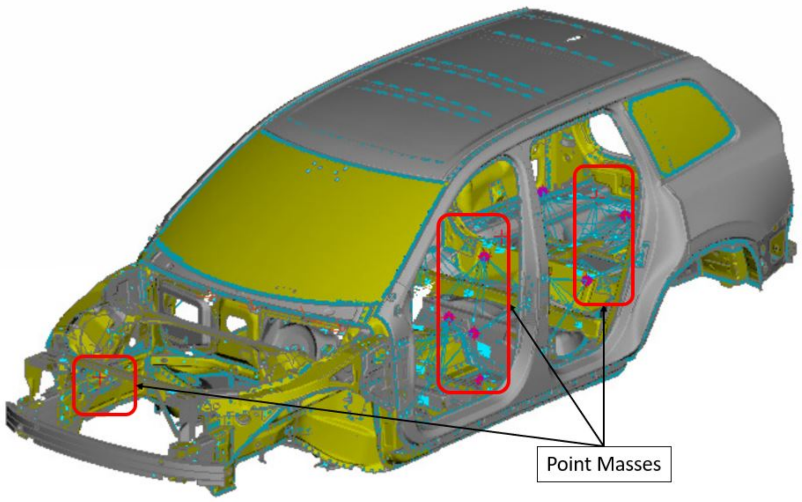


Figure 2.4: Trimmed model used in ADAMS

2.4 Trimmed Model in ADAMS

In ADAMS simulation, a trimmed model of the BIG is used. A trimmed model refers to the same Nastran model of the BIG with added point masses for the doors, tailgate, seats, engine blocks and other parts related to these.
The figure 2.4 shows a reference image of the trimmed model used in ADAMS. In the figure, small pink points represent the point masses (highlighted inside the box).

3

Methodology: Part-A

3.1 Body Torsional Stiffness Evaluation

The body torsional stiffness for a car structure is calculated by two methods - 1) From data of a test-rig experiment and 2) data from computer simulations. The calculation procedure is explained in detail below.

3.1.1 Test-rig Experiment

The test-rig experiment involves a fully assembled car model and a 4-poster setup to evaluate the body torsional stiffness. The procedure is:

1. Two inclinometers are mounted in the front and the rear of the car to capture the angle of twist. In the front end, a fixture is built between the two shock towers and the inclinometer is mounted at the center of this fixture. In the rear, a fixture is attached on the rear sub-frame and the inclinometer is mounted at its center.
2. The car with mounted inclinometers is resting on the four load cells (4-posters) and raised 150 mm above bottom position. The readings are noted on all four load cells and the inclinometers.

(a) Front: Between the strut towers

(b) Rear: Between sub frame attachement

Figure 3.1: Location of the inclinometers

Figure 3.2: Data readings on the four poster control (for ref)

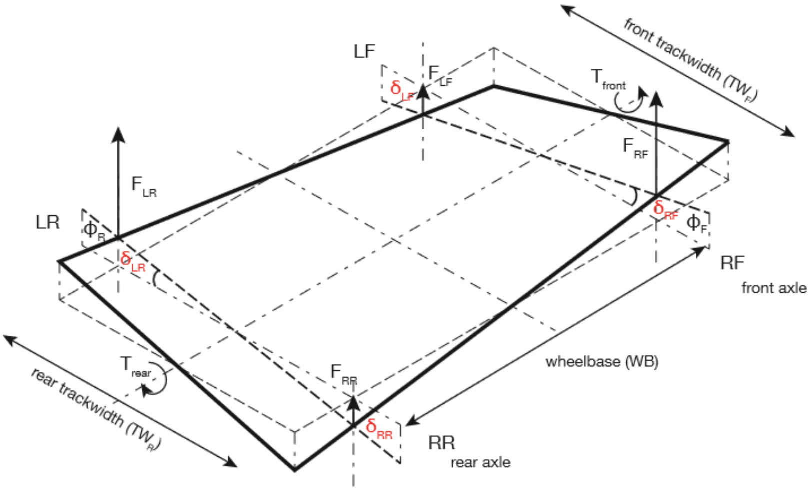


Figure 3.3: Simplified FBD of the car on KSK2-rig used for calculations

3. The load-cell with the lowest force reading and the one diagonally opposite are lowered and the other two wheels are raised until one of the wheels hangs free on the air and the car rests on three wheels. This defines the maximum displacement. This process is done in steps of 5 mm with a ramp duration of 20 seconds and by trial and error, the exact displacement when the wheel is just off the load cell is measured.
4. All the load cells are brought back to the initial position (where the car is 150 mm up). Now that the maximum displacement is known, the experiment is carried out with 5 mm ramps and the load readings and the inclinometer readings are measured after every such ramp.
5. The experiment is repeated three times to ensure repeatably.

3.1.1.1 Calculation of Torsional Stiffness

After the readings are recorded from the test rig, the values are evaluated to obtain the body torsional stiffness.
Ideally, the moment about the center of gravity should be considered. Since the mass distribution of the car is almost symmetric over the center line, it is fair to

assume that the center of mass, when viewed along the length of the car, lies on the center line. Thus, the moments are calculated about the center-line of the car.

$$T_F = \frac{1}{2} cos\phi_F \cdot TW_F \cdot (F_{RF} - F_{LF}) \tag{3.1}$$

$$T_R = \frac{1}{2} cos\phi_R \cdot TW_R \cdot (F_{LR} - F_{RR}) \tag{3.2}$$

Where,

$$\phi_F << 1 \rightarrow cos\phi_F \approx 1 \tag{3.3}$$

$$\phi_R << 1 \rightarrow cos\phi_R \approx 1 \tag{3.4}$$

The angle of twist is calculated by adding the measured values of the twist angle in the front and in the rear.

$$\Delta\phi = \phi_F + \phi_R \tag{3.5}$$

Ideally the the evaluation of test data for all the moments acting on the body should be equal. But the test impression leads to a difference in front and rear torques. For the evaluation of twist stiffness this is averaged out as follows.

$$K_F = \frac{T_F}{\Delta\phi} \tag{3.6}$$

$$K_R = \frac{T_R}{\Delta\phi} \tag{3.7}$$

$$K_{total} = \frac{K_F + K_R}{2} \tag{3.8}$$

3.1.2 CAE Simulations

Body torsional stiffness has historically usually been calculated in simulations by a conventional way, i.e., using a FE-model of the BIG and performing a static loading with appropriate loading and boundary conditions. This is the CAE method. This method did not correlate well with the new test-rig experimental results. Hence, we opted for a new method using the tool **MD ADAMS** to simulate the test-rig conditions. Both the methods are explained in detail below.

3.1.2.1 Static Simulation

This method involves the use of the tools ANSA/META with MSC Nastran (solver) to run a static body torsional stiffness simulation on the BIG (Body-In-Gray) model. The procedure is as below.

1. The CAE method is used to compute body torsional stiffness as two different values - KTS and KTG. For the Solidity team, the focus is on KTG.
2. The following table 3.1 and figure 3.4 shows the nodes used for loading and boundary conditions.

Part Name	Part Location	Node Number
Front Bumper Beam	Front Middle	1000
Front Strut Tower	Left	1055
Front Strut Tower	Right	2055
Rear Strut Tower	Left	3055
Rear Strut Tower	Right	4055
Front Side-member	Left	1355
Front Side-member	Right	2355
Rear Side-member	Left	3355
Rear Side-member	Right	4355

