Optimal Stiffness Design of Joint Structures of a Vehicle for Vibration

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Abstract

Idle shake vibration characteristics of a vehicle are mainly influenced not only by the stiffnesses of the beam type structures such as pillars and rockers, but also by the stiffnesses of the joint structures, at which several beam structures are jointed together. In the early design stage of the car body structure a simple FE model has been used, in which joints are modeled as linear springs to represent the stiffnesses of the joint structures. In this paper a new modeling technique for the joint structure is presented using an equivalent beam, instead of using a spring. The modeling technique proposed is utilized to design optimal joint structures that meet the required vibration performance of the total vehicle structure.

I. Introduction

The vehicle structure is an assemblage of sheet metal stampings reinforced by properly placed beams, columns, and corrugations[1]. Structurally, it is made of large, thin panels with typical gage of 0.7 to 1.2mm. Nonuniform beams, such as pillars, roof rails, and rockers, serve as a major load-carrying members. In these vehicle structures, joint parts at which several pillars are assembled together have the main influence on dynamic characteristics of the vehicle, such as natural frequency and vibration mode shape, etc[2][3]. The joint structure tends to introduce stiffness mismatches between thin-walled beams and shell-like structures that produce relatively soft local structural zones usuall referred to as local compliances[3]. Therefore, to consider the local compliance, the joint structure has been modeled by using torsional spring elements, of which spring constants represent the flexibility of the joint structure. But these spring elements have no mass, so it is impossible to optimize the design of the joint structure using the MSC/NASTRAN optimization routine, since the MSC/NASTRAN does not support the frequency optimization run for massless elements. However, the rotational stiffnesses of the joint structure has significant influence on static and dynamic characteristics of the whole vehicle structural system. To overcome this problem, a new modeling technique for the joint substructure is proposed in this paper. The local joint compliance is modeled using the equivalent beam element, of which section properties represents the equivalent local compliance[4][5]. Section properties of the equivalent beam element can be optimized to obtain the required joint stiffnesses, by the MSC/NASTRAN optimization run. Once the optimal joint stiffnesses are computed, the thickness of the thin panels forming the joint structure and the section shape of the joint must be determined.

Fig. 1 shows the joint analysis and design procedures proposed in this paper.

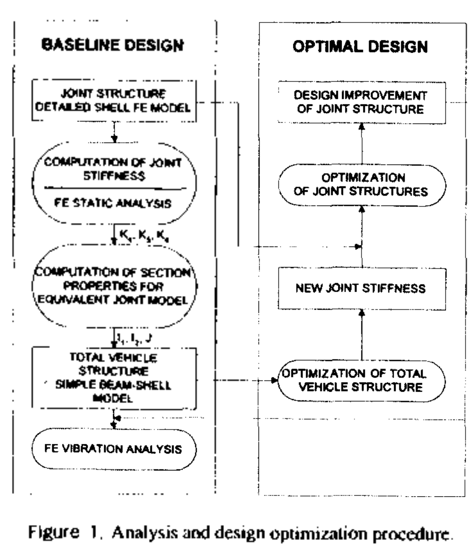

Figure 1. Analysis and design optimization procedure.

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Manuscript Received : April 18, 1998.

II. FINITE ELEMENT MODEL OF THE VEHICLE STRUCTURE AND ITS VIBRATION ANALYSIS

The vehicle body structure used in this study is a compact size passenger car. A simple finite element model that is often used in the early design stage of the vehicle structure is developed using the beam, shell, and spring elements. In the finite element model, beam type structures such as pillars and rockers are modeled using beam elements; the roof and floor panels are modeled as shell elements; the joint structures are modeled as linear spring elements. Fig. 2 and Fig. 3 show the typical low frequency vibration modes of the finite element vehicle model, which are obtained by conducting finite element vibration analysis.

Fig. 4 shows the side view of the typical vehicle structure. In this figure the solid lines represent the beam members that are modeled as beam elements, and the squares represent the joints that may be modeled as equivalent beam elements. Fig. 5 shows the two joint modeling techniques; one with the spring element and the other with the equivalent beam element for the B-pillar to roof joint.

III. COMPUTATION OF JOINT STIFFNESS

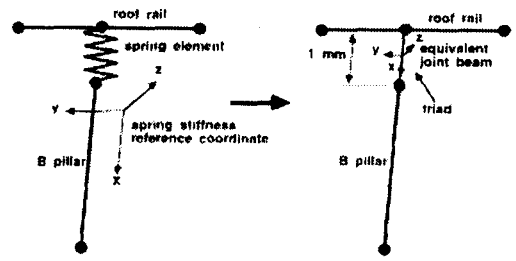


Figure 5. Equivalent beam modeling of B-pillar to roof joint.

To compute the joint stiffness, the joint structure is modeled using detailed finite shell elements, as shown in Fig. 6. Unit moments are applied at the tip of the B-pillar with both ends of roof rails fixed for the B-pillar to roof joint. Finite element static analysis gives the deformed rotation angle due to the applied unit moment, which represents the rotational stiffness of the joint. If the finite element model gives a reasonably accurate representation of the load-deflection behavior at the joint, then a local compliance spring or equivalent beam is chosen to satisfy joint substructure connectivity requirements.

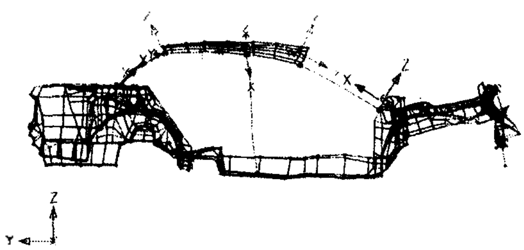


Figure 2. 1st torsional vibration mode.

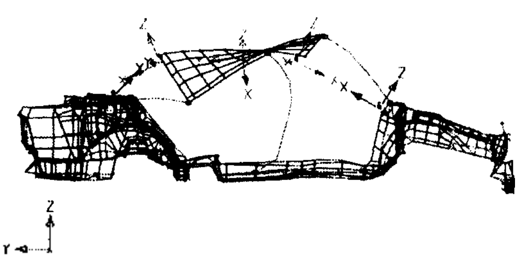


Figure 3. 2nd torsional vibration mode.

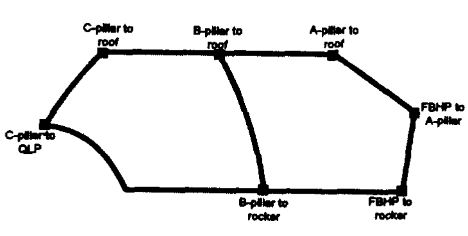
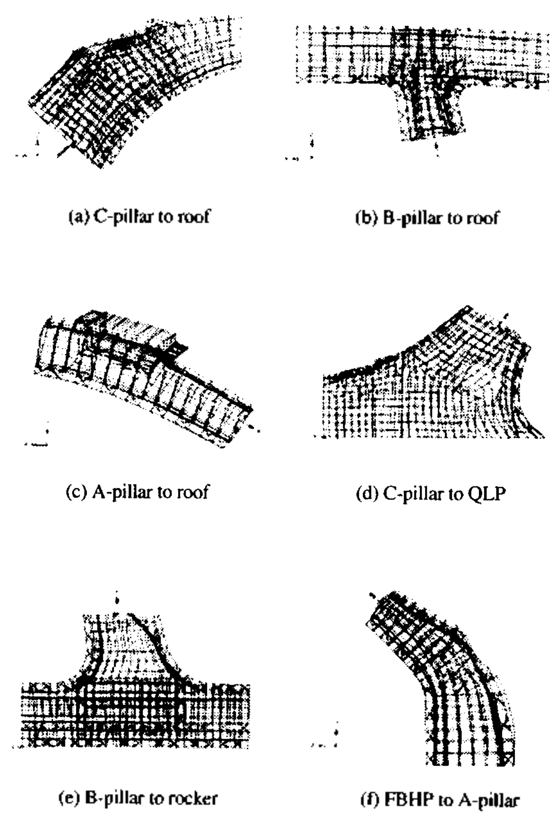


Figure 4. Position of joint equivalent beam element.



(a) C-pillar to roof (b) B-pillar to roof

(c) A-pillar to roof (d) C-pillar to QLP

(e) B-pillar to rocker (f) FBHP to A-pillar

Figure 6. Finite element models of

Using the simple stick model described in the previous section, the total vehicle structure is optimized using MSC/NASTRAN optimization run. Since the joint structures are modeled as equivalent beam elements as shown in Fig. 5, if their section properties such as I_y, I_z, J are chosen as design variables, their optimal values can be obtained to meet the frequency requirements for the given total vehicle structure. Table 1 shows the design change guideline to obtain the optimal joint stiffness. Once the optimal section properties of the equivalent beams, the corresponding joint stiffness can be determined using Eq. (1).

$$K_x = \frac{GJ}{L}, \quad K_y = \frac{EI_y}{L}, \quad K_z = \frac{EI_z}{L} \tag{1}$$

Where, I_y is the second moment of inertia of the area about Y direction; I_z is the second moment of inertia of the area about Z direction; J is the torsional constant about X direction; E is the elastic modulus; G is the shear modulus; and L is the length of the equivalent beam element. Using these equations, the desired rotational stiffness of each joint can be calculated. Direction of design change for the joint structures can be guided using the Fig. 7. For example, while the rotational stiffness of C-pillar to roof joint for Z direction may be reduced, the rotational stiffness of the FBHP(Front Body Hinge Pillar) to rocker joint for X direction needs to be increased.

Table 1. Percentage change of design variables (unit : %).

	I_y	I_z	J
A-pillar to roof	+25.6	-34.9	+39.9
B-pillar to roof	+30.0	+7.0	+7.0
C-pillar to roof	+16.9	-66.9	-46.0
FBHP to A-pillar	-63.6	-20.8	-16.1
FBHP to rocker	-0.2	+8.9	-3.1
B-pillar to rocker	+20.6	-1.1	-53.4
C-pillar to QLP	-63.8	+0.1	-0.6

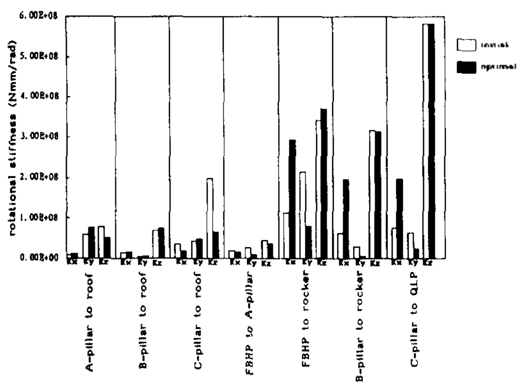


Figure 7. Comparison of joint stiffnesses.

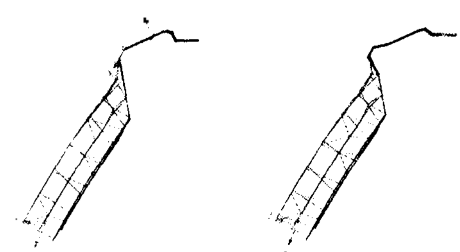
Ⅳ. OPTIMAL DESIGN OF THE JOINT STRUCTURE

This section explains the method used to design the optimal joint structure satisfying the required stiffness that is determined in the previous section. In this study, the optimal shape of the B-pillar to roof joint is searched. Fig. 7 implies that the Y directional rotational stiffness of the B-pillar to roof joint needs to be increased, while the other directional stiffnesses may be decreased. Therefore, the rotational stiffness about Y direction must be changed through optimal design, and the result of joint parameter study shows that design parameters influencing the rotational stiffness about Y direction are the thickness and the shape of the reinforcement panel[6].

Therefore the thickness and nodal coordinates of reinforcement panel are established as design parameters, where the reinforcement panel may move to between the outer panel and inner panel of the roof rail part, which may change the shape design to satisfy the required joint stiffness. The objective function is set up for minimizing the mass, and the constraints are set up for the required rotational stiffness of the joint structure. Using the optimization module of the MSC/NASTRAN[7], the optimal thickness and shape can be obtained as shown in Table 2 and Fig. 8.

Table 2. Design change of B-pillar to roof shell joint (unit : %).

stiffness	K_x	7.48
	K_y	34.15
	K_z	7.18
design variables	thickness	33.33
	shape	refer to Fig. 8
objective function	mass	7.72



(a) initial shape (b) optimal shape

Figure 8. Design change of B-pillar to roof joint (Reinforcement panel).

Ⅴ. CONCLUSION

In this study, a systematic optimal design method for the vehicle structure was presented. An equivalent beam modeling technique for the joint compliance was applied to the finite element model for the vehicle structures, instead of using the conventional spring element, to reflect the joint compliance. Using the equivalent beam elements for the joint compliances, optimal joint stiffnesses can be determined, and corresponding optimal joint structures can be designed by changing the panel thickness and the shape of the joint structure. The methodology presented in this paper can be used as a design guideline tool for the optimal design of the vehicle structure including joints to meet the frequency target in the early design stage.

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