Formulating Structural Design and Analysis Practices Using First Order Analysis

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Formulating Structural Design and Analysis Practices
Using First Order Analysis

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Abstract. This paper addresses the issues encountered in both the conventional and advanced automobile design and analysis method by introducing First Order Analysis (FOA). Conventional method idealizes automobile structure as framework structure connected by beams. Sizing of the beam sections is based on simplified structural of mechanics calculation. Beam idealization is simple but potentially causes design faults, as it does not correspond to panel/surface models appearing in subsequent stages of product development. This problem complicates further due to highly redundant automobile structures whereby closed form solutions are not possible. Fortunately, these issues have been resolved with the application of Finite Element Method (FEM) in the early eighties. FEM has become the "de facto" numerical tool for solving statically indeterminate structure, which is a characteristic of modern automobile structure. FEM faces greater challenge as each design problem is unique and 'off- the-shelf' FEM software is usually not applicable. Customization of software is inevitable to construct appropriate FE idealization for design and analysis. Design and analysis practices using FOA are therefore being introduced to realize a less complicated and inexpensive tool for design engineer to evaluate behaviour of automobile structure to swiftly estimate "close to right" design. The template formulated by the FOA in this paper is the deployment of the best practices for automobile structures.

1. Introduction
In the mid-seventies, trying to create anything sophisticated in three-dimension (3D) to be meshed in the Finite Element Analysis (FEA) software was a nightmare. Depending on model complexity, the FEA jobs would queue up and run over the next few days on pitiful processing speed. It was too easy to get bogged down with the intricacies of FEA input syntax and the output was on miles of fanfold paper. Debugging of failed jobs was challenging and time-consuming.

Incredible improvements in computing power and software efficiency over the subsequent 40 years have positioned FEA to be the design and analysis tool for designer. FEA has also been recognized to replace prototyping and experiments. But rapid changes in geometry demand swift re-generation of mesh, loads and boundary condition in FEA. In this scenario, FEA has difficulty in keeping up with the latest geometry unless they are linked together.

New company engaging in the design and analysis of automotive vehicles faces tremendous challenges. This company does not have any legacy data to construct CAD geometry for analysis. Furthermore, FEA cannot be used to establish a concept before the start of modelling or idealization as a complete geometrical definition must first be defined before analysis can be performed. Due to these underlying issues, First Order Analysis (FOA) is formulated to evaluate structure performances nearly simultaneously and in parallel with model changes. The ability to inject sensible design inputs into the
early development cycle when it has the most potential to optimize design. FOA is a complimentary tool to FEA to promptly estimate ‘close to right’ section design before embarking in expensive and time-consuming FEA. Section design information will be used to configure the appropriate section property values and yield structural function values. This tool also enables design interrogation to be carried out on the model to shortlist design parameters that contribute to structural optimization based on basic knowledge of structure of mechanics such as beam and shell theory.

2. Thin-walled idealization of vehicle structure

Vehicle body panels are predominantly thin-walled to realize the advantage of high-stiffness to weight ratio. The larger is this ratio, the more optimal is the structure. In addition, thin-walled structures enjoy the unique position of having extremely high aesthetic value in various shape designs. To qualify as “thin”, the wall thicknesses should be small, typically 10% or less when compared to the overall cross-section dimensions. Automobile body structure typically consists of panels with thin-walled sections in which the width-to-thickness ratio, $b/t$, is large ($b/t > 60$).