Table 3.1: Standard numbering and the PID's used in the calculation

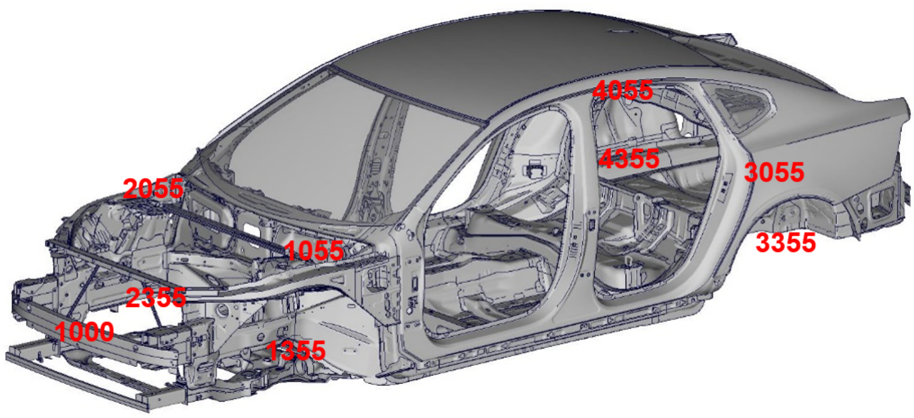


Figure 3.4: Location of the nodes used for the analysis

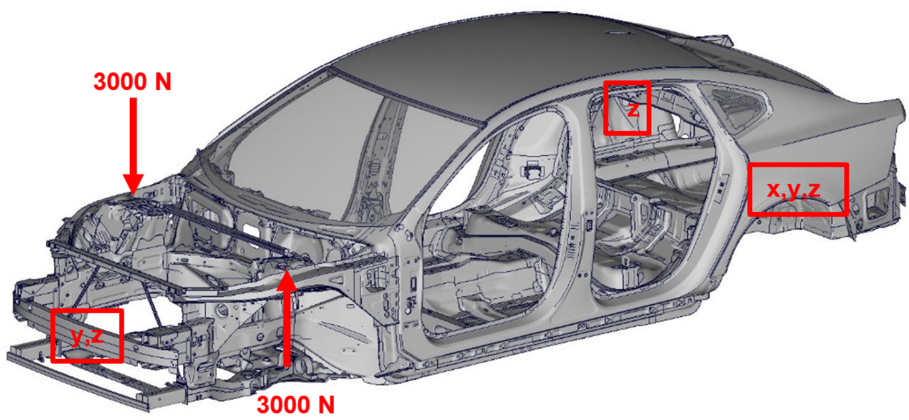


Figure 3.5: Loads and boundary conditions used for the analysis

3. The figure 3.5 shows the loading and boundary conditions used for the body torsional stiffness simulation. It can be seen that the loads are prescribed at

the front shock absorber attachments (Strut Tower nodes - 1055 & 2055). The translational constraints are indicated by x,y,z. Here, the front bumper beam (node 1000) is constrained in y and z translation, the rear-left side-member (node 3355) is constrained in all translation directions and the rear-right side-member (node 4355) is constrained in z-translation.

4. The job is submitted for analysis in *NASTRAN* and post the analysis, the z-displacements (dz) for the nodes 1355, 2355, 3355 and 4355 are noted down from the *.pch* file.

3.1.2.1.1 Calculation of KTS and KTG

The moment of the applied forces is calculated at the center line along the length of the car. The formula is as below.

$$M = (f_1 + f_2) \times 0.5\ d = 2f \times 0.5\ d \tag{3.9}$$

where, M is the applied moment stemming from the applied twisting forces $f_1 = f_2 = f$.
d is the distance between the points where the forces are applied. i.e at points 1055 and 2055.
The torsional stiffness is calculated as

$$KTG = M/\alpha \tag{3.10}$$

$$\tag{3.11}$$

where, KTG is the Body Torsional Stiffness and
α is the Angle of Twist which is calculated as follows.

$$\alpha_1 \ = \ tan^{-1}\Big[(dz_1 + dz_2)/d_2\Big] \tag{3.12}$$

$$\alpha_2 \ = \ tan^{-1}\Big[(dz_3 + dz_4)/d_3\Big] \tag{3.13}$$

$$\alpha \ = \ (\alpha_1 - \alpha_2) \tag{3.14}$$

where, dz_1, dz_2, dz_3 & dz_4 are the z-displacements of nodes 1355, 2355, 3355 and 4355 respectively.
The distance between nodes 1355 and 2355 and distance between nodes 3355 and 4355 are d_2 & d_3 respectively.

3.1.2.2 Dynamic Simulation

New methods were investigated to evaluate body torsional stiffness similar to that of the test-rig method. One such method is the dynamic simulation performed using **MD ADAMS**.
Here, the four-poster rig is replicated and a simulation is performed with the car modelled as a flexible body with rigid components attached. The flexible body is the trimmed model (see section 2.4 for definition) of the BIG model (.nas file). The rigid components include the bushings, suspensions, dampers and sub-frames.
The procedure to simulate and evaluate the body torsional stiffness is explained below.

- The four-poster rig conditions is imported into ADAMS using a script developed by Volvo called *"Prepare sim script for 4-post"*. The loaded assembly along with the four-poster is shown in figure 3.6.
- The full-vehicle analysis is performed in ADAMS using prepared scripts as *Simulate → Full-Vehicle Analysis → ACME Four post*.
- A dialog-box pops up to setup the analysis. It is as shown in the figure 3.7.

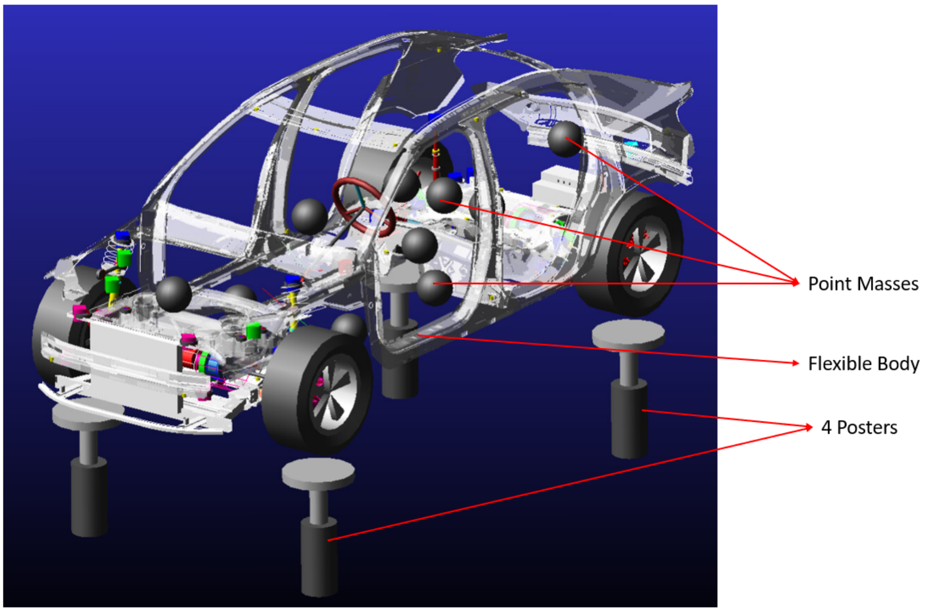


Figure 3.6: Assembly in ADAMS

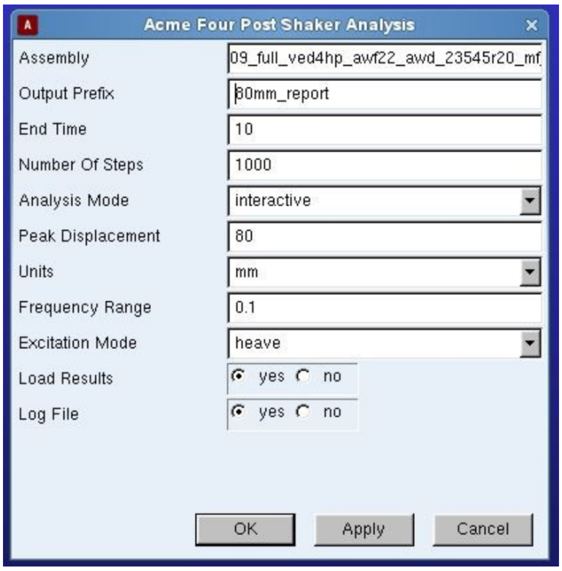


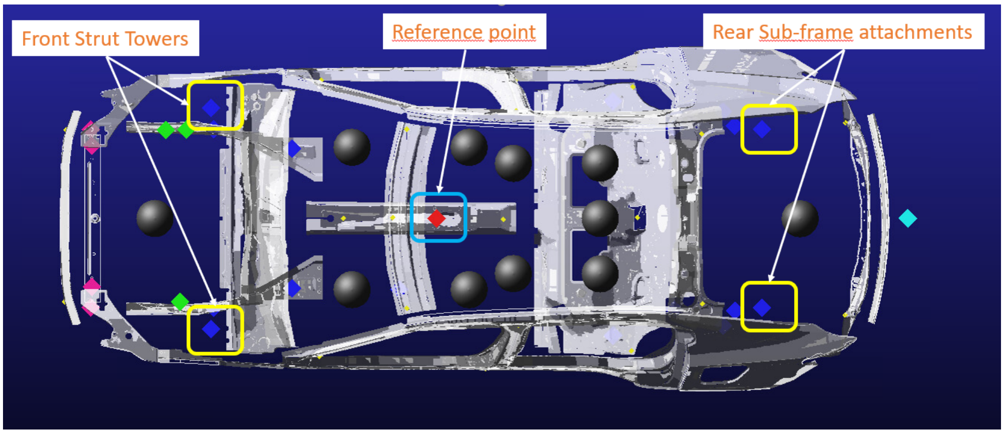
Figure 3.7: Data for simulation setup in ADAMS

- The required data is entered as shown in the figure 3.7.The simulation adjusts the position of the car model on the four poster.
- The *"Prepare sim script for 4-post"* runs the simulation. An input file specifies the torsional stiffness by using various parameters. Displacements on the particular nodes, forces on the wheels, torque from the force and torsional angle are calculated in the simulation.
- Once the simulation is completed, an n-built post-processor can be used to visualize the results.

