Proceedings of ICAD2004
The Third International Conference on Axiomatic Design
Seoul – June 21-24, 2004
ICAD-2004-07

AXIOMATIC APPROACH FOR STRESS RELAXATION IN AUTOMOTIVE MULTI-SHELL-STRUCTURES

Hyungyil Lee[1]
hylee@sogang.ac.kr
Department of Mechanical Engineering
Sogang University, Korea

Gyung-Jin Park[2]
gjpark@hanyang.ac.kr
Department of Mechanical Engineering
Hanyang University, Korea

ABSTRACT

Recent attempt to enhance safety against collision has reshaped the simple single-shell structure into the integrated multi-shell structure. Moreover, due to various regulations continuously tightened for environment, weight reduction of automobile becomes an increasingly important issue. Weight reduction is mainly accomplished by better redesign, adoption of lighter materials, and small-sizing of auto (parts). Focusing on the local redesign among three, we attempt to determine the thickness of each subpart-shell of an integrated multi-shell structure by axiomatic design approach. Based on the finite element stress calculations, we relieve the stress of a box type subframe by varying the thickness of each subpart-shell. The redesign method successfully brings both a preset amount of stress relaxation and weight reduction. This kind of axiomatic approach can be extended to the other multi-shell structures.

Keywords: Axiomatic design, Multi-shell-structure, Subframe, Finite element method, Stress relaxation

1 INTRODUCTION

In Europe and USA, safety regulations are constantly strengthened, which leads automotive manufacturers to make various endeavors for safety of the passenger (Lam, 2003). Consequently, the technology level for the design of a safe auto-body becomes a main measure of international competitiveness of an auto manufacturer. This trend has changed some front parts of an auto-body from a single simple shell structure to an integrated multi-shell structure (Prange and Schneider, 2001; Shin *et al*., 2002).

When the multi-shell structure is utilized, weight reduction can be easily realized because patches can be added to different locations of the structure. Structural optimization has been popularized for weight reduction (Rangachargulu and Done, 1979; Schmit, 1981; Vanderplaats, 1982; Hansen and Vanderplaats, 1990). Structural optimization is classified into size, shape and topology designs according to the characteristics of the design variables (Haftka and Gurdal, 1993; Min *et al*., 1999; Barbarosie, 2003). They are quite efficient and sophisticated since they are mathematically well defined. However, the design problem should be defined to fit into the paradigm, which each technology pursues. Therefore, some other optimization methods such as fully stressed design (Hinton and Sienz, 1995; Mueller *et al*., 2002) and growth strain method (Han and Lim, 2002) have been developed as variances for mathematical optimization although they do not generate a mathematical optimum. These methods are fairly good in that they give moderate design solutions.

Design variables must be defined, when one of the above optimization technologies is adopted. Some applications of optimization have been performed for the automobile structures (Botkin, 2002; Shin *et al*., 2002). The researches show that the performance of a structure is improved. However, the design is carried out in a restricted sense to fit into the optimization paradigm. In practice, designers for the subframe want to define design variables in various manners and optimization is quite difficult and expensive with this freedom. Therefore, engineers design the subframe by using their intuition obtained from their experiences and many trial-and-error types of FE analyses although the application of optimization is not impossible. Design for weight reduction of the component often induces overly-stressed weak regions. Entire redesign for stress-relieving of the weak region is, however, far from practical. If methods for supplementing the weak region of as-designed part are framed, a substantial amount of time and cost for redesign can be saved. Consequently, patches are often added to various weak locations of subframe.

In this work, an automobile subframe is analyzed and redesigned. Finite element (FE) method is adopted for the analysis. The analysis results should be well incorporated into the design process. First, structural and load characteristics of the #-type subframe model are analyzed via FE stress calculations. We then present the axiomatic design approach to determine the optimal thickness of each subpart-shell in the integrated #-type subframe. The Independence Axiom of axiomatic design is adopted in this procedure (Suh, 1990; 2001). Functional requirements (FRs) are defined to relieve maximum stresses, and design parameters (DPs) are defined by a set of thickness of each panel. A design matrix is established from the investigation of the FR-DP relation. It is found that the design matrix is a decoupled one. Therefore, the Independence Axiom is satisfied and the design sequence is determined by the design matrix. At each step of the sequence, the thickness of each panel is optimized for weight reduction.

2 FINITE ELEMENT STRESS ANALYSIS OF MULTI-SHELL STRUCTURES

2.1 MODELING OF MULTI-SHELL STRUCTURES

The boundary conditions for the #-type subframe having higher crash resistance are shown in Fig. 1. The front (B_s) and rear (C_s) parts of the #-type subframe are connected to the main body. The middle (A_s) and end (G_s) parts are connected to the control arm. The integrated #-type subframe consists of No. 1, No. 2, (left and right) upper members, center mounting brackets and G-point brackets, as shown in Fig. 2. No. 1 connects with the front part of auto-body and spot welds join its upper and lower panels. No. 2, left and right upper members respectively play the roles of cross and center members of prior T-type subframe. Center mounting bracket and G-point bracket connects with suspension unit and lower control arm respectively so as to transfer driving load to auto-body. In short, integrated #-type subframe for medium class passenger car is also a welding assembly of 8 subparts. Therefore, for more credible modeling, all subparts are modeled one by one. I-DEAS (1996) preprocessing program is used for finite element (FE) modeling of subframe. FE models of subparts are joined at weld using rigid beam element (MPC: ABAQUS User's Manual, 2001) for proper modeling of seam weld connecting each part continuously. Final FE model formed through these processes is shown in Fig. 3. The whole FE model consists of about 34000 S4R elements and about 36700 nodes. We then perform linear elastic analyses with the material SAPH41P, hot rolled high strength steel plate for automobile structure (Young's modulus E = 200 GPa, Poisson's ratio ν = 0.3, yield strength σ_y = 277 MPa).

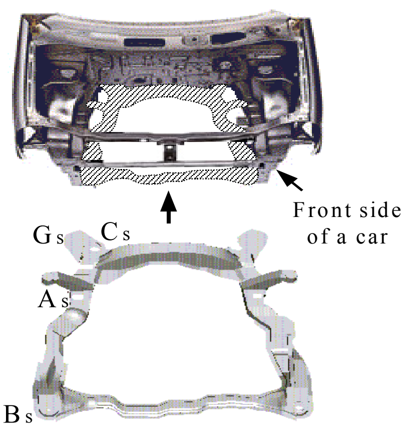


Fig. 1 *Position and shape of subframe.*

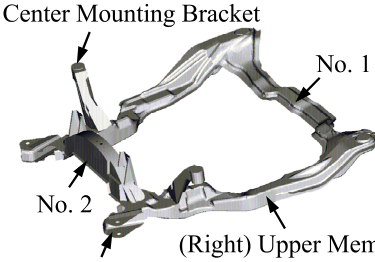


Fig. 2 *Shape of eight parts of subframe*

2.2 FE STRESS SOLUTIONS OF STRUCTURES

Boundary and load conditions applied to finite element analyses of subframe are as follows. Since B_s and C_s parts are connected to the main body, relative displacements to body are zeroes. Therefore, we fixed the *x, y, z*-directional displacements and rotations of the fixture parts used for connection. We also apply some boundary conditions proposed by car manufacturers to D_s parts connected to suspension unit. The parts A_s and G_s are installed to lower control arm, and loading conditions at those parts depend on the driving conditions of front wheels. Our preliminary analyses revealed that, among various driving conditions, the sudden brake generates the most severe loadings on the parts A and G. Sudden brake condition is thus selected as the FE loading conditions for the subframe. Those boundary and loading conditions under sudden brake are summarized in Fig. 3 and Table 1. Finite element stress solutions in the sudden brake loading conditions are shown in Fig. 4. We observe that the stress concentrates on the left and right upper loading point connected to lower control arm. Maximum stress (314 MPa) is 1.13 times the yield strength of SAPH38P (277 MPa). These stress solutions clearly suggest that certain measures need to be taken for stress relaxation in pre-designed subframe. To relax stress and reduce weight, we present the axiomatic design approach for determination of optimal thickness of each subpart-shell.

Table 1 Boundary & loading conditions of parts at sudden brake (see Fig. 3)

Parts	Boundary conditions	Loading conditions (kgf)					
Sub-frame	B_s, C_s, D_s (B_s', C_s', D_s')	A_s (A_s')			G_s (G_s')		
		F_x	F_y	F_z	F_x	F_y	F_z
	All fixed	-640	1191	-87	-220	-977	-81

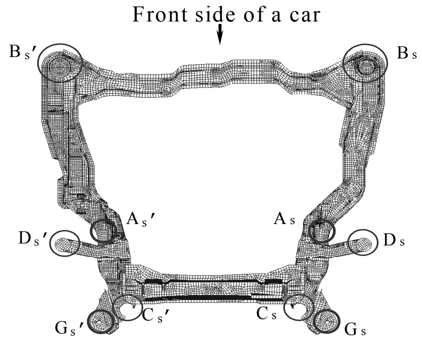


Fig. 3 *FE models and BCs of subframe.*

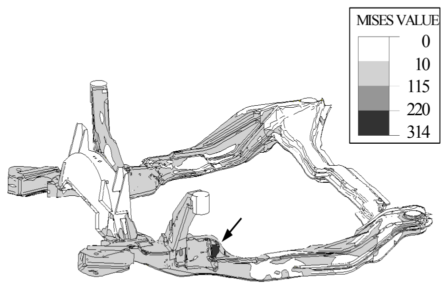


Fig. 4 *Equivalent stress distribution at sudden brake.*

3 DESIGN ENHANCEMENT BY AXIOMATIC APPROACH

3.1 A BRIEF OF AXIOMATIC DESIGN

The premise of axiomatic design is that there exists a fundamental set of principles that determines good design practice. Two axioms are proposed by noting the common phenomena shared by all cases. The first independence axiom states that independence of functional requirements characterizing functional needs must be maintained during the design process. The second information axiom states that, among all the designs that satisfy the first axiom, the one with minimum information content is the best. The minimum information content means that the probability for success is the highest. From these two axioms, many theorems and corollaries are derived (Suh, 1990).

In the (axiomatic) