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Structural And optimization Analysis of A Monocoque Car Body

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ABSTRACT: *The use of a monocoque structure-type in car industry becomes the ruler because of high torsional and bending stiffnesses of the structure; which are the prerequisite of independent wheel suspension needed for soft ride, and low ground clearance; which improve vehicle stability and performance. Stress analysis is carried out on an existing car model 'DOGAN 1.6 liter', because of availability of exact engineering drawings, to ensure structural integrity and continuity, mainly in the load paths. The modeling process is obtained by representing the real car body as an idealized structural model valid for applications of different structural analyses. Two main loading cases imposed on the car structure are studied namely: the bending and torsion cases by implementing different structural analysis theories. The main load carrying elements are identified for the different load paths from numerous bodywork structural components that performing both functional and load carrying members. The results obtained from different three- dimensional stress theories are compared at the three extreme internal stresses locations obtained from the one-dimensional load envelopes reveals good agreement between the results obtained from different stress analysis theories, yet the results obtained from both Finite Element Method (FEM) and Simple Structural Surfaces (SSS) models are more accurate so they're appropriate to be used in the latest stage of design. Optimization analysis is performed on the main load carrying members of the car body structure to reach their minimum weight in both bending and torsion loading cases using predefined stress, stiffness, and deflection constraints.*

Keywords: *Monocoque, Car Body Structural Analysis, Finite Element Analysis, Optimization Analysis*

INTRODUCTION

The main objectives in car structural design are to obtain a minimum weight design which makes the best utilization of material by arranging for each member to support as near as possible its maximum load potential, to make the structure direct and continuous by providing an unbroken path from point of application to point of reaction, to minimize weight; by spreading concentrated loads as much as possible; which are efficiently distributed to react applied load with the minimum redundancy, to avoid buckling of the thin sheet metals in monocoque structural type, and to provide passenger protection in accidents. [1] The stiffness of a vehicle structure has important influences on its handling and vibrational behavior. The vehicle structure has to be designed with enough stiffness to lower the structural natural frequencies to a range of 20-30 Hz to avoid the danger of excitation by suspension frequencies and subsequent loss of handling control. It's also important to insure minimum deflections due to extreme loads to impair the functionality of vehicle structure, e.g. the doors will not close, or suspension geometry is altered. Low stiffness can lead to unacceptable vibrations, such as 'scuttle shake'. [2]

There're two types of structural stiffness, depending on the loading cases applied namely, the bending and the torsional stiffnesses. As a structural design criteria, the mid-span bending deflection should not exceed (1.27 mm) and the door aperture deformation should not exceed (1.27 mm) for a 680 Kg mid-span load.[1], while the torsional stiffness between the front and rear axles in a typical family saloon car should be higher than (8,000 : 10,000 N.m/deg) and increases with luxury demands. [3] Although the modern passenger car has surfaces with high curvature due to aerodynamic and styling requirements, the structure behind these surfaces can be approximated to components or subassemblies that can be represented as plane surfaces. [4] The idealized representative structural model of the principal internal structural load carrying members is obtained so as to give a simple and accurate representative idealization of a generic real sedan car body structure [2]. This idealization is then applied on a real existing saloon car "Nasr DOGAN 1.6 liter" to obtain its structural model as shown in Fig. 1. The cross-sectional properties and gravimetric analysis need to be specified in the model as a part of the idealization of vehicle body structural model and to be used in the stress analysis that depends mainly on these cross-sectional properties.

The different types of loading cases are then calculated using The Dynamic load Coefficients "m" to replace repeated dynamic loads ‘P_{dyn}’ with static load ‘P_{st}’ in the analysis. Different methods of car structural analysis are applied; using suitable structural theories; for different structural models based on the structural idealization concepts.