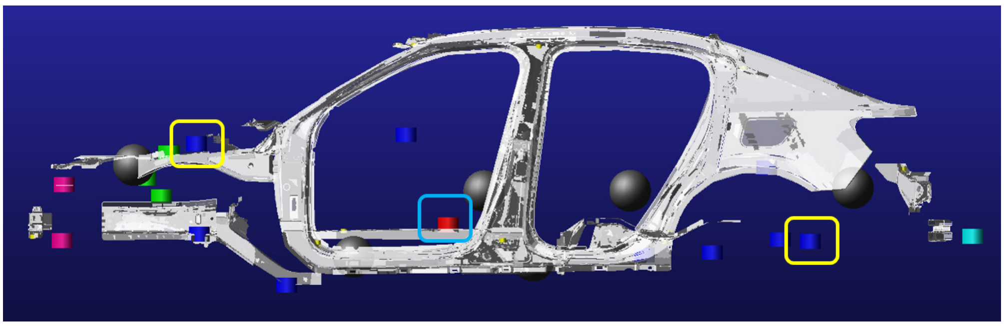
3.1.2.2.1 Calculation of Body Torsional Stiffness

The input file for simulation is used for computing different parameters that can be visualized in the post-processor. This script does not take into account the point masses, as the flexible body and its connected sub-components are the ones involved in this simulation. The steps involved in the calculations are:

- Similar to the static simulation, the torsional stiffness is defined from the motion of 4 nodes (see figure 3.8). Four markers (blue) are placed on these nodes and another marker (red) is placed on the tunnel as a reference point.
- The four markers are used for calculating the displacement relative to the reference point.



(a) Plan view



(b) Side view

Figure 3.8: Calculation reference in ADAMS

- The front twist angle is calculated by taking the difference of the displacements in the front nodes and divide that by the distance between the nodes and vice-versa for the rear twist angle.
- The force on each wheel is calculated from the four poster pads by the force acting on them.
- The front torque is calculated as the difference between the forces on the front wheels and divided by the distance between the front posters. Its vice-versa for the rear torque as well.
- The front torsional stiffness is calculated as the front torque divided by the front twist angle and the rear torsional stiffness is calculated as the rear torque divided by the rear twist angle. The total torsional stiffness is the mean of the front and rear torsional stiffness.

3.2 Validation of the Results

After calculating the body torsional stiffness through the various methods (mentioned above), a comparison is made between the test-rig method and the simulations. This is to validate the simulation method that correlates closer to the test-rig method. This is important to understand the impact of the chassis in the FE simulations.

4

Results and Discussion: Part-A

We started with the simulations on the **Model-A** and were able to simulate the BIG model and the full car FE-Model (BIG with *Chassis* and trunk lid). As the test car is unavailable for carrying out the test-rig experiment, we had to study another model **'Model-B'** for the validation purpose. Hence, in this chapter we present the results of the methods discussed in the previous chapter for the two models.

4.1 Model-A

The results of the simulations and the experiment for the Model-A are as follows.

4.1.1 CAE - Static Simulation

For this car model, both a BIG model and a full car FE model are used for the static simulation, along with different loading and boundary conditions.

4.1.1.1 BIG Model

For the BIG, only the standard loading and boundary conditions are used (see figure 4.1).

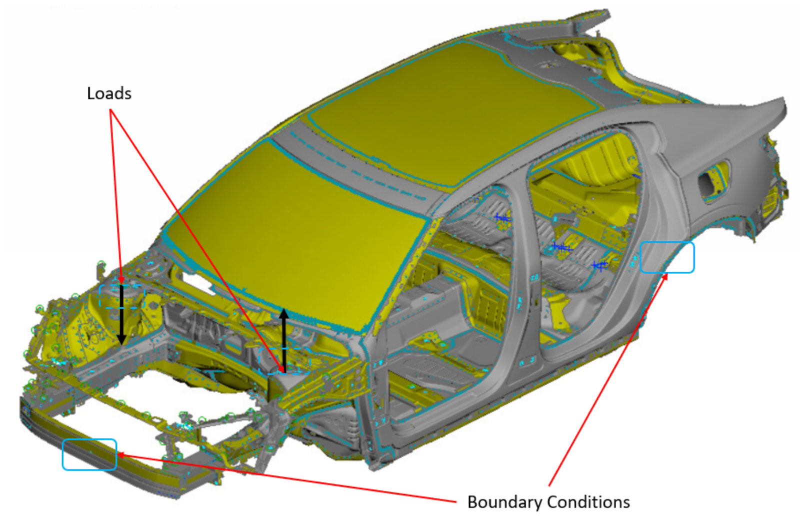


Figure 4.1: Model-A BIG simulation setup

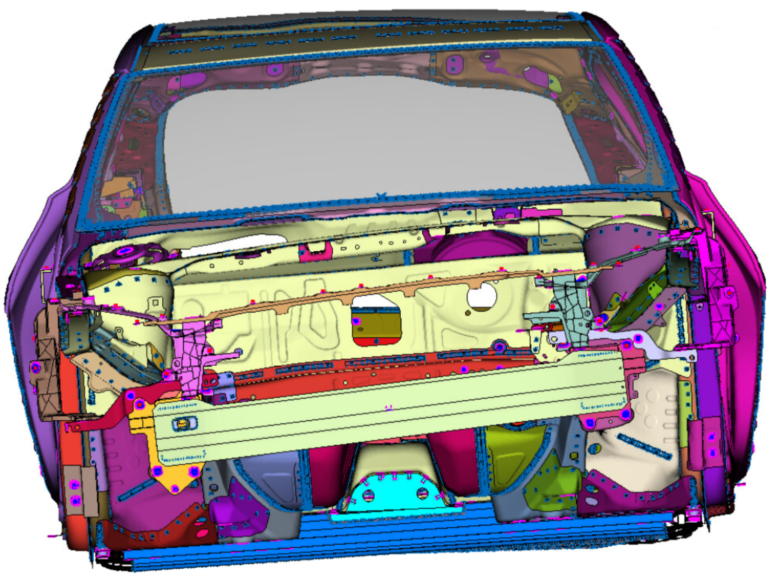


Figure 4.2: Post-processed result of the Model-A BIG from Meta - Rear view

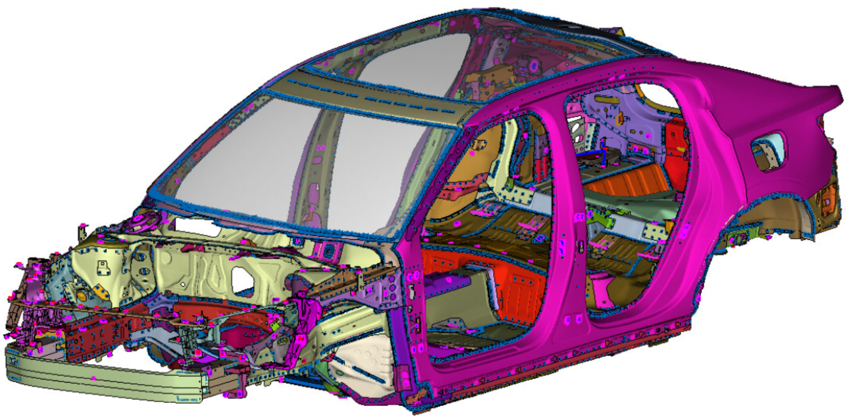


Figure 4.3: Post-processed result of the Model-A BIG from Meta - Iso view

The simulation results illustrated in figures 4.2 and 4.3 show the front view and iso view of the Model-A (BIG) at maximum torsional angle. This is the angle at which the maximum stiffness is obtained. A scaling factor of 10 was used to visualize the twist in the body.
At this angle and from the equations 3.10 and 3.14, the KTG value for this model is calculated to be:
KTG = 29.84 kNm/deg

4.1.1.2 Full car FE-model

4.1.1.2.1 Standard Loading and Boundary Conditions
The full car FE-model is first simulated with standard loading and boundary conditions as shown in the figure 4.4. The results are shown in figures 4.5 and 4.6.