Thin-walled sections can be open (figure 1) or closed (figure 2). The theoretical developments of open-thin-walled beam are briefly reviewed here. Researches in the stability of open thin-walled members have been carried out extensively since the early works of Vlasov [1], and Timoshenko and Goodier [2] based on the theory of elasticity. The section torsional constant $J$ is approximately equal to the sum of the $J$’s for the constituent thin walled members as given by equation (1).

$$J \approx \frac{1}{3} \frac{b}{t} \sum_i^n$$  \hspace{1cm} (1)

While the maximum shear stress ($\tau_{\text{max}}$) is estimated as

$$\tau_{\text{max}} = \frac{T t_{\text{max}}}{J}$$  \hspace{1cm} (2)

![Figure 1. Common open thin-wall sections [3]](image1)

![Figure 2. Common closed thin-wall sections [3]](image2)

Closed thin-walled sections are much more effective in resisting torque than similarly configured open thin-wall sections, sometimes by orders of magnitude. This effectiveness is particularly significant for automobile panels because it determines the weight utilization and hence fuel economy in the long run.

For thin-walled closed sections, the general formula for torsion constant is given by [4]:

...
where \( A_E \) is the enclosed area by the walls, \( t \) is the thickness, \( ds \) is a length element along the perimeter. Integration is performed over the entire perimeter \( s \). It is important not to confuse this with the cross section area, also called the material area or the wall area. For thin-walled closed sections the shear flow \( q \), at any point is given by [5].

\[
q = \frac{T}{2A_E} \tag{4}
\]

The maximum shear stress occurs when the thickness is minimum \((t_{\text{min}})\), i.e.

\[
q_{\text{max}} = \frac{T}{2t_{\text{min}}A_E} \tag{5}
\]

Maximum shear stress evaluation provides an excellent performance indicator for automotive thin-walled panels as these panel can only take shear load. Out-of-plane load is insignificant.

3. First Order Analysis (FOA)

First Order Analysis was initiated and widely used by the Toyota Central Research & Development Laboratory (CRDL) in the early twenties. Kojima [6] introduced FOA to achieve pre-product shape template function and spreadsheet function that allowed a product to be divided into parts with performance calculation embedded. Geometry shape and design parameter can be optimized subsequently. On the other hand, [7] proposed FOA that was linked with ANSYS Parametric Design Language (APDL) to handle other elements (besides beam and panel) and functions. ANSYS has a powerful solver for structural and multi-physic analyses. It is very useful for users who are well-trained and have thorough FEA theory and applications knowledge. It is not the intention of this paper to elaborate the advantages of FOA linked with ANSYS, rather to unveil the potential of realizing the designer and analyst role as one without engaging in expensive tool such as ANSYS. This paper introduced an extremely fast route to ensuring the vehicle’s body structure meets or exceeds the relevant performance targets. The objective of developing a ‘right first time’ approach to the design and analysis of an automotive body structure is examined. Early vehicle structure assessment is conceivable. The potential weakness or problem can be identified and solutions can be developed to minimize downstream changes and churns. The main components of the aforementioned FOA are illustrated in figure 3.

3.1 Load flow rationalization

In reference to figure 3, the first three components are carried out based on classical Structural of Mechanics approach. Appropriate assumptions are applied and only loads that cause elastic strains and stresses are studied. Small deflection conferring to Hooke’s law is observed.

For the fourth component of figure 3, a well-accepted tool termed as Simple Structure Surfaces (SSS) method developed by Dr Janusz Pawlowski of the Warsaw Technical University is presented as a means of organizing the process for rationalizing the basic vehicle body load paths [8]. As vehicle structure is highly redundant, it is necessary to make simplifying assumptions to attain most desirable load path. There is more than one load path and the sharing of the loads is a function of the component relative stiffness and geometry. The vehicle to be modelled using SSS method will be represented
using a number of plane surfaces or panels. Each plane surface must be held in equilibrium by a series of edge forces. These forces are created by virtue of the weight of the components attached to them. The rails that are attached to adjacent plane surfaces would provide reactions to maintain equilibrium. Equal and opposite forces will be exerted to the adjacent members. The loads on each SSS member are propagated through the whole structure from one rail to another until the overall equilibrium is achieved. This way, any deficiency of the plane structures such as discontinuity in load path can be determined easily. Any SSS that is not supported adequately due to omission of a suitable adjacent component or panel will be revealed. This revelation in turn indicates stiffness deficiency. This method allows fast calculation of edge loads acting between simple structural surfaces based on the knowledge of structures of mechanics. Accompanying equilibrium equations developed will be embedded in Microsoft Excel to perform the calculation.

