DESIGN SENSITIVITY ANALYSIS FOR DURABILITY DESIGN OF BODY STRUCTURES

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ABSTRACT

Design sensitivity analysis (DSA) computes the derivatives of structural response quantities (e.g., displacements, stresses, modal frequencies, mode shapes) with respect to design variables (e.g., cross-sectional properties such as area, moments of inertia, torsional constant). These derivatives, defined as design sensitivity coefficients, give the designer a feel as to how the structure will respond to a proposed design change. Although the general concept of DSA has been well established, the application of this method to vehicle body durability design is relatively new. The current paper examines the relation between the body overall stiffness/strength characteristics and fatigue life. It also demonstrates how DSA can be employed to effectively identify design variables most affecting fatigue life through the body overall stiffness/strength evaluations. The methods and concepts are demonstrated using a very simplified finite element model which conceptually simulates a body structural system.

INTRODUCTION

Design sensitivity analysis (DSA) computes the derivatives of structural response quantities (e.g., displacements, stresses, modal frequencies, mode shapes) with respect to design variables (e.g., gages, cross-sectional properties). These derivatives, defined as design sensitivity coefficients, give the designer a feel as to how the structure will respond to a proposed design change. A survey of the methods applicable to the calculation of structural sensitivity derivatives for finite element modeled structures can be found in [1].

Fatigue life is a measure of how long a structure can last under specified operating conditions (e.g., vehicle proving ground durability course). It also indicates whether a structure is under-designed (inadequate life) or over-designed (excessive life). In either case, design changes are required in order to improve structural efficiency. This is the stage at which design sensitivity analysis plays the most useful role in a vehicle durability assessment process.

The ideal way to identify design variables most affecting the fatigue life of a body structure would be to perform sensitivity analysis using element fatigue life as the response quantity. However, element fatigue life must be derived from element component stress histories through a process involving stress equivalencing, cycle counting and cumulative damage analysis. Therefore, it is very difficult to establish a direct mathematical relationship between the design variables mentioned above and fatigue life.

Fatigue life, to a large extent, is dictated by the overall stiffness and strength characteristics (e.g., deflections under bending and torsional loads, frequencies of bending and torsional modes, stress influence coefficients under unit loads) of a body structure. The derivatives of deflections, frequencies and stresses with respect to design parameters can easily be derived, as indicated in [1]. Moreover, the design sensitivity analysis capability based on these derivatives is available in several commercial codes like MSC/NASTRAN. It thus becomes feasible to integrate the existing design sensitivity analysis method into a general durability assessment process.

The present paper demonstrates the effectiveness of the DSA method for durability design using a FE model conceptually simulating a body structure. The relation between the body overall stiffness/strength characteristics and fatigue life is also demonstrated. Although the analysis methods and concepts are demonstrated using a simplified body model, they are applicable to any body system.

BASELINE FATIGUE ANALYSIS

The analysis model is shown in Figure 1 and has:

146 nodes
116 shell elements
16 spring elements
18 rigid elements
92 beam elements

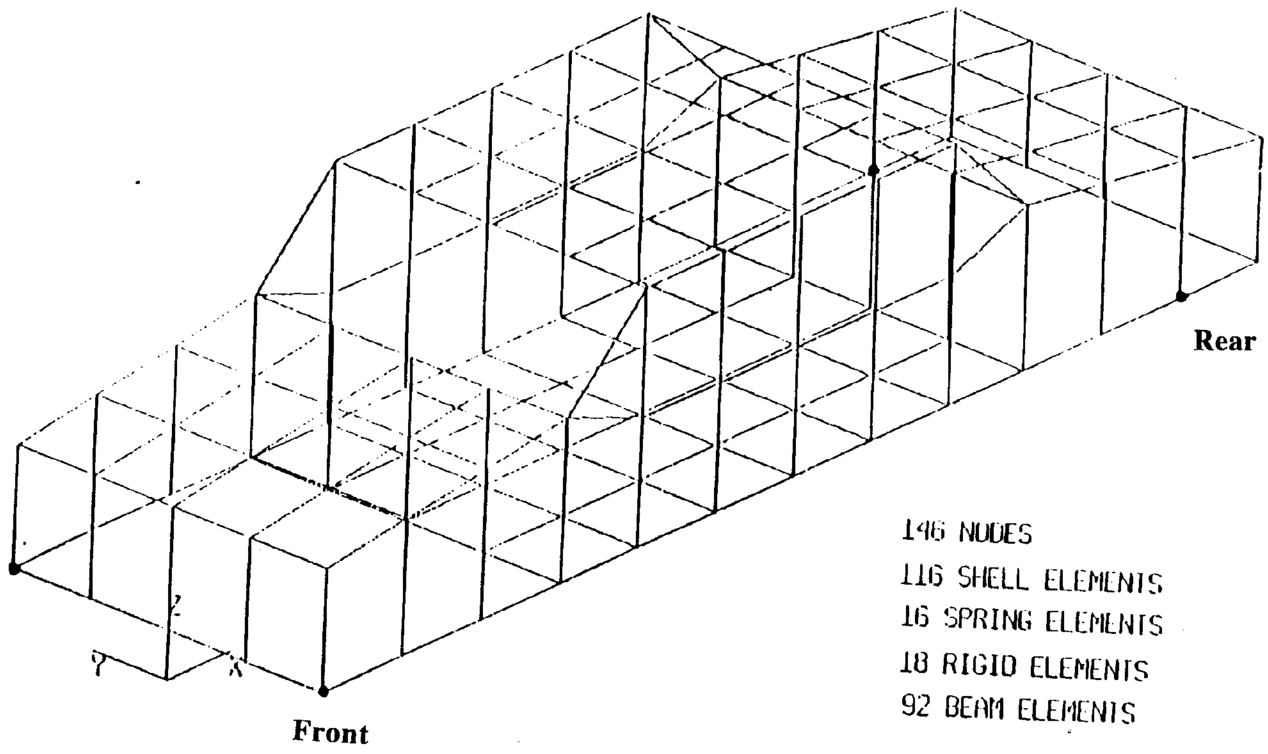

Figure 1 Concept model

For simplicity, all the shell elements were given the same thickness. The load carrying members such as the rockers, pillars, roof rails and radiator support were modeled as beams to provide reasonable stiffness to the model. Beam properties (area, area moment of inertia and the polar moment of inertia) were adjusted to yield similar stress levels as observed in a typical production body model.