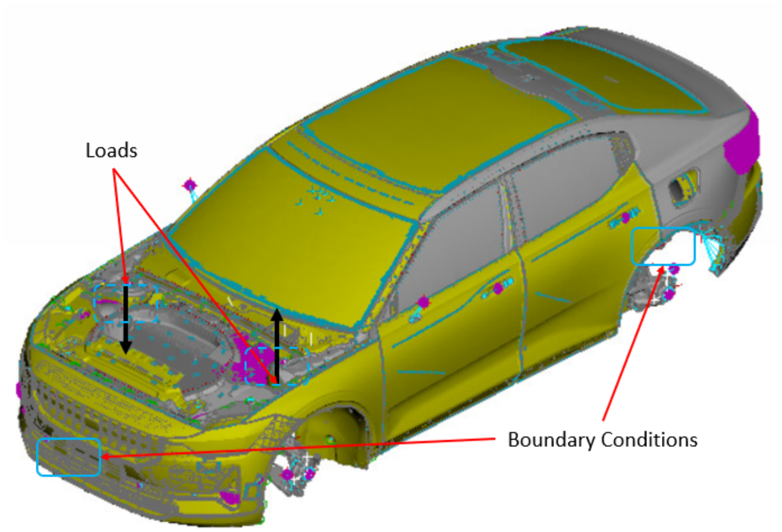


Figure 4.4: Mode-A full car simulation setup

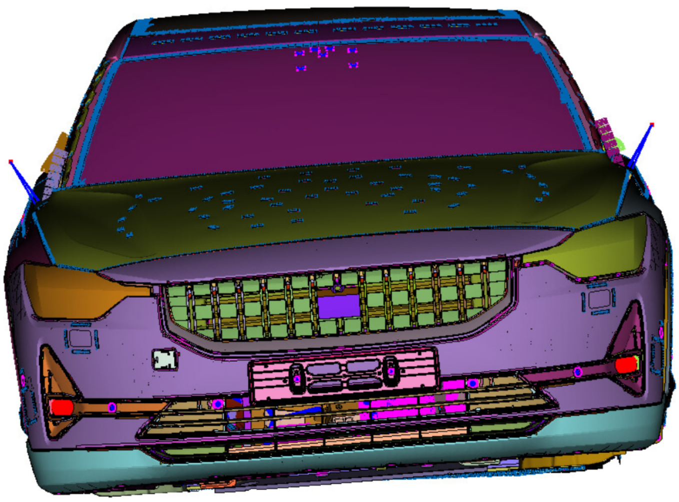


Figure 4.5: Post-processed result of the full car simulation from Meta - Front view

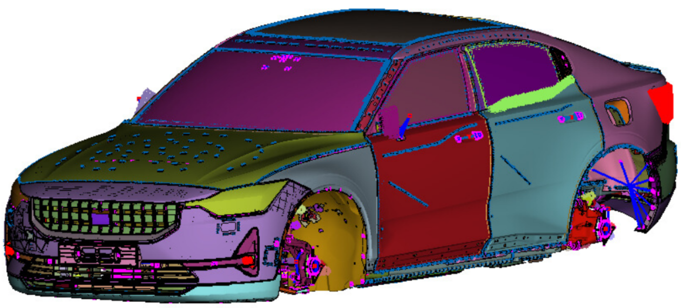


Figure 4.6: Post-processed result of the full car simulation from Meta - Iso view

The figures 4.5 and 4.6 show the front view and iso view of the Model-A (full-car) at the maximum torsional angle (similar to BIG). A scaling factor of 10 was used to visualize the twist in the body.
At this angle and from the equations 3.10 and 3.14, the KTG value for this model is calculated to be:
KTG = 37.26 KNm/deg

4.1.1.2.2 Full car - Wheel Loads

The full car FE-model is simulated with the same boundary conditions but the loads are applied on the knuckles of the front wheel attachment points, as shown in the figure 4.7.
The results are shown in the figure 4.8. It shows the front view of the Model-A (Full-car with wheel load) before and after the simulation at maximum torsional angle. A scaling factor of 50 was used to visualize the twist in the body, since this load case seem to result in a smaller twist.

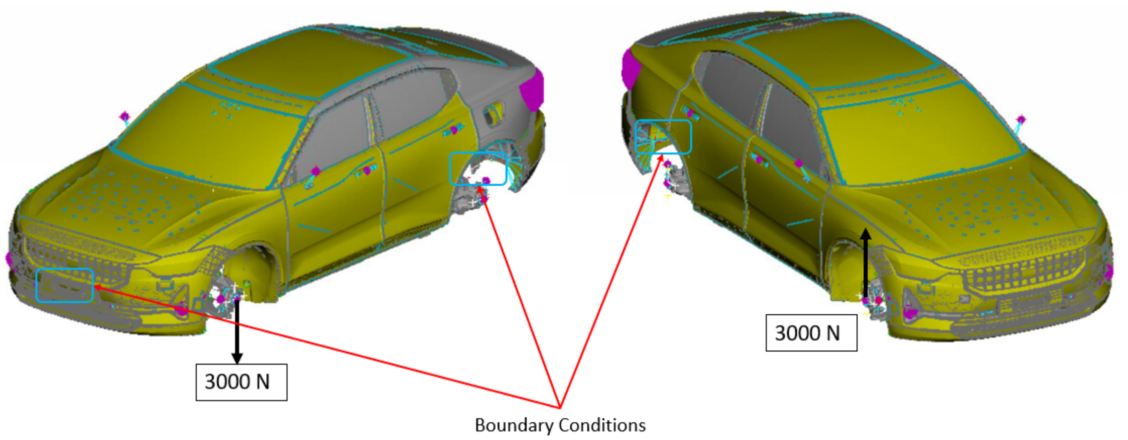


Figure 4.7: Mode-A full car wheel load simulation setup

Figure 4.8: Post-processed result for the wheel load from Meta - Front view

At this angle and from the equations 3.10 and 3.14, the KTG value for this model is calculated to be:
KTG = 39.83 KNm/deg

4.1.2 CAE - Dynamic Simulation

Figure 4.9 shows the graphical representation of the total body torsional stiffness of the Model A. The simulation is carried out in such a way that it yields maximum twist at 7 seconds. The KTG value at the time step is:
KTG = 37.9 kNm/deg.
The simulated images of the model-A from ADAMS are shown in figures 4.10 and 4.11.