START

Determine external loads

Draw free body and loading diagrams

Calculate reaction forces

Rationalize load flow using SSS method

Formulate the matrix equations to solve for the edge forces (in Excel)

Size structural elements

Linking SSS method with Excel (verification and interrogation)

Fulfill design requirements?

Yes

No

END

Figure 3. First Order Analysis methodology flow chart.
3.2 Formulating equilibrium (matrix) equations

Two commonly used load cases to evaluate the performance of a vehicle are bending and torsion. As torsion case is the most difficult to design for, both [8] and [9] agreed that torsional stiffness should be used to benchmark the vehicle structure performance. If a structure is adequately design for torsion, bending should not be an issue. Figure 4 shows the torsional stiffness benchmarking for body-on-frame passenger vehicle structure extracted from [10].

![Figure 4. Body-on-frame torsional stiffness benchmarking [10].](image)

Vehicle under torsional loading condition is believed to undergo twist on the body as illustrated in figure 5. The torque $T$ is applied about axis O-O as $R_{FT}S_F$ at the front suspension. Equal and opposite rear couple $R_{RT}S_R$ is produced to balance this. Torque, front and rear reaction forces ($R_{FT}$ & $R_{RT}$) are evaluated once the front distance $S_F$ and rear distance $S_R$ are available.

![Figure 5. Vehicle in pure torsion.](image)

Adopting systematic SSS approach to analyze the vehicle in torsion. The vehicle panels are analyzed one at a time starting with panels containing the input loads, which are calculated separately. Figure 6 shows an exploded view of a 4 x 4 pickup truck under torsional loading to demonstrate the SSS method. The SSS edge loads based on this condition are evaluated accordingly. It can be clearly seen that edge loads and end loads are needed to ensure equilibrium in all SSSs. Equations for the internal forces $Q_1$ to $Q_9$ are derived based on these forces in each panel as illustrated below.
Figure 6. End and edge loads for a 4 x 4 Pickup Truck in torsion.

Left and right front fender 1 and 2 and left and right rear fender 3 and 4

For moment equilibrium, the couple caused the offset $L_1$ due to forces $R_{FT}$ is balanced by complementary shear force $P_{FT}$ at the top and bottom of the panel. The upper and lower flanges of these fenders are usually treated as a ‘boom’ as they are mounted to the rails consisting of a box member.

\[ T = R_{FT}S_F = R_{FT}S_R \]  

\[ R_{FT} \frac{T}{S} \]  

\[ S \frac{T}{S} \]  

\[ R_{FT}L_1, P_{FT}h_1 \]  

\[ R_{RT}L_2, P_{RT}h_2 \]  

Engine bulkhead 5

The shear forces $R_{FT}$ are transferred from the fender webs to the engine bulkhead, resulting in a couple $T = R_{FT}S_F$. This couple is reacted by $Q_1$ and $Q_2$ shear forces acting on the edges of the bulkhead forming couples $Q_1h_1$ and $Q_2B$. For moment equilibrium:
\[ Q_1 h_1 \quad Q_2 B_2 \quad T_f \]  

**Front parcel shelf**

The front parcel shelf carrying the couple \( P_{FT} S_F \) out to the side frame at the mid \( A \)-pillars. The end forces \( Q_3 \) form a couple to balance this.

\[ Q_3 B_2 \quad P_{FT} S_F \]  

**Front windshield**

Shear force from the top of the engine bulkhead is reacted by an equal force on the bottom of the windshield frame.

\[ Q_1 h_1 \quad Q_4 B_2 \]  

**Roof**

Shear force \( Q_1 \) from the top of the windshield frame is fed to the front of the roof. This will be balance by similar edge force \( Q_1 \) at the rear. These edge forces form a couple \( Q_1 L_4 \) which in turn is balanced by a complementary couple \( Q_3 B \) from the edge forces \( Q_3 \) on the roof sides. To maintain moment equilibrium:

\[ Q_1 L_4 \quad Q_3 B_2 \]  

**Rear window frame**

Similar to roof, shear force \( Q_1 \) flow downs from the roof formed a couple \( Q_1 h_4 \), and the complementary couple \( Q_6 B \) from edge forces \( Q_6 \) on the sides of the rear window frame balance this.

\[ Q_1 h_4 \quad Q_6 B_2 \]  

**Rear parcel shelf**

Identical to front parcel shelf, the rear parcel shelf carrying the couple \( P_{RT} S_R \) out to the side frame at the mid \( C \)-pillars. The end forces \( Q_7 \) form a couple to balance this.

\[ Q_2 B_2 \quad P_{RT} S_R \]  

**Rear seat bulkhead**

The couple \( T = R_{RT} S_R \) due to the rear fender webs is fed to this panel. Shear forces \( Q_1 \) passed from the rear window frame and the floor, and the forces \( Q_8 \) on its sides form a complementary couple to balance \( T \).

\[ Q_1 h_2 \quad Q_1 B_2 \quad T_R \]  

**Floor**

The floor receives equal and opposite edge forces \( Q_1 \) from the front and rear bulkheads and \( Q_9 \) at the sides (both) from the sideframes. The forces \( Q_1 \) and \( Q_9 \) are oriented so as to form complimentary couples \( Q_1 L_3 \) and \( Q_9 B \). These will balance out couples due to \( P_{FT} S_F \) and \( P_{RT} S_R \) from the front and rear fenders. For moment equilibrium:

\[ Q_1 L_4 \quad Q_9 B_2 \quad P_{FT} S_F \quad P_{RT} S_R \]  

**Sideframes** and

Figure 7 illustrates the edges of the sideframe react edge forces \( Q_2, Q_4, Q_5, Q_6, Q_8, Q_9 \) from the surfaces attached to it. While \( Q_3 \) and \( Q_7 \) are edge forces reacting from the front and rear parcel
shelves. The opposite sides of the sideframe experience identical edge loads, but in opposite direction. It is evident that the sideframes are paramount in assembling the edge forces from essential surfaces.

Take an arbitrary point \( G \) to ensure moment equilibrium. Point \( G \) is located at \( X \) from the A-pillar and \( Z \) above the rocker centreline. Giving \( r_2, r_4, r_5, r_6, r_8, \) and \( r_9 \) are the respective moment arms for forces \( Q_2, Q_4, Q_5, Q_6, Q_8, \) and \( Q_9 \) about point \( G \), thus (taking clockwise moment as positive):

\[
r_2Q_2 - r_4Q_4 - r_5Q_5 + r_6Q_6 + r_8Q_8 - r_9Q_9 = 0
\]  

(19)

\[Q_6(h_2 Z)\sin\theta_B (L_5 X)\cos\theta_B Q_7(h_2 Z) Q_8(L_5 X) Q_9 Z = 0\]  

(20)

For the pickup truck under torsional load, equations (11 – 19) are rearranged and solved in Microsoft Excel. Solutions for these nine simultaneous equations with nine unknowns are obtainable once the relevant dimensions are defined and inputs values of \( T, P_{FT}, P_{RT}, Q_4 \) and \( Q_6 \) are provided in terms of \( T \).

3.2.1 Solving for the edge forces using Microsoft Excel

The systems of equations developed for the torsional loading are solved in Excel to complete the FOA tool. Graphical user interface (GUI) is created in Excel to allow for design communication and interrogation. The GUI data is arranged in rows and columns and editable easily. It is constructed to enable access to the whole workbook. That is, it should be possible to switch quickly between the individual sheets in the workbook when editing the spreadsheet. A data entry form providing support for changing global parameters is configured. Figure 8 summarizes the complete computation process in Excel to obtain the solutions. All computation processes are performed within the same interface. Matrices would be solved using the two built-in functions as follows:
MMULT(matrixA,matrixB) for finding the matrix product AB
MINVERSE(matrixA) for finding A⁻¹

Figure 8. Solving systems of equations for the 4 x 4 Pickup Truck in torsion using Microsoft Excel.

3.2.2 Interactive design assessment using FOA

Essentially, the GUI created enables easy and efficient design interrogation. One can perform design queries as and when needed during the design process as it is possible to switch among the individual sheets quickly in the workbook when editing the spreadsheet. Figure 9 consolidates the interrogation processes. a) It shows the CAD embedded 4 x 4 pickup truck in exploded view with the main load bearing members. b) Clicking ‘External forces’ text button will open up the sheet that is used to determine the external forces. c) Clicking on the “Edge forces matrices menu” reveals all the internal forces computed. d) To interrogate the model in more detail, manipulate the slider bars on the Engine Bulkhead sheet, the value of the associated forces would be updated in the “Forces flow diagram” sheet. Any sheet that used the same dimension would be re-calculated and updated at once as well. e) Logically, summary of all edge forces sheet links to “Edge forces matrices menu” and all the other associated calculations. Therefore, shear forces on each member are apparent to the designer to size the beam section without performing cumbersome analysis or computation. The thickness of vehicle panel is usually designed according to industry norms and subjects to manufacturing constraints. Once it is defined, output from this “best first guess” model is ready for detailed design downstream.

The same procedure can be repeated for any other panel to investigate the internal forces simply by manipulating sheets in the workbook.
4. Structural design and analysis best practices

Simple Structure Surface method reviewed above helps to highlight the possible load flows through the structure. It helps to highlight where high loads are to be found and how the adjacent members are sharing loads. Any new designer can refer to the characteristics and ‘best first guess’ values of the loads on major components. This information ensures required structural features are incorporated in the initial stage of the design where models that are more tangible are lacking. This work takes one-step forward by integrating SSS method with Microsoft Excel to allow for testing or examination of various concepts ahead of time. Harvesting on this inspiring effort is the First Order Analysis (FOA). One can establish the boundaries or limits from which the detailed design can start, especially if the possible designs are being debated. Overall sizing envelopes for the major structural components can be determined. Information regarding shape and material is left to the detailed stage unless it is of significant risk to justify early assessment.

SSS Method linked with Excel can be adopted as a template for best practices for any new designer to reveal the internal load path at the design concept stage and during the ongoing evaluation stage by FEA. Graphical User Interface (GUI) that is developed in Excel embedded with calculation results for the panels or models mimic the designer’s thinking process and to address ‘what if’ scenarios throughout the product development cycle. It is easy to carry out design interrogation for any new designer. Needless to say, outputs (loads) from FOA transferred to detailed CAE will shorten the process of structural optimization eventually as sensible geometry is provided to start with. With this, design know-how is retained within the model instead of the designer.
Process flows for the integration of SSS and Microsoft Excel are described in figure 10. It is also obvious that only basic equations are used to provide appropriate sections for the prescribed loadings. This approach is applicable for visualizing internal load path when only limited information such as length, height, external load inputs about the vehicle body structure are available since very basic parameters are known or estimated especially during the early design stage.

Figure 10. First Order Analysis best practices flow chart
5. Conclusion
By evaluating structure characteristics nearly simultaneously and in parallel with model changes, the designer can assess the effect of changing each dimensional parameter on the resulting forces. The sensitivity of basic dimensional changes is reflected by the resulting forces and moments. The magnitude of load changes and SSS edge forces can be used to judge the risk of introducing a new proposal. FOA allows more design freedom for ‘what if’ analysis on the stiffness characteristics of the shear panels. That is one can easily modify the size of any member in a fully parametrized way without further need of re-drawing or re-meshing the model to influence the shear flow to achieve the most suitable modifications in order to improve the functional performance of the full vehicle body.

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