Six channels of durability loads measured on a production vehicle at a proving ground were used as input to the analysis model. These loads were applied to the model at four locations (indicated with black dots in Figure 1) and in the following six directions:

- Left front vertical
- Right front vertical
- Left rear lateral
- Left rear vertical
- Right rear lateral
- Right rear vertical

An Inertia Relief analysis method, which takes into account the inertia effects of rigid body motion, was employed to determine the stress distribution in the model. The constraint conditions used in the analysis are shown in Figure 2. Fatigue life was computed for each element using a Ford in-house fatigue analysis program.

The elements with fatigue life less than 1 block are shown in Figure 3 (1 Block = total test miles on a rough road durability course = 100,000 miles of service life). The numbers refer to the fatigue life (in blocks) of the respective elements. The fatigue life values are asymmetric about the vehicle center line due to asymmetric loading although the analysis model is perfectly symmetric.

The following sections present different sensitivity analysis methods (Table 1) employed to identify the design variables which will affect fatigue life significantly when they are changed.

SENSITIVITY OF STATIC DEFLECTIONS

It is a known fact that the continual shake and distortion of a vehicle body under various operating conditions can cause a gradual weakening of the structure and lead to durability problems as indicated in [2]. Adequate overall body static bending and torsional stiffnesses are a necessary condition to avoid NVH and durability problems.

Static Bending Analysis

To demonstrate the effects of overall bending stiffness on fatigue life of a body structure, a traditional static bending analysis was conducted using the loads and boundary conditions shown in Figure 4. The deflection along the rails and rocker was determined and is given in Figure 5 (the curve connecting points represented with squares).

To identify the design variables most influencing the overall bending stiffness, a design sensitivity analysis was conducted using the maximum deflection as the response quantity and the thickness of all plate elements and cross-sectional properties (area, area moments

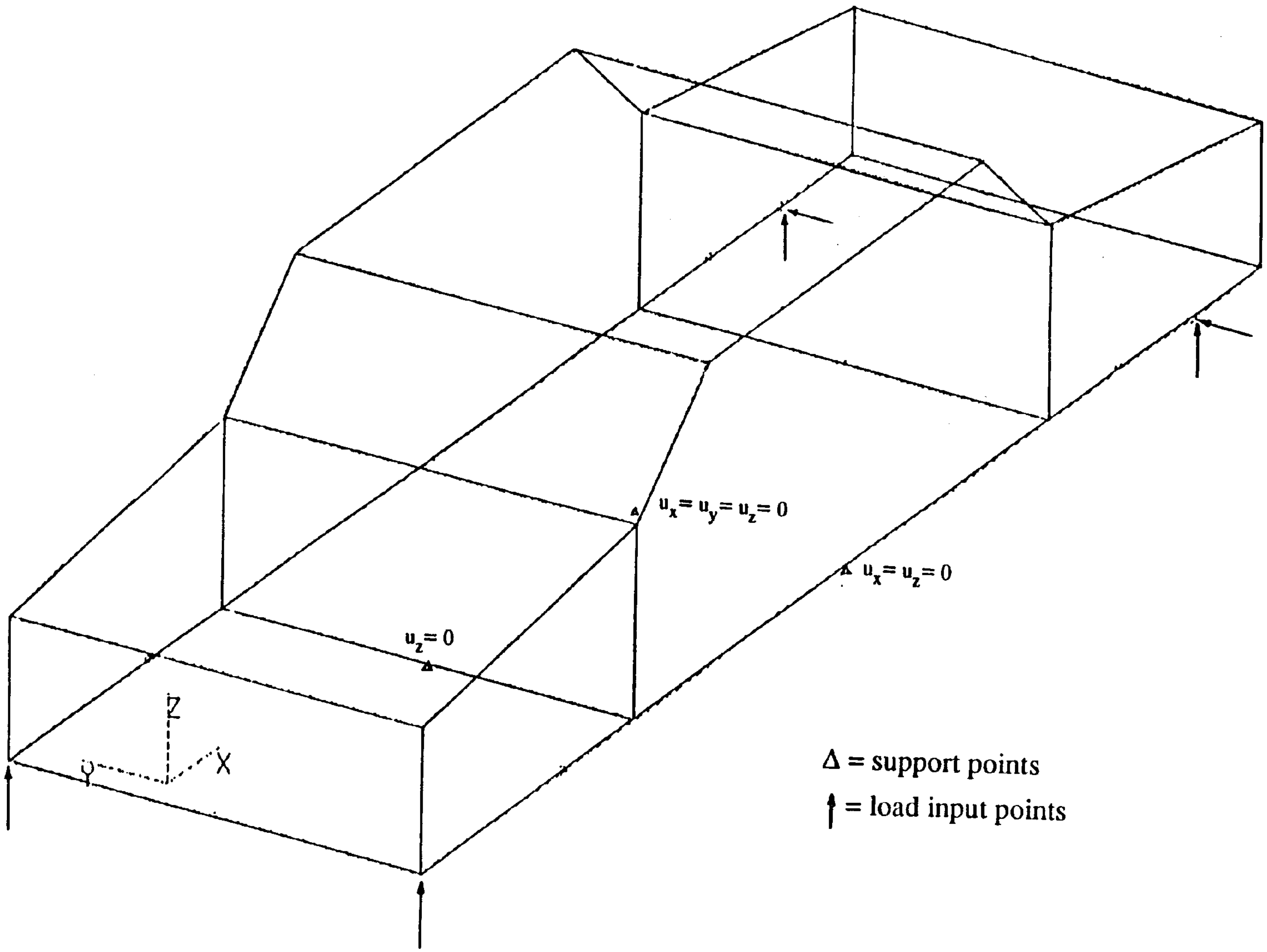

Figure 2 Inertia relief (stress, deflection as responses) - loads are applied separately at the six load input points

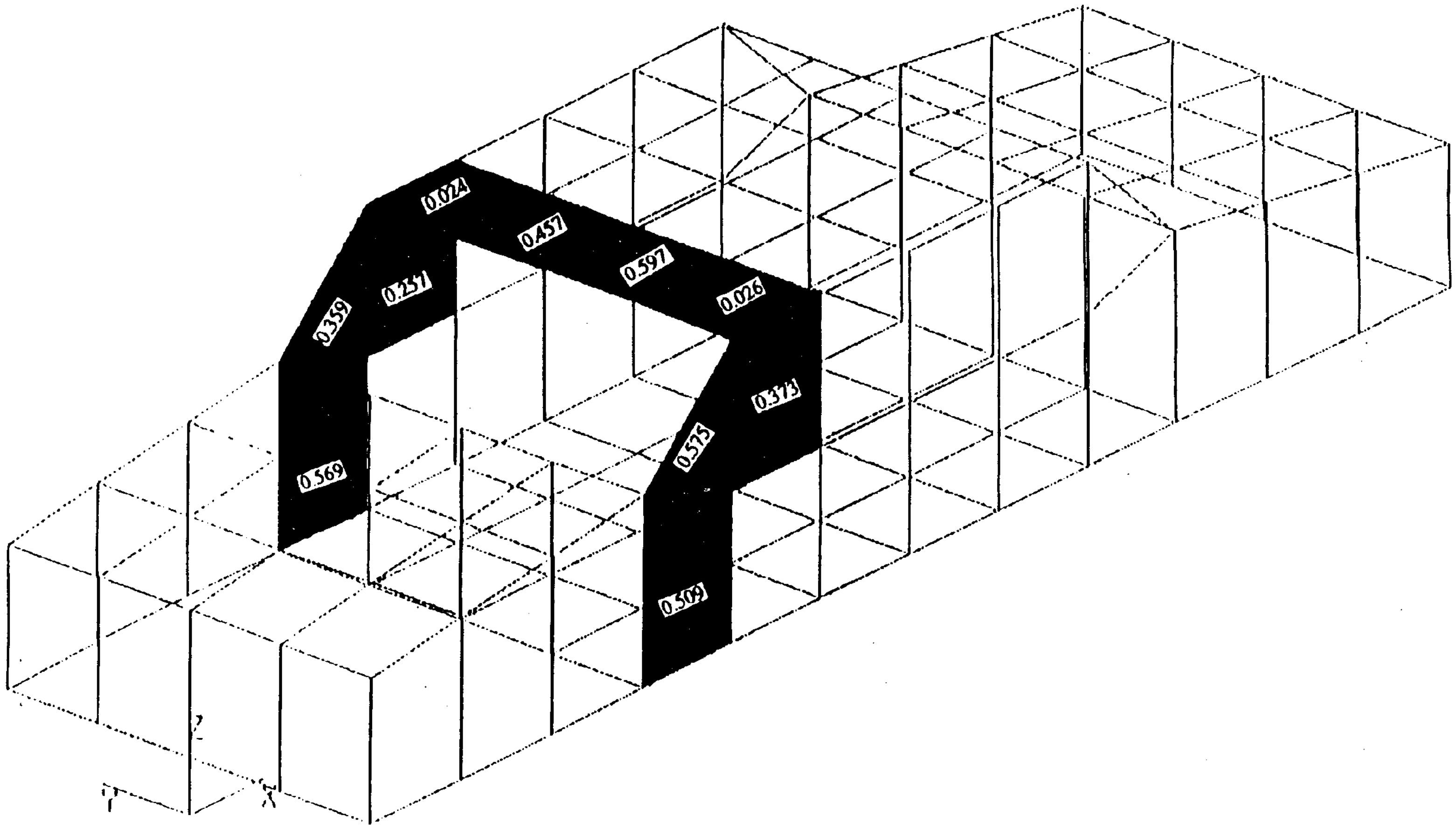

Figure 3 Elements (10) with fatigue life less than 1 block for baseline run.

Table 1 Design Sensitivity Analysis Summary Table

Basic Analysis	Boundary Conditions	Loads	Response Quantity (ϕ)	Design Variable (B_i)	DSA
Static deflection analysis	Constraint at shock towers	Bending and torsional loads	Max. deflection along rails and rockers (δ)	Panel thickness, cross-sectional areas, moment of inertias	$\frac{\partial \delta}{\partial B_i}$
Normal mode analysis	Free	N/A	Frequencies of first bending or torsional mode (f)	same	$\frac{\partial f}{\partial B_i}$
Inertia relief analysis for stresses	Support to eliminate rigid body motion	Unit load at critical attachment points	Maximum stress at critical area (σ)	same	$\frac{\partial \sigma}{\partial B_i}$
Inertia relief analysis for deflections	Support to eliminate rigid body motion	Unit load at critical attachment points	Total work done by unit loads at critical attachment points (U)	same	$\frac{\partial U}{\partial B_i}$

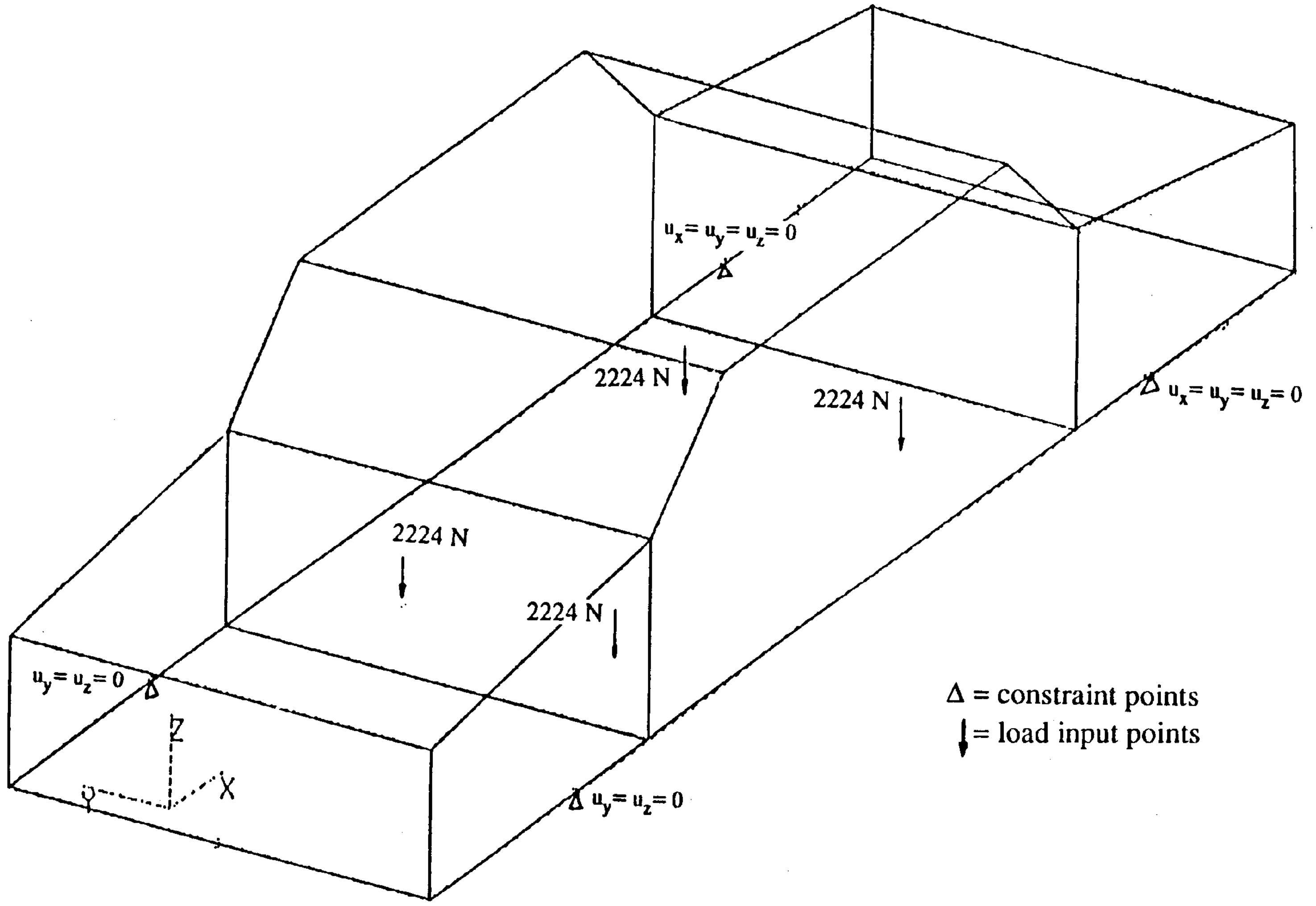

Figure 4 Static bending (deflection as a response) - loads are applied simultaneously at the four load input points

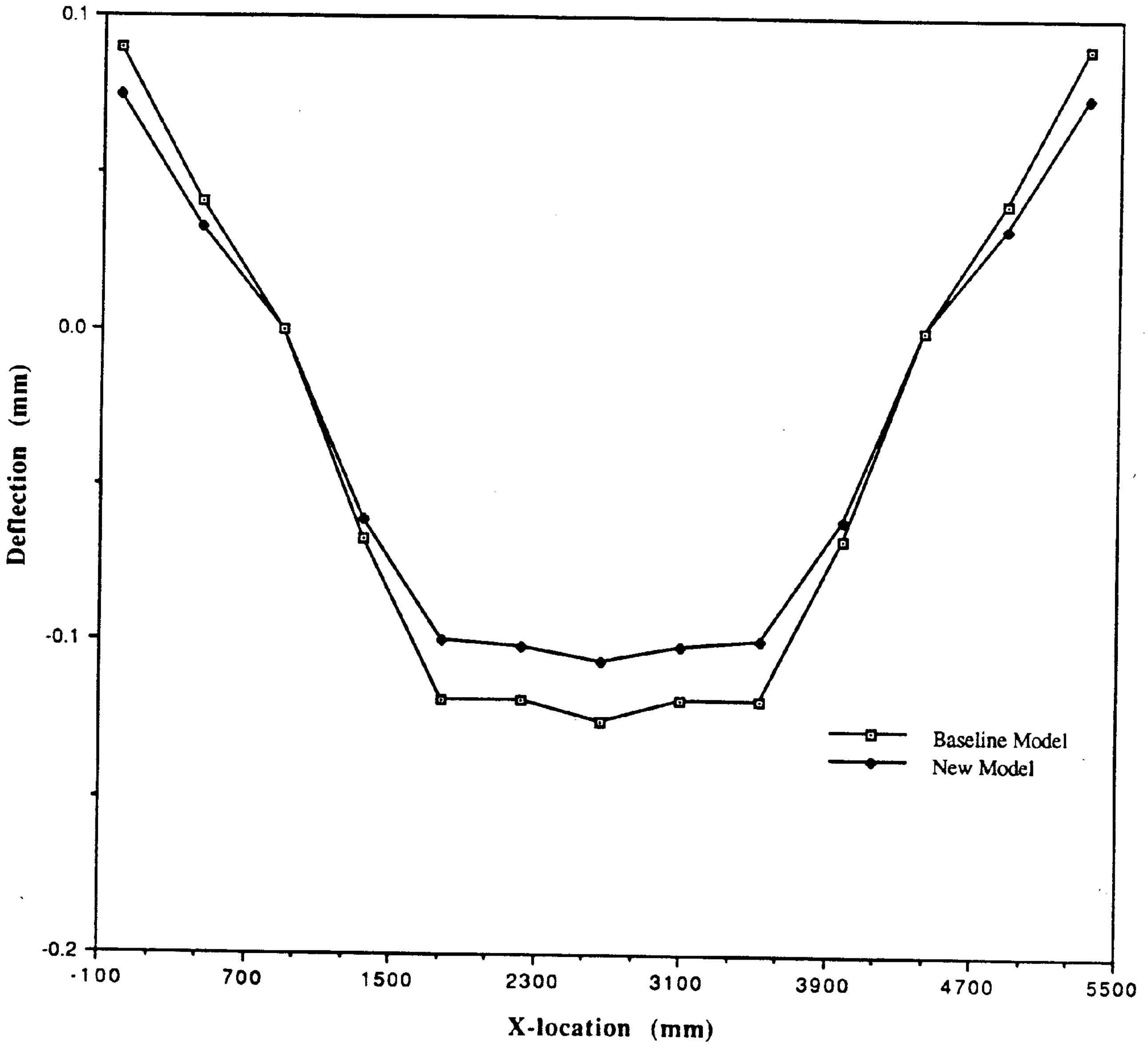

Figure 5 Bending deflections for two models (baseline and new model).

of inertia, torsional constant) of all beam elements and stiffness of all spring elements as design variables. It was found in the analysis results that the door thickness (0.8 mm) is the design variable which corresponds to the maximum DSC and thus would affect the overall bending stiffness the most.

A static bending analysis was conducted using a new door thickness (1.2 mm) and the deflections are compared with the baseline data in Figure 5. The figure shows that the new design resulted in a stiffer structure as compared to the baseline design.

Fatigue life was computed for the model with the new door thickness and the result is compared with that of the baseline in Table 2. It can be seen in this table that an 18.08% increase in bending stiffness resulted in a 245.83% increase in fatigue life. The elements with fatigue life less than 1 block are shown in Figure 6. The numbers refer to the fatigue life (in blocks) of the respective elements.

Static Torsional Analysis

To evaluate the effect of overall torsional stiffness on the fatigue life of the body structure, a traditional static torsional analysis was conducted using the loads and boundary conditions shown in Figure 7. The angle of twist along the length of the structure has been computed and is given in Figure 8 (the curve connecting points represented with squares).

To identify the design variables most influencing the overall torsional stiffness, a design sensitivity analysis similar to the one discussed above was conducted using the maximum deflection on one side of the front rail (deflections are anti-symmetric on both sides) as the response quantity. The analysis results showed that the door thickness is the design variable affecting the overall torsional stiffness the most.

A static torsional analysis was conducted using a 1.2 mm door thickness and the angle of twist is compared with that of the baseline in Figure 8. The new fatigue life resulting from this change is identical to the one computed in the bending analysis discussed above. The new torsional stiffness was computed and is given in Table 2. This table indicates that a 42.08% increase in torsional stiffness led to a 245.83% increase in fatigue life.

SENSITIVITY OF FREQUENCY OF MAJOR TORSIONAL MODE

To study the modal characteristics of the body structure, a normal mode analysis was conducted with a free boundary condition. It was found that the first elastic mode is an overall torsional mode at 1.106 Hz as shown in Figure 9. This low frequency in torsional mode is consistent with the low torsional stiffness indicated in the static torsional analysis.

To demonstrate the effect of the frequency of this torsional mode on fatigue life of the body structure, a design sensitivity analysis was conducted using this frequency as the response quantity and design variables identical to the ones employed in the static deflection analysis discussed earlier. The design variable corresponding to the maximum DSC identified in the analysis again implied that the door thickness is the design variable

Table 2 Listing of various values for baseline and the new model (after the design changes are made) for different analyses

Basic Analysis	Response Quantity	Baseline Response Value	Design Changes	New Response Values	% Change	Lowest Fatigue* Life (Baseline)	Lowest Fatigue Life (New)	% Change
Static deflection	Bending stiffness	70.59 KN/mm	door thickness (0.8 to 1.2 mm)	83.35 KN/mm	+ 18.08	0.024	0.083	+ 245.83
	Torsional stiffness	0.423 KNm/rad	door thickness (0.8 to 1.2 mm)	0.601 KNm/rad	+ 42.08	0.024	0.083	+ 245.83
Normal mode	Frequency	1.106 Hz	door thickness (0.8 to 1.2 mm)	1.279 Hz	+ 15.64	0.024	0.083	+ 245.83
Inertia relief analysis	Stress	0.241 MPa/N	roof thickness (0.8 to 1.2 mm)	0.023 MPa/N	- 15.77	0.024	0.036	+ 50.00
	Work done	0.175 Nmm	door thickness (0.8 to 1.2 mm)	0.116 Nmm	- 33.52	0.024	0.083	+ 245.83

* Fatigue Life in blocks (1 block = 100,000 miles on public roads).

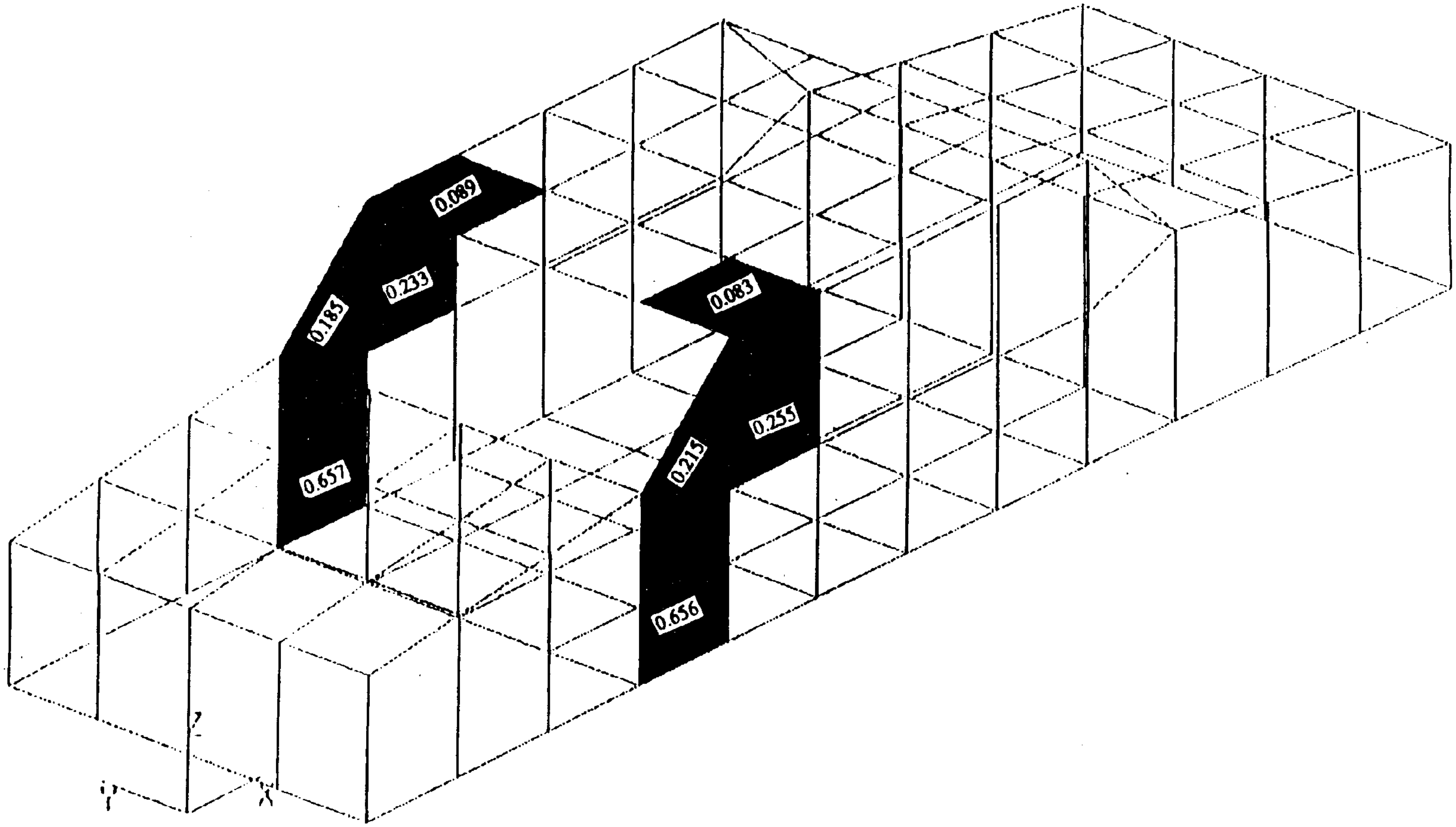

Figure 6 Elements (8) with fatigue life less than 1 block for the new model for which changes were made using the DSA. These changes were identical for the case of static bending& torsion analysis, modal analysis and inertia relief analysis using work done as a response.