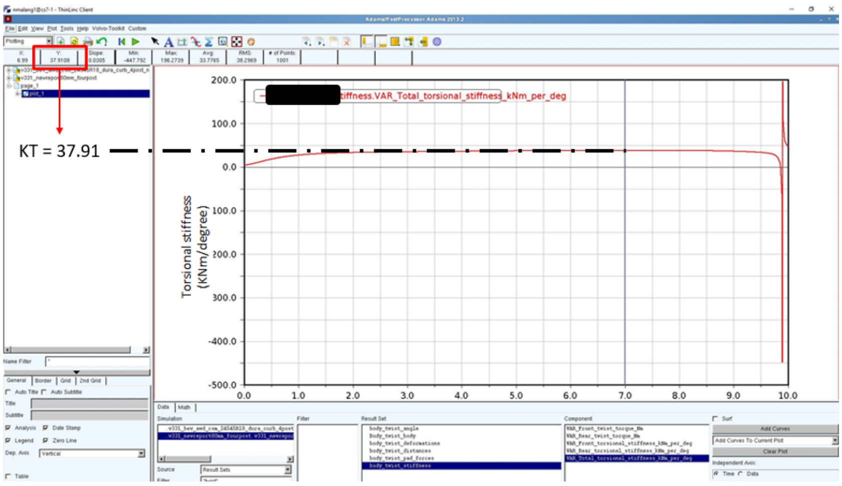


Figure 4.9: Multi-body dynamic simulation result- Model A

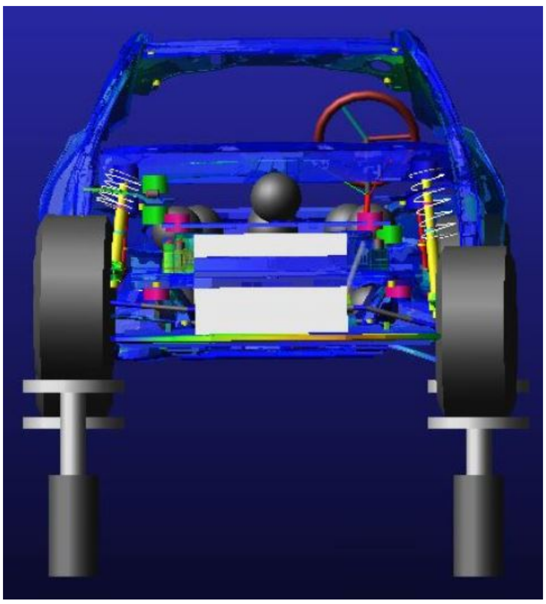


Figure 4.10: Simulation result- Model A front view

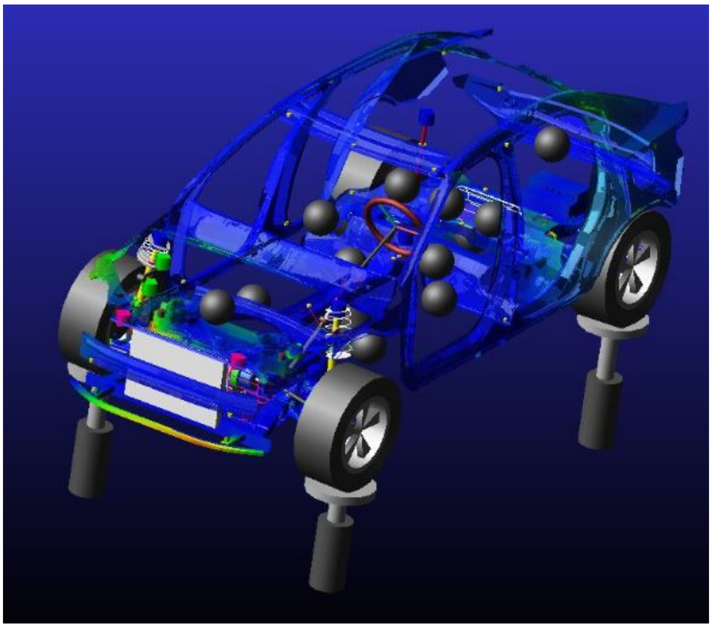


Figure 4.11: Simulation result - Model A iso view

4.1.3 Test-Rig Experiment

As mentioned in the start of the Results section, the test car is unavailable for performing the test on the four poster rig and so no results are obtained.

4.2 Model-B

The results of the simulations and the experiment for the Model-B are as follows.

4.2.1 CAE - Static Simulation

For this car model, only a BIG model is used for the static simulation with standard loading and boundary conditions.

4.2.1.1 BIG Model

4.2.1.1.1 Standard Loading and Boundary Conditions

For this model, only the standard loading and boundary conditions is used, as shown in the figure 4.12. The results are shown in figures 4.13 and 4.14.

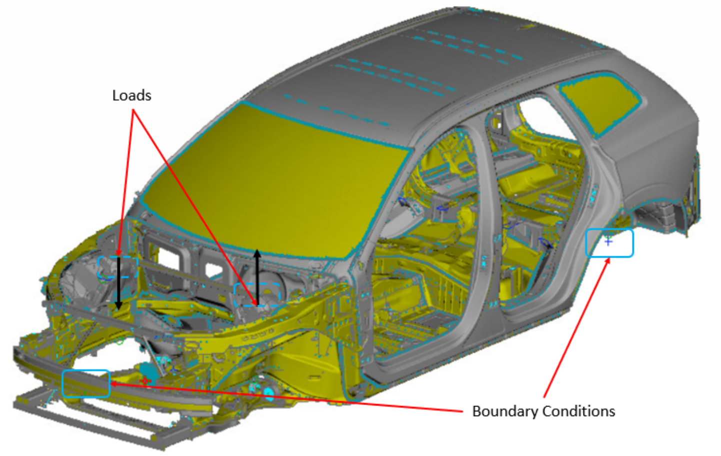


Figure 4.12: Mode-B BIG simulation setup

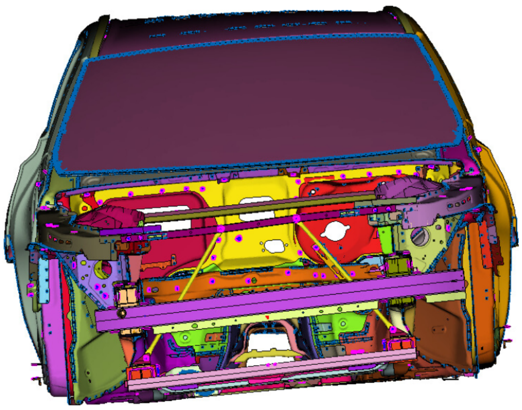


Figure 4.13: Post-processed result of the Model-B BIG from Meta - Front view

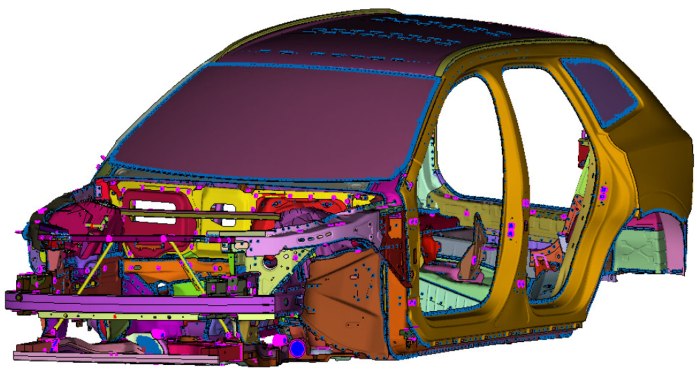


Figure 4.14: Post-processed result of the Model-B BIG from Meta - Iso view

Figures 4.13 and 4.14 show the front view and iso view of the Model-B (BIG) at maximum torsional angle. A scaling factor of 10 was used to visualize the twist in the body.

At this angle and from the equations 3.10 and 3.14, the KTG value for this model is calculated to be:
KTG = 25.9 KNm/deg

4.2.2 CAE - Dynamic Simulation

Figure 4.15 shows the graphical representation of the total body torsional stiffness of the Model A. The simulation is carried out in such a way that it yields maximum twist at 7 seconds. The KTG value at the time step is:
KTG = 27.1 kNm/deg
The simulated image of the Model-B from ADAMS is shown in the figure 4.16.

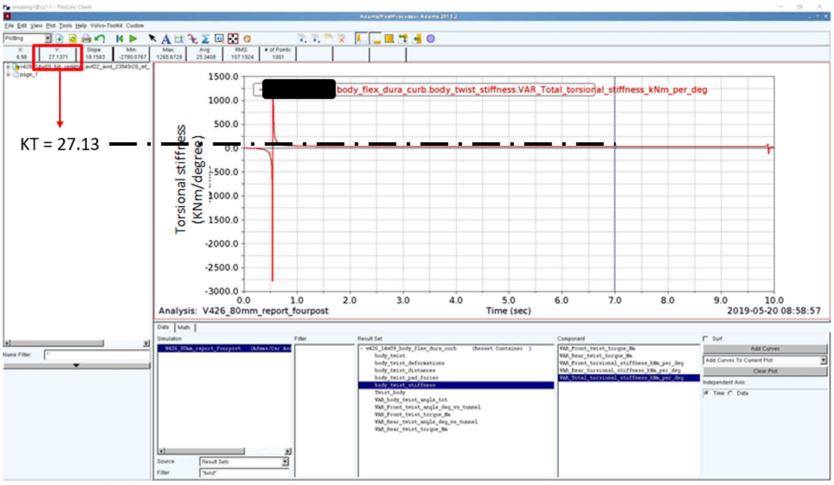


Figure 4.15: Multi body dynamic simulation result- Model B

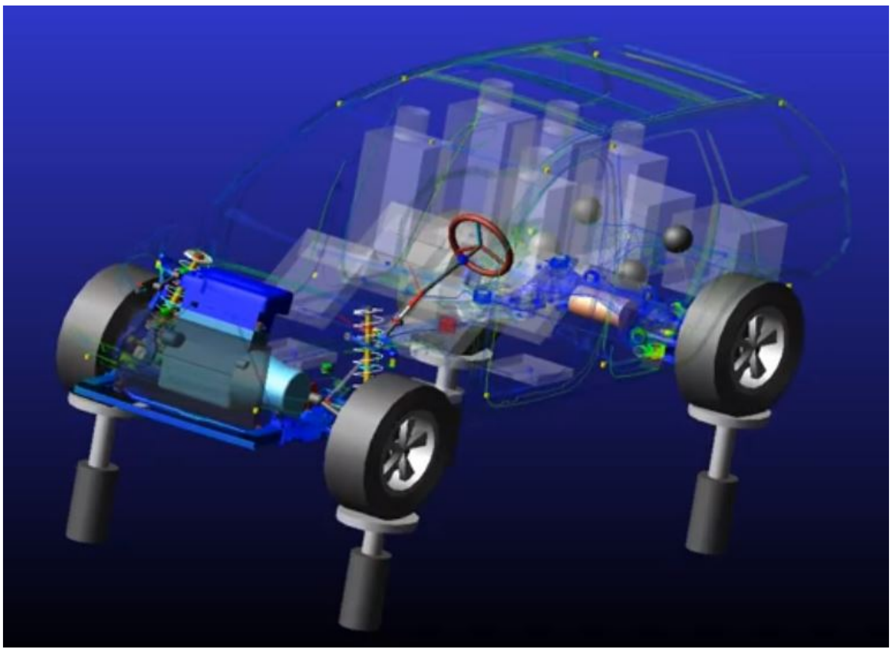


Figure 4.16: Simulation result- Model B Iso View

4.2.3 Test-Rig Experiment

Figure 4.17 shows the maximum twist of the Model-B during the experiment. At this step, the results in the figure 4.18 are obtained.
This is the output of an excel macro which calculates the total body torsional stiffness. The body torsional stiffness obatined for Model-B is ***34.4 kNm/deg***. This is the highest value of the torsional stiffness among the three methods used.

Figure 4.17: Maximum twist in the Model B for the test-rig experiment

1	Calculation of torsional stiffness										
2											
3	Model-B		Front effective trackwidth [m]		Rear effective tackwidth [m]		Wheelbase [m]	LF	RF		
4			1,655	[m]	1,66		2,865	4,649	4,715	955	
5								LR	RR		
6			Conversion					3,674	3,906	773	
7			9,81	kN-->kg				total weight	1727	[kg]	
8											
9			LF		RF						
10			kN	kg	kN	kg					
11		1	7,856	800,8	1,656	168,8					
12		2	7,823	797,5	1,662	169,4					
13		3	7,789	794,0	1,663	169,5					
14			LR		RR						Output Result
15			kN	kg	kN	kg					
16		1	0,318	32,4	7,081	721,8					
17		2	0,345	35,2	7,074	721,1					
18		3	0,350	35,6	7,096	723,3					
19											
20				Torque [kNm]		Total					
21				Front	Rear	[kg]	Torsion angle (degrees)		T_F [Nm/deg]	T_R [Nm/deg]	
22			1	5,185	-5,417	1724	0,153		33,89	-35,42	
23			2	5,153	-5,389	1723	0,154		33,57	-35,11	
24			3	5,124	-5,404	1722	0,154		33,30	-35,11	
25								Average	33,59	35,21	34,40 kNm/deg

Figure 4.18: Results of the test-rig experiment of Model B (using Excel)

4.3 Validation of the Results

S.No.	**Model**	**Body Torsional Stiffness (kN m/degree)**				
		CAE Simulations				**Test-rig Experiment**
		Nastran			**Adams**	
		BIG	**Full Car**			
			Standard	**Wheel Load**		
1	V331	29.84	37.26	39.83	37.9	NA
2	V426	25.9	NA	NA	27.1	34

Table 4.1: Results compilation

Table 4.1 shows the results for the body torsional stiffness evaluation through both the simulations and the test-rig experiment.
It can be seen from the table that there is no result (yet) for the test-rig experiment of the Model-A. Also, the results of the full car FE simulation performed in **Nastran** is similar to that of the **Adams** simulation.
For Model-B, the **Adams** simulation result is somewhat closer to that of the test-rig experiment.

5

Methodolgy: Part-B

5.1 Design Optimization

Once the different methods to evaluate body torsional stiffness are validated, the design optimization of the BIG is performed to reduce the weight of the BIG structure. It was observed that the Model-A had a higher torsional stiffness value compared to the Model-B. This provided room for weight optimization of the Model-A. The optimization is carried out using the tools **Hypermesh** (Pre-Processor), **Optistruct** (Solver) and **Hyperview** (Post-Processor).
Figure 5.1 shows the flowchart for the optimization procedure.

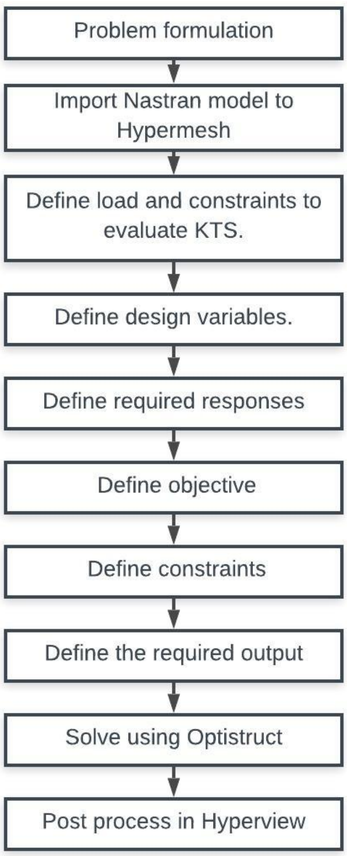


Figure 5.1: Design optimization procedure – Flowchart

The steps involved in this optimization procedure are explained briefly and they are as follows.

1. The optimization process starts with a problem definition. The problem definition here is to reduce the weight of the BIG structure of model-A with no change in the body torsional stiffness. The next step is to input the .nas file of the Model-A (with the geometry, material and element properties) into the Hypermesh pre-processor. Figure 5.2 shows the user interface of *Hypermesh* to setup the model for design optimization.
2. The next step is to setup the loads and boundary conditions for calculating the body torsional stiffness. As shown in figure 5.2, the forces are applied on the nodes 1055 (+3000 N) and 2055 (-3000 N) on the strut towers and the constraints are applied on the nodes 3055 (1,2,3 translation dof fixed), 4055 (1 translation dof fixed) and 1000 (2,3 translation dof fixed).
3. Further, the design variables are identified. In this case, the design variables represent all the shell element panels whose thickness are to be optimized. The model is carefully screened to pick only the shell elements. The upper and lower bounds of the optimization thickness values are defined as ± 20% of the initial thickness value. It is ensured that the lower bound for critical components does not go below 0.6mm, considering manufacturing limitations. After defining the design variables, the different responses are specified. These include the z displacements of the nodes 1055 and 2055 as well as the torsional stiffness value defined by an equation.
4. After defining the responses, the objective function is defined as the minimal mass and the constraint is set to fix the torsional stiffness. The required output files for post-processing are defined and the solverdeck is exported. Figure 5.3 shows the bulkcard file which is solved in the Optistruct solver.

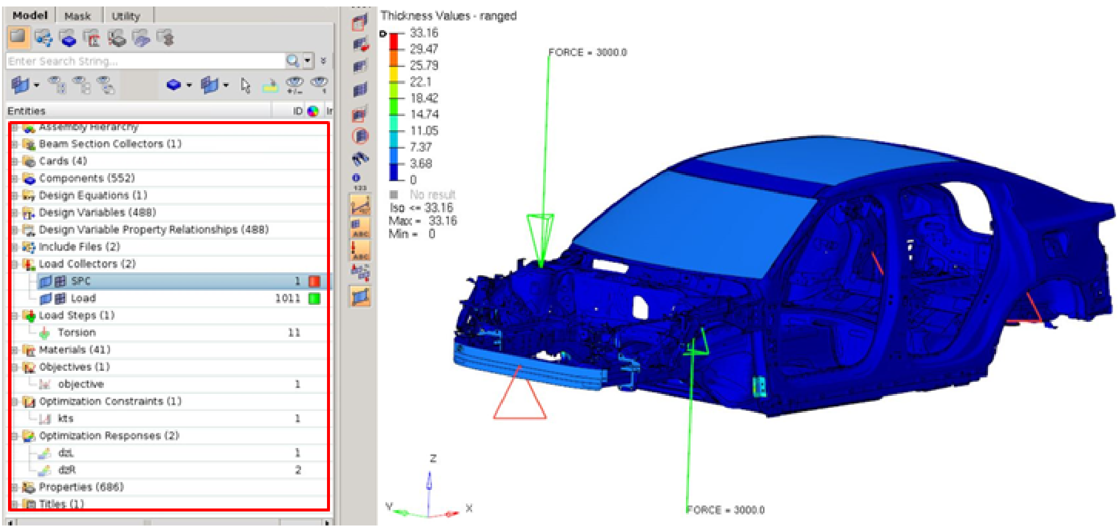


Figure 5.2: Design optimization procedure in Hypermesh

```
OUTPUT,H3DGAUGE,FL,ALL
OUTPUT,MSSENS,FL,ALL
$$------------------------------------------------------------------------$
$$                      Case Control Cards                                $
$$------------------------------------------------------------------------$
$ Objective function
DESOBJ(MIN)=4
$
$
$HMNAME LOADSTEP              11"Torsion"       1
$
SUBCASE       11
  LABEL Torsion
  SPC =         1
  LOAD =      1011
  DESSUB =        2
$ ------------------------------------------------------------------------$
BEGIN BULK
$ ------------------------------------------------------------------------$
INCLUDE '/vcc/cae/backup/solid/users/rmalang1/opti/V331_trial4/v331_18w15_v2_BI0_with_battery_frame.nas'
INCLUDE '/vcc/cae/backup/solid/users/rmalang1/opti/V331_trail5/v331_des_var_finalopt.fem'

$ ------------------------------------------------------------------------$
PARAM,CHECKEL,NO
CONTPRM CONTGAP YES
$
$ ------------------------------------------------------------------------$
$HMNAME OPTICONTROLS       1"optistruct_opticontrol"
$
$DOPTPRM DESMAX  20      MATINIT 0.3     MINDENS 0.0001  DISCRETE2.0
$
$ -------------------- DESIGN RESPONSE ------------------------------------$
DRESP1  1       dzL     DISP                            TZ          1055
DRESP1  2       dzR     DISP                            TZ          2055
DRESP2  3       KTS             1
+       DRESP1          1       2
DRESP1  4       mass    MASS
$DRESP2  5       KT0             5

DEQATN  1       KTS(dzL,dzR)=(3*1179.4*0.001)/((ATAN((dzL-dzR)/(1179.4))
+       *180.0/3.14159))

$DEQATN  3       fif(dzL,dzR)=(ATAN((dzL-dzR)/(1049.4))*180.0/3.14159)

$DEQATN  4       fir(dzLr,dzRr)=ATAN((dzLr-dzRr)/1057.5)*180.0/3.14159

$DEQATN  5       KT0(fif,fir)=((3*972.9*0.001)/(fif-fir))
$
$ -------------------- DESIGN CONSTRAINTS ---------------------------------$
$HMNAME OPTICONSTRAINTS        1kts
$                        >than   <than
DCONSTR         1       327.0
$DCONSTR         3       530.0
DCONADD         2       1
$ -------------------- LOAD COLLECTORS ------------------------------------$

$$  SPC Data
$$
SPC             1    3055  123    0.0
SPC             1    4055  3      0.0
SPC             1    1000  23     0.0
$$
$$  FORCE Data
$$
FORCE       1011    2055        01.0    0.0     0.0     -3000.0
FORCE       1011    1055        01.0    0.0     0.0     3000.0
$$
ENDDATA
```

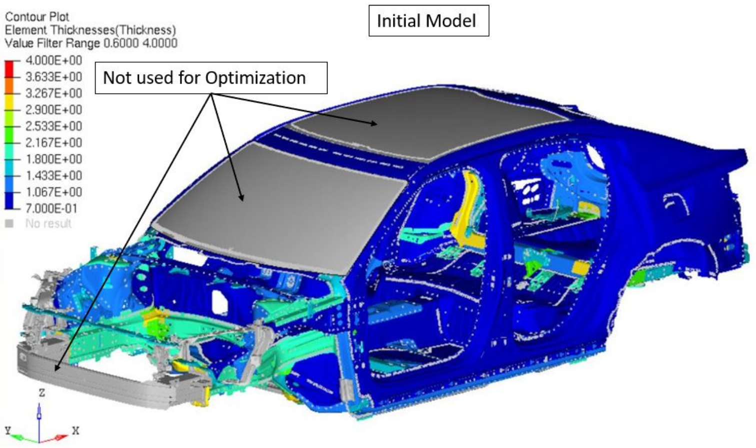
Figure 5.3: Bulk card input for optimization

5. The final step in the optimization is to retrieve data from the result file using the post-processor Hyperview and identify the change in thickness of the panels.

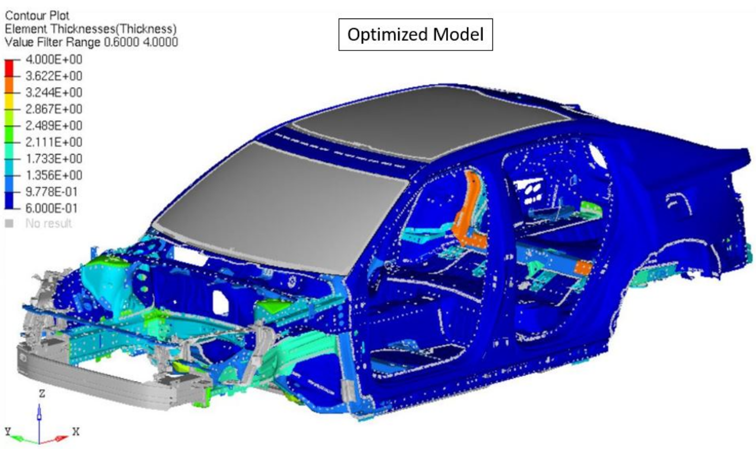
6
Results and Discussion: Part-B

6.1 Design Optimization

The results of the gauge (thickness) optimization is shown in figure 6.1. As seen in figure 6.1a, a few components are highlighted in grey. These parts are not considered for optimization. Figure 6.1b visualizes the results after the optimization. Here, the parts highlighted in red are the ones with increase in thickness and the blue ones represent the parts with decrease in thickness during the optimization. A filter is set to show only the panels with the thickness range of 0.6 mm to 4.0 mm.



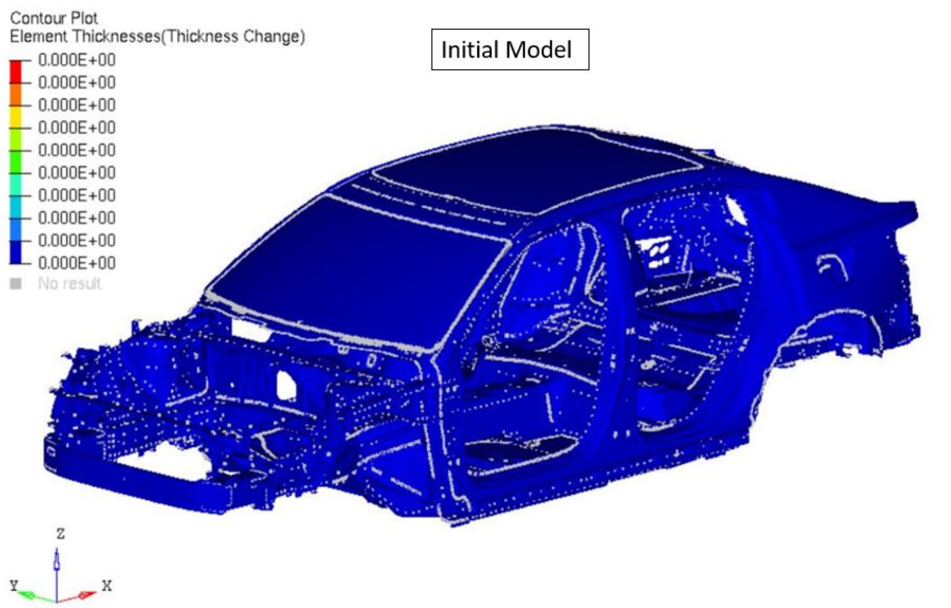
(a) Initial model



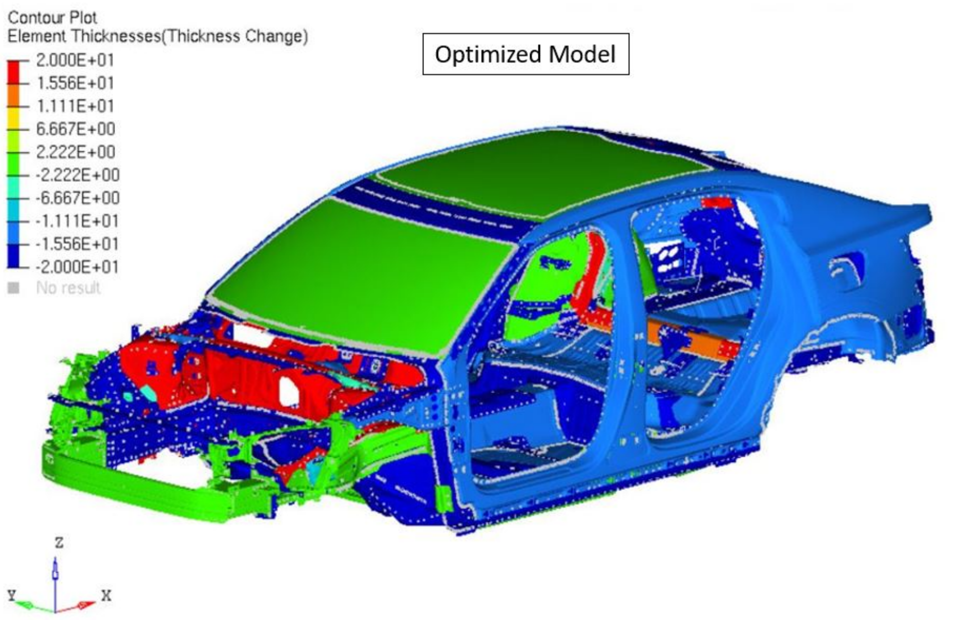
(b) Optimized model

Figure 6.1: Design optimization results - thickness of the parts

Fgure 6.2 shows the percentage change in thickness of the parts. As mentioned earlier, blue indicates a reduction in thickness and red indicates an increase in thickness.

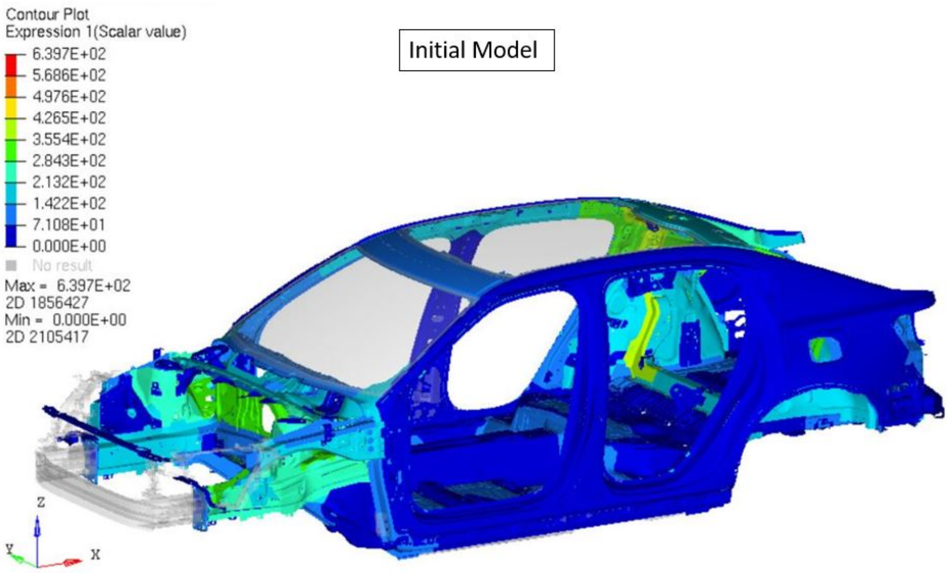


(a) Initial model

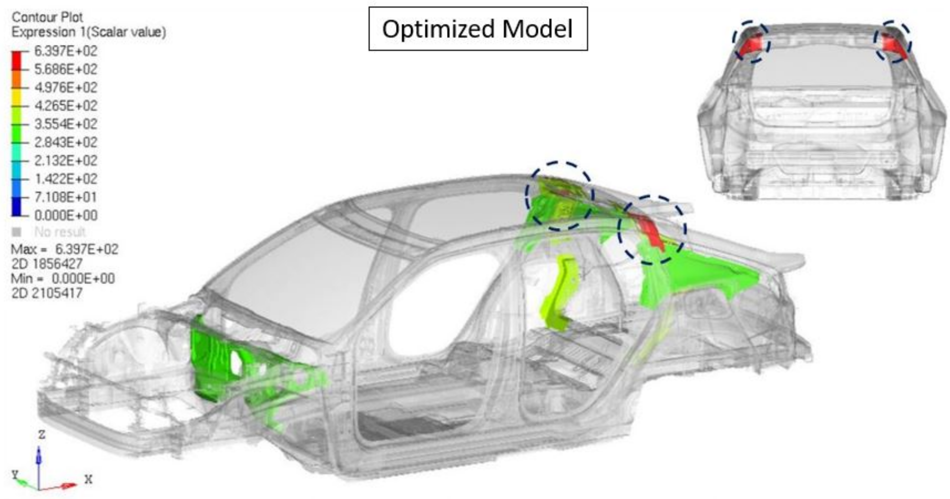


(b) Optimized model

Figure 6.2: Design optimization results - thickness change



(a) Initial model



(b) Optimized model

Figure 6.3: Design optimization results - KTG/Mass sensitivity

Figure 6.3 shows the sensitivity analysis. It is a representation of stiffness sensitivity in relation to the mass sensitivity. This helps to identify parts that influence the body torsional stiffness in the most weight efficient way. Figure 6.3b shows the area above the C pillar being critical for body torsional stiffness and can be seen as the most weight efficient component with high body torsional stiffness. This type of sensitivity contour is done to visualize the parts that can be optimized further, whether to make them stiffer or lighter; i.e. green and cyan parts can be either increased or decreased in thickness to achieve either stiffer or lighter parts respectively.

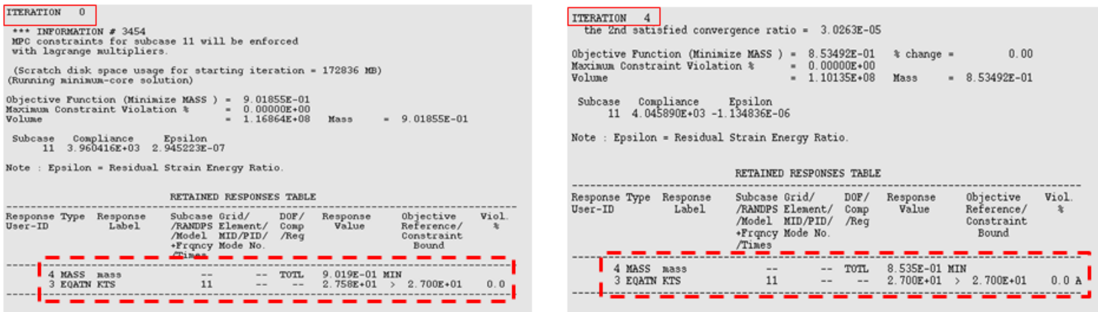


Figure 6.4: Design optimization results - mass change

Figure 6.4 shows output card images of mass change. Initial mass of the BIG structure is **901.9 kg** and final mass after optimization is **853.5 kg**. Mass has been reduced by 48.4 kg which represents 5.3 % of total mass of the BIG structure.

7 Conclusion

Body torsional stiffness is one of the crucial aspects when it comes to squeak and rattle. In this thesis, different methods to evaluate the body torsional stiffness have been validated. Also, a design optimization for torsional stiffness load case is carried out. This section provides conclusions of the thesis work.

Since the Model-A does not, yet, have results from the test-rig experiment, the results of **Adams** simulation can be considered as the best available result for the validation purpose. It can be observed that the full-car FE model with the chassis and other parts included, contribute more to the body torsional stiffness. Thus a near correlation is achieved for the Model-A with ADAMS results as the validation result.

For achieving a better correlation, more models are required and also same design status model should be used in the simulation. Since there are model and method discrepancies, a clear correlation pattern could not be established here. Also, it can be seen from the table 4.1 that the results from **Adams** simulation are closer to test rig data, since this simulation method replicates the test-rig with the four-poster and its loading conditions. Hence, it can be concluded from the work put forward here that the **ADAMS Simulation** can be preferred for validating the test-rig experiment results.

As for the design optimization, only the body torsional stiffness load case is used with a constraint on torsional stiffness value to achieve weight reduction. The optimization had turned out to be a preliminary/basic optimization study with a reduction of about 50 kg of weight from the overall BIG structure.

In the following chapter, a few suggestions on the thesis future work and an extended scope of the thesis are described briefly.

8

Future Work

This section describes the future work and further recommendations that could be carried out on the current thesis work.

- Carry out the test-rig experiment method of evaluating the body torsional stiffness for Model-A. This will help in validating the test-rig experiment results against the simulations.

- The full-car FE model for Model-B can be built and a simulation similar to that of Model-A can be carried out on this. These results can then be compared with Model-A for a better understanding of the pattern in the results.

- The next suggestion is to have an ADAMS model with the same final design version of Nastran file as used in the static simulation. During the thesis, an earlier version of the model was used, since it fulfills the needs for the calculations. But the option of using the ADAMS model with the final design version was ruled out because of the time constraints. As the ADAMS results have shown closer correlation to the experimental values, using the same design version of the Nastran file can improve the accuracy of the results.

- Only two models were used in the thesis. An accurate trend could possibly be established if more models are to be considered and a similar validation is carried out. This can establish an accuracy percentage of the different methods, which is hard to establish with only two models.

- Regarding design optimization, as indicated by the results shown in the figure 6.3, the regions near the C-pillars can be further explored and the specific parts can be optimized to reduce weight.

- The design optimization is carried out using only torsional stiffness as the load case. But in reality, other attributes such as NVH, safety and durability all come into play. It could be fruitful to explore design optimization by considering all the different load cases simultaneously.

Bibliography

[1] BIW definition, referred from slideshare.com – BIW

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[3] Establishing Correlation between Torsional and Lateral Stiffness Parameters of BIW, referred from sae.org – Correlation

[4] Numerical assessment of the impact of vehicle body stiffness on handling performance, referred from citeseerx.ist.psu.edu – Impacts