II. METHOD OF APPROACH

Applying the FEM representative structural model, in which the structural surfaces are represented as combinations of the general beams and shells, in such a way that the structure to be capable of transferring all kinds of stresses, as shown in Fig.2, with the following properties: [5]

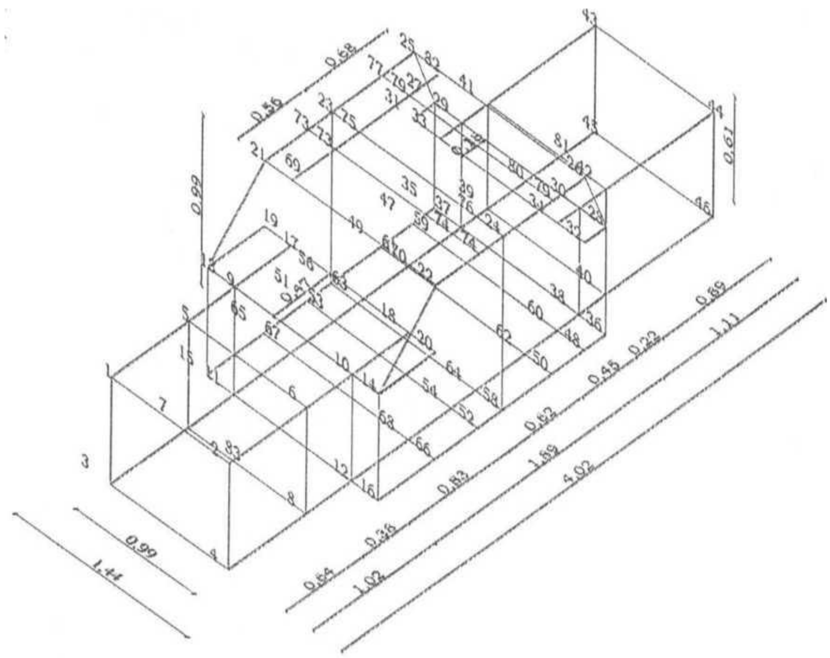


Fig .1. Structural model for “DOGAN” car body.

The types of elements used are:

a - Beam ‘4’ for symmetric cross-section beams with 6 DOF at each node.

b - Beam ‘44’ for un-symmetric cross-section beams with 6 DOF at each node .

c - Shell ‘63’ for the general rectangular shell capable of transferring membrane and bending stresses with 6 DOF at each node.

1. The analysis type is static with small deflection.
2. In the coarse mesh model:

a. The number of nodes is ‘82’ nodes.

b. The number of elements for each type are:

1) Beam ‘4’ has 161 elements.

2) Beam ‘44’ has 35 elements.

1. Shell ‘63’ has 54 elements.
2. The boundary conditions at the four contact points are simply supported.

The meshing effect is then studied to ensure the credibility of the obtained results and the convergence of the solution to an extent that insuring sufficient element mesh for accurate results maintaining less time consuming. This will be done by making three models with different meshing stages as follows:

a- The coarse or the original model with (83) nodes.

b- The intermediate mesh model with (147) nodes.

c- The fine mesh model with (453) nodes.

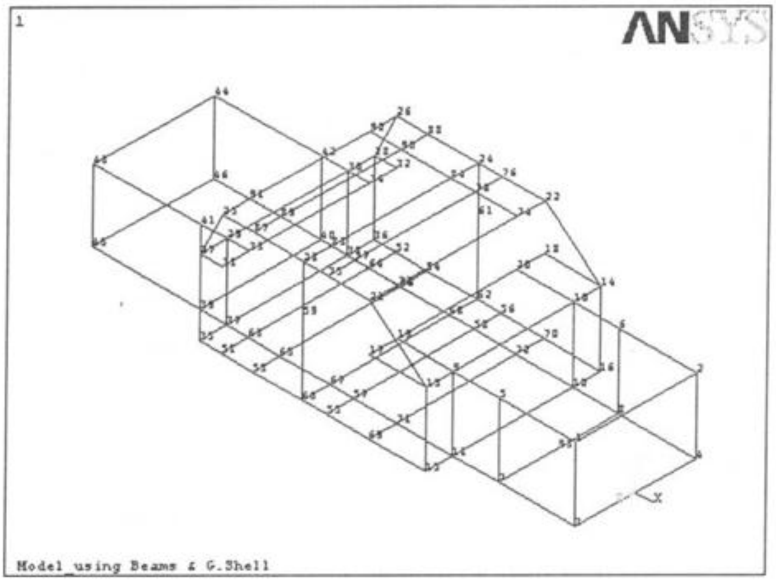


Fig .2. Three-Dimensional model FEM idealization using general shells and beams

Also, the effect of changing the type of fixations (strained DOF) of the structural model on the obtained results will be investigated in the above three models by changing the two rear contact points to be of the roller support type with one DOF constraints instead of simply supported contact points. Applying the SSS-Method which is a powerful conceptual tool for the auto body design to represent the vehicle structure as a course mesh finite element idealization of structural surfaces in early design stages allowing a good approximation of the behavior of the main elements in different loading cases with the ability to alter the design within one of them without the changes affecting the adjacent surfaces.

- Free body diagrams for both bending and torsion loading cases are drawn showing forces acting on the surfaces, and these are used to trace the most significant load paths through the structure, as shown in Fig.3.

erforming structural optimization process; to get the optimal construction under different constraints (strength, deflections, and stiffness) that is capable of carrying the required loads and meets all specified requirements with a minimum component weight for both bending and torsion cases; by using the suitable optimization method. The First Order Method is based on 'Castigliano's Theorem' that depends on strain energy and computes the partial derivatives of the nodal displacements and other state variables with respect to structural design variables. Defining the structural model properties in terms of parameters from which Design Variables (DVs) are selected. DVs are selected from the beam and shell thicknesses, and specify the State Variables (SVs) that serve as design constraints.

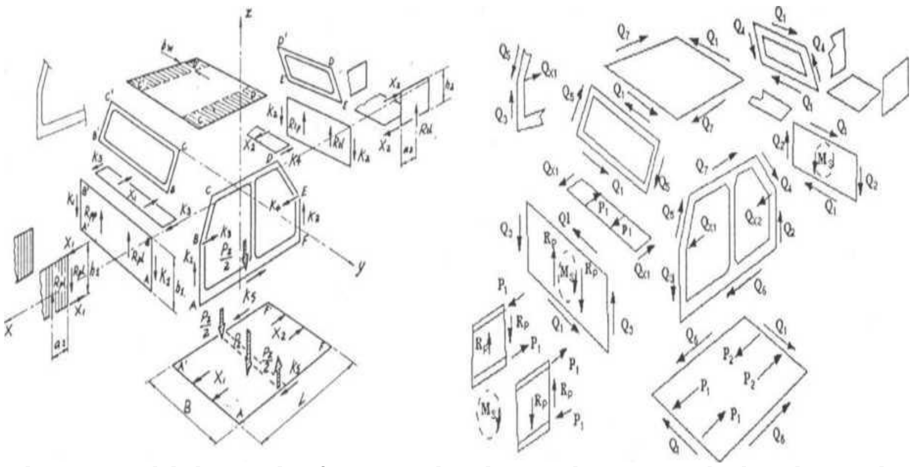


Fig .3. SSS idealization model shows edge force in a closed integral private car both under Bending and Torsion cases showing the total applied loads and the corresponding internal shear forces between different structural surfaces.

II. DISCUSSION OF THE RESULTS.

The comparison between the different 3-Dimensional analysis models and the one-Dimensional model at the three extreme high stress sections is done in Table 1. From the comparison; it is obvious that the One-Dimensional analysis has almost the higher values than any other stress analysis since it was carried out on the load envelopes (the extremes of the different loading combinations), and the following conclusions can be deduced and summarized:

1- FEM representative model is the closed one to the real model.
2- The values obtained from different analysis theories compared with that obtained from the FEM are more conservative and less expensive, except the SSS-method.
3- One-Dimensional and 3-D SFM can be used only in early stages of design.
4- In the latest stage of design, both FEM and SSS-Method can be used for getting more accurate results, yet SSS-Method is more preferable, in the first design stages, for its simplicity and easy detection of structural discontinuity.

1- Effect Of Type Of Fixation And The Degree Of Meshing On The Obtained Results

The comparison of the obtained results from different meshing stages, showed a convergence of the deflection components at all nodes as going through from coarse to fine mesh stages. There are some local areas in which the deflections at the corresponding nodes, especially the vertical components, have high values, namely : the main floor assembly, B-pillar beam, and the roof assembly. It is noticed that all these nodes are in the passenger compartment because there is no intermediate vertical shear panels. So, making a fine mesh at these areas will be reflected on the results and get a clear picture on their behavior.