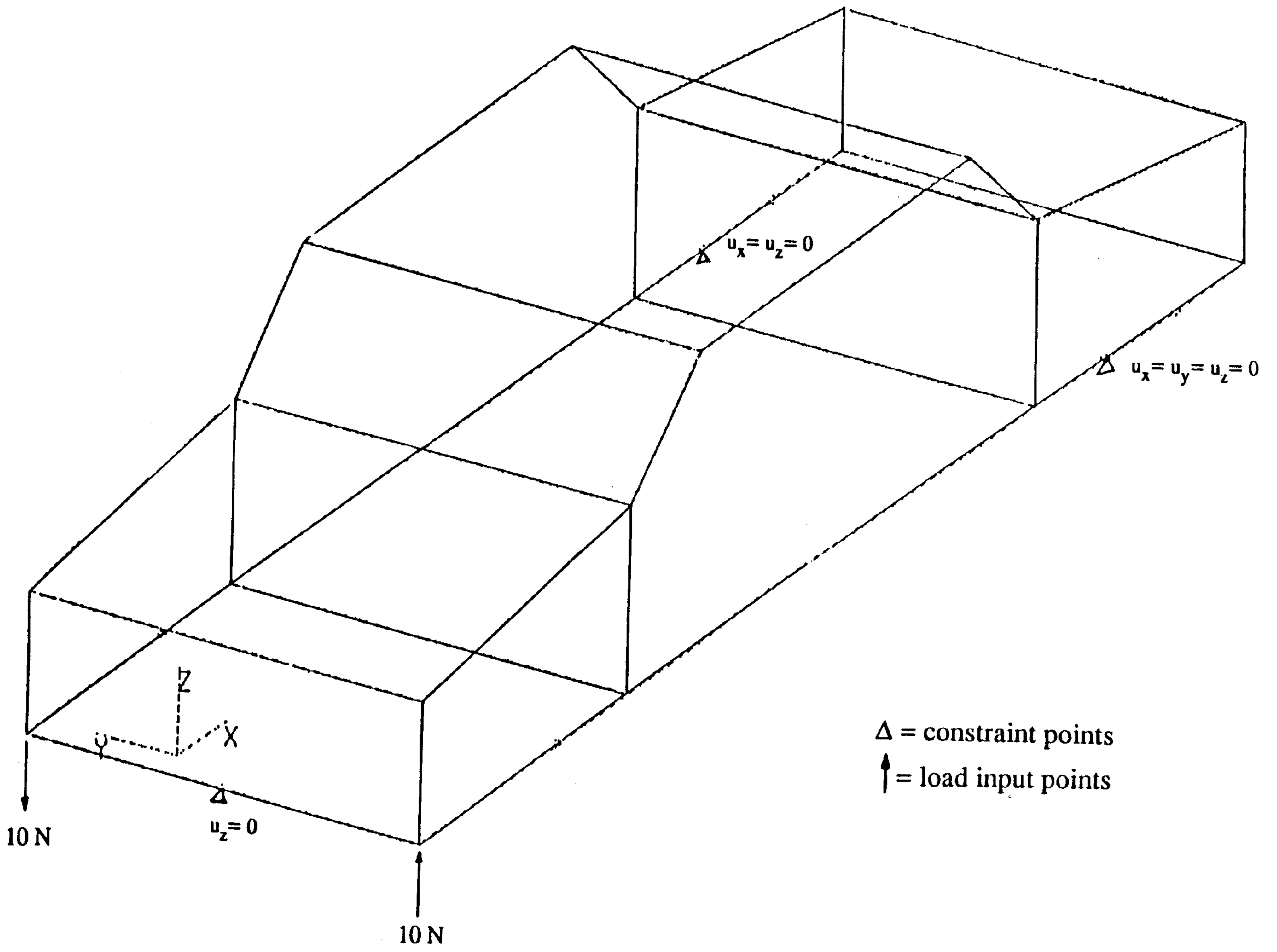

Figure 7 Static torsion (deflection as a response) - loads are applied simultaneously at the two load input points

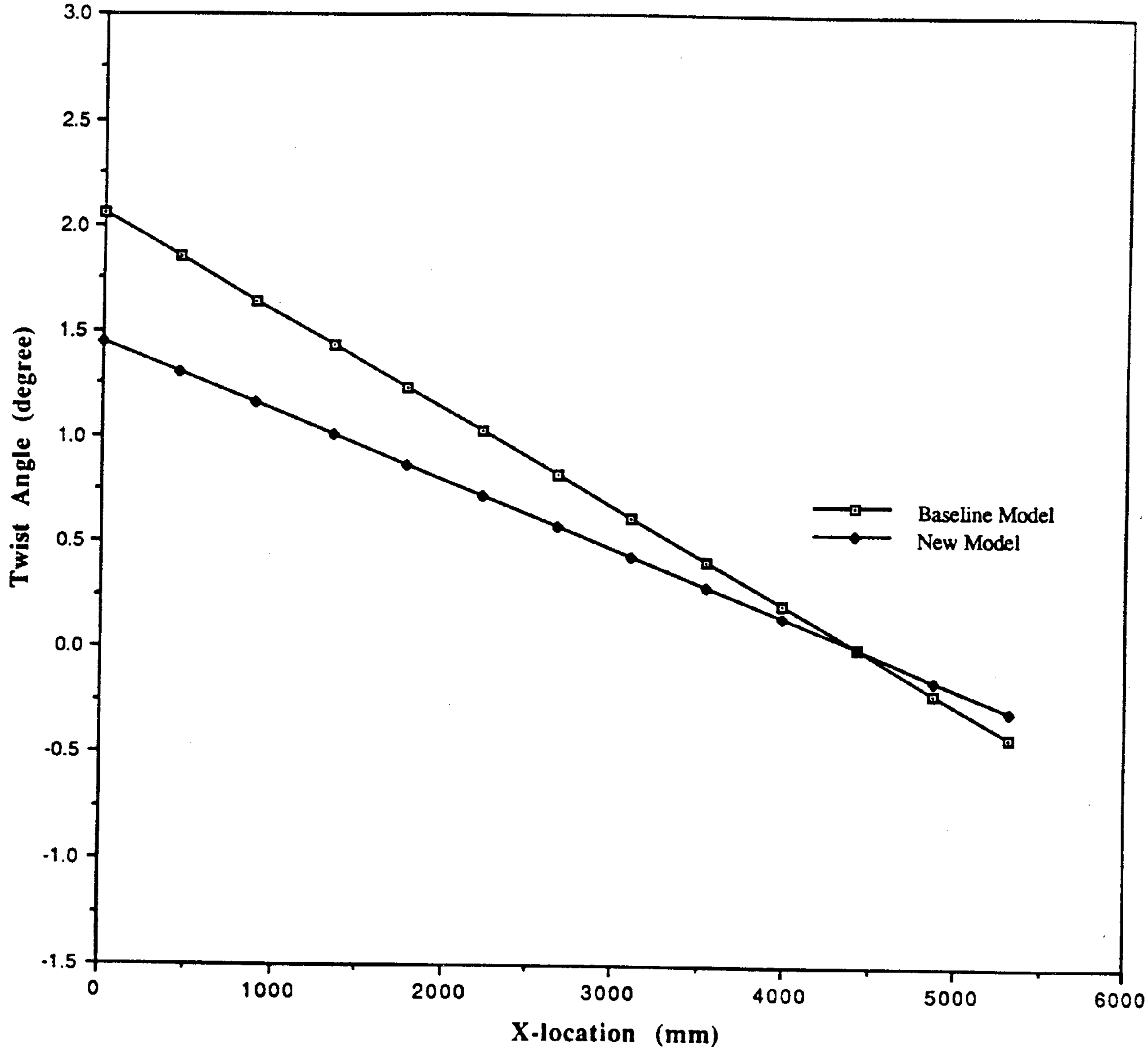

Figure 8 Twist angles under torsion for two models (baseline and new model).

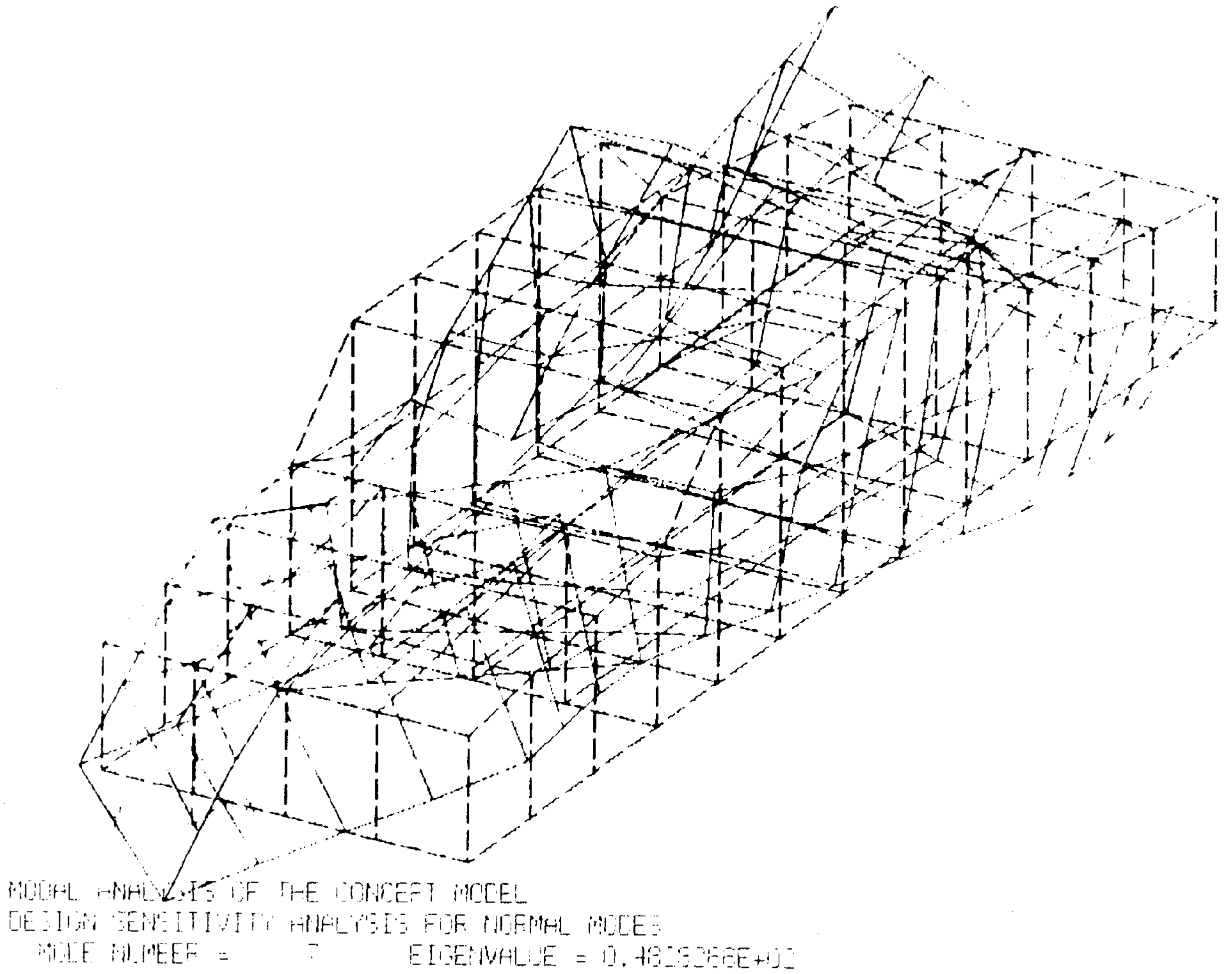

Figure 9 Deformed and undeformed shape for overall torsional mode (f = 1.106 Hz)

affecting the frequency of the overall torsional mode the most (same as in static deflection analysis).

A normal modes analysis was conducted based on a new door thickness of 1.2 mm and resulted in a new frequency 1.279 Hz for the torsional mode. This means that a 15.64% increase in torsional frequency yielded a 245.83% increase in fatigue life as shown in Table 2.

STRESS SENSITIVITY

To identify the most critical load, element fatigue life was computed for each load independently applied to the model. The lowest fatigue life in the model under each load is given in Table 3. This table indicates that the vertical load at the left front end resulted in the lowest fatigue life or maximum damage. It is thus logical to select this load as an input load in the DSA for the purpose of improving the fatigue life of the body structure.

Durability loads used in this paper are time history data and the resulting stresses under these loads are also time dependent. It would be very complicated to conduct DSA using a time-dependent stress as the response quantity. Instead, a constant load (1 newton) was applied to the model in the vertical direction at the left front end and the stress distribution was determined using the inertia relief analysis with the same constraint conditions shown in Figure 2. A design sensitivity analysis was then performed using the maximum Von Mises stress as the response quantity. The design variable most influencing this stress was identified to be the thickness (0.8 mm) of the roof panel.

A fatigue life analysis was conducted with a new thickness (1.2 mm) and the result is given in Table 2. This table indicates that a 15.77% decrease in the maximum Von Mises stress resulted in a 50.0% increase in fatigue life. The elements with fatigue life less than 1 block are shown in Figure 10. The numbers in the figure refer to the fatigue life (in blocks) of the respective elements.

SENSITIVITY OF WORK DONE BY UNIT LOAD APPLIED AT CRITICAL LOAD LOCATION

Stresses are related to the deformation field which in turn depends on the work done on the structure by external loads. This motivated a study to investigate how external work affects fatigue life.

The derivative of external work with respect to structural parameters is not directly available on MSC/NASTRAN. However, external work U is defined as :

$$U = (1/2)\, F\, \delta$$

where, F = applied load (here 1 N), and
δ = deflection at load application point in the direction of the load

The derivative of external work can be derived from the derivative of deflection at the load application point as follows:

Table 3 Fatigue life due to individual load

Location	Load	Fatigue Life in Blocks *
Left front end	Vertical	0.32
Right front end	Vertical	0.43
Left rear end	Lateral	2.79
Left rear end	Vertical	9.27
Right rear end	Lateral	478.31
Right rear end	Vertical	679.52

* 1 Block of life = Total test miles on a rough road durability course
= 100,000 miles on public roads.

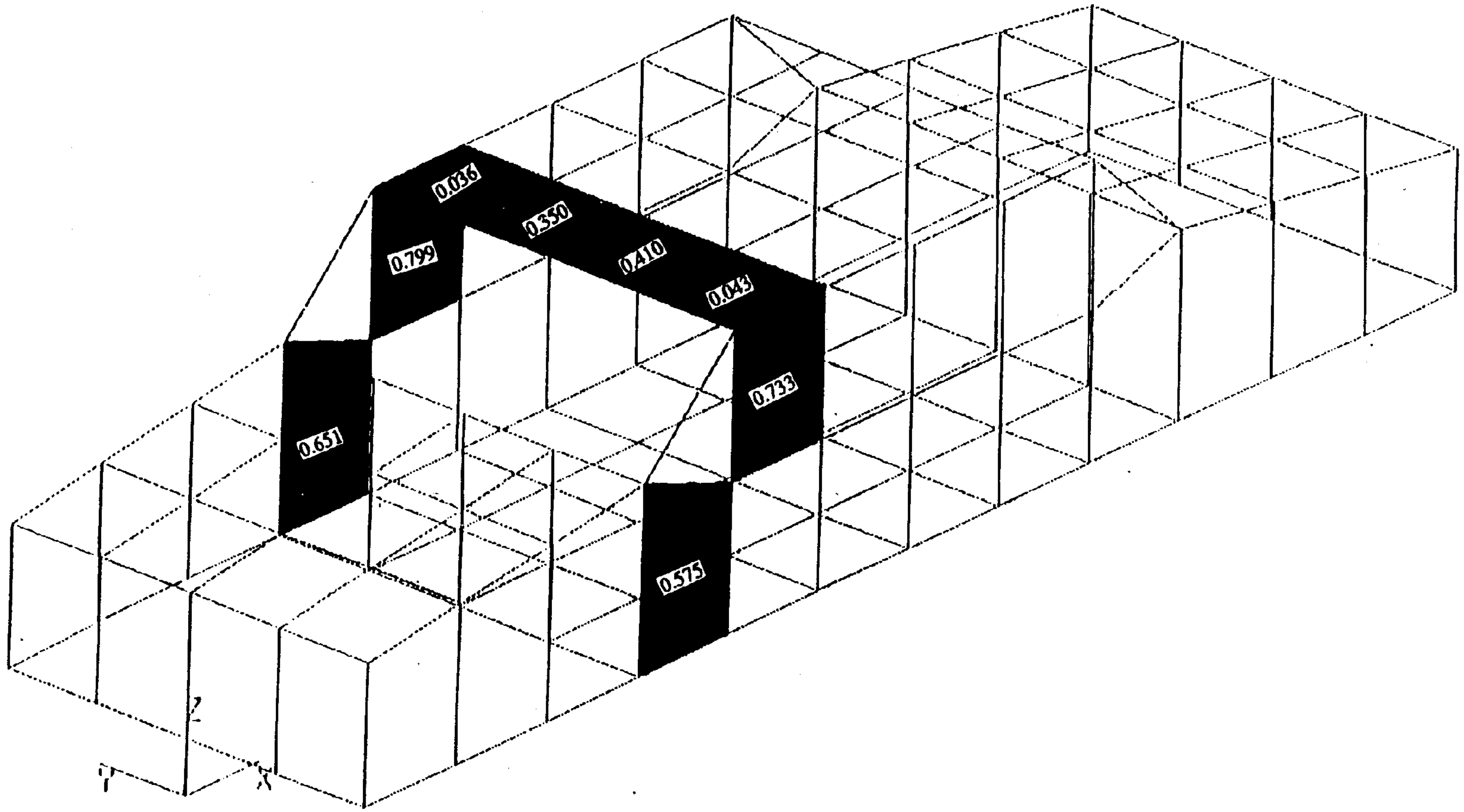

Figure 10 Elements (8) with fatigue life less than 1 block for the new model for which changes were made using the DSA. These changes correspond to the case of inertia relief analysis using stress as a response.

$$\frac{\partial U}{\partial B_i} = \frac{1}{2} F \frac{\partial \delta}{\partial B_i}$$

where, B_i is a structural parameter.

Therefore, a design sensitivity analysis was conducted using the deflection at the load application point as the response quantity. The design variable most influencing this deflection was identified to be the door thickness (0.8 mm).

A similar inertia relief analysis was conducted for the model with a new door thickness (1.2 mm) and the resulting external work done on the structure is shown in Table 2. It can be found in this table that a 33.52% decrease in external work resulted in a 245.83% increase in fatigue life.

The results given in the above two design sensitivity analyses indicate that the design variable identified in the DSA using the external work as the response quantity resulted in more improvement in fatigue life than that using the maximum Von Mises stress did. It also implies that the most effective design change for fatigue life improvement does not always occur in the highest stress area.

CONCLUSIONS

It has been demonstrated how the MSC/NASTRAN design sensitivity analysis capability can be employed to effectively identify design variables which lead to an improvement in fatigue life of a vehicle body structure when changed.

ACKNOWLEDGEMENTS

The authors wish to thank K. H. Huebner and N. J. Pritula of Ford Motor Company for their support during the course of this work.

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