2- Optimization Results

A comparison was made for the best feasible set of the optimization data files; that satisfy all specified constraints within allowable limits of DVs and producing the minimum weight; obtained from both the Torsion and the Bending Cases. Then selection of the DVs that satisfy both the Bending and Torsion cases, which converge to the nearest integers that could be used as a sheet metal thickness, and running the programs to obtain the model with the best selection of DVs that are reliable at the worst operating conditions and satisfying all stiffness constraints. Results of the sensitivity analysis of the optimization processes are shown in Table 2.

Table 1. Comparison Between The Different Three-Dimensional Stress Analysis Models And The One-Dimensional Model, At The Three Extreme Sections.

Sections	Theory	One-D. Beam	3-D FEM	3-D SFM	3-D SSS_M-
Front wheel panel section	Shear Force (N)	2181.5	1598.7	2653.8	3500±1453
	Shear stress (Wheel panel) (Mpa)	2.39	2.79	4.88	2.34
	Normal stress - suspension beam (Mpa)	4.04	2.53	4.91	2.64
	Normal stress - UPR panel beam (Mpa)	2.82	0.79	6.29	3.19
	Normal stress - LWR panel beam (Mpa)	2.23	1.05	1.87	2.54
Front crossmember section	Shear Force (N)	1066	1301.5	637.86	1585.58
	Shear stress (Roof-floor panel) (Mpa)	1.245/0.42	0	0.2	0.69
	Normal stress - B-pillar beam (Mpa)	6.27	4.22	3.9	9.33
	Normal stress - UPR sill beam (Mpa)	11.72	2.75	-1.94	4.88
	Normal stress - LWR rail beam (Mpa)	3.98	0.53	4.43	1.1
Rear wheel panel section	Shear Force (N)	4272.5	1549	776.71	4000H453.
	Shear stress (Wheel panel) (Mpa)	2.94	-3.12	-1.93	2.34
	Normal stress -suspension beam (Mpa)	0.95	0.68	0.5	0.52
	Normal stress - UPR panel beam (Mpa)	10.83	-0.36	2.2	2.86
	Normal stress - LWR panel beam (Mpa)	1.97	1.17	3.98	0.92

Table 2. Comparison between the starting, best optimization in Bending and Torsion, and the manual combined optimization model for the most governing DVs.

DVs and SVs	Starting	Best	Best	Final
Beam thickness# 1(mm)	1.0	1.76	1.35	1.8
Beam thickness# 2(mm)	1.5	1.49	1.62	1.5
Beam thickness# 3(mm)	2.0	1.46	0.79	1.5
Beam thickness# 4(mm)	2.5	0.5	0.5	2.0
Beam thickness# 5(mm)	1.5	1.39	1.23	1.4
Shell thickness# 1(mm)	1.0	1.00	1.0	1.0

Shell thickness# 2(mm)	2.0	1.49	1.31	1.5
Shell thickness# 3(mm)	2.5	0.71	0. 8	2.0
Shell thickness# 4(mm)	3.0	0. 96	0. 5	2.5
Total car volume (m^3)	0.04504	0.03975	0.034314	0.04509
Torsional stiffness, calculated (N-m/deg)	16876	20201	—	19526
Torsional stiffness, Theoretical (N-m/deg)	14686	20483	—	19582
Total strain energy.	11.903	10.003	15.11	10.581
Mid-span deflection, (mm)	2.7426	—	2.3941	2.3595

IV. CONCLUSIONS

The structural analysis was performed, on a real existing sedan closed integral monocoque passenger car, by using dynamic load factors to represent the dynamic loading conditions imposed on the structural model to be nearly as the reality.Results revealed that the torsion loading case has higher internal forces and stresses in some members of the load paths than the corresponding bending loading case.

Results showed that beams with high internal forces can be summarized in the following sections:

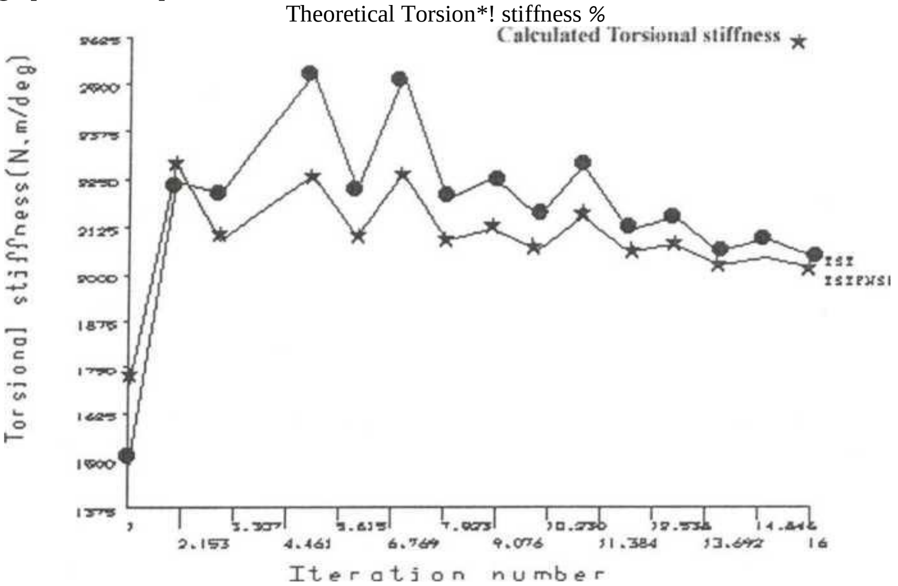
a. Front and rear suspension strut support beams.
b. The lower forks (the longitudinal internal rails that extends from the front end assembly under the main floor assembly).
c. The two main longitudinal rails that extend allover the structure in both the central part (passenger compartments) and the rear-end assembly.
d. The upper and lower beams of the front and rear bulkheads.
e. Rear seat cross-member.
f. The optimization analysis showed that some sheet panel assemblies thicknesses can be reduced without affecting the total structural performance, and the increase in thickness of some beam cross-sections can enhance the total structural Torsional stiffness.

g. The comparison between the theoretical and the calculated torsional stiffnesses of the overall car body structure in the optimization process for the combined torsion loading cases are almost consistent and are enhanced through the optimization process, as shown in Fig.4.

Fig .4. Comparison between the calculated and theoretical total torsional stiffnesses variation during optimization looping, Torsion Case.

h- The comparison between the total car volume, which is a function of its weight) shows a decay in its value through the optimization process, Fig.5. shows a comparison between the total car volume in both Torsion and Bending optimization process.

Theoretical Torsion*! stiffness %



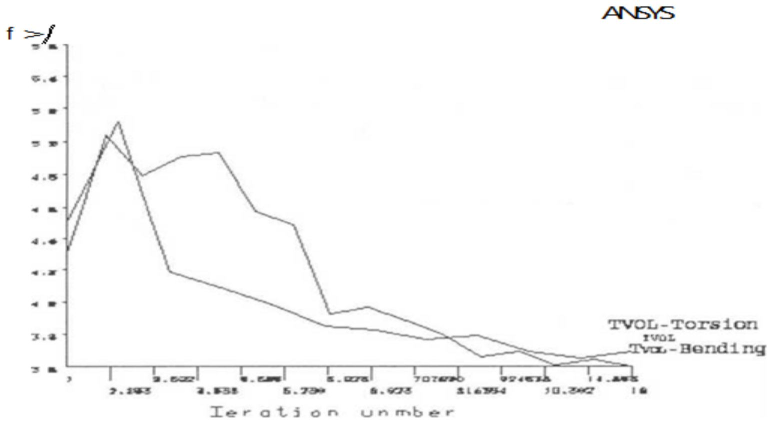


Fig .5. Comparison between the total car volume variation during optimization looping in both Bending and Torsion cases.

The maximum mid-span deflection, which is used as a criterion of the combined bending loading cases, has always lower values below the permissible allover the optimization process. Although both the bending and torsion best optimization results give smaller DVs values, they take values near the starting in the final manual selection because of constraints on "DVs" imposed on them as follows:

1. - Beam thickness# 3 (engine cross-member) because of bunching and fatigue imposed on
a. it during operation.
2. - Shell thickness# 4 (Floor panel assembly) because of having sufficient stiffness for the payload and other point-loads acting perpendicular on it, witch are corrugated at some local areas.

The torsion optimization analysis results are affected by an increase in 'Beam thickness# 1' which include the following beams (Front wheel panel upper beam, Inner front bulkhead beam, A-pillar upper and lower beams, Front windscreen upper and lower beams, Rear windscreen upper and lower beams, Upper sill beam (Roof), and D-pillar upper and lower beams) which form stiffening closed box of the passenger compartment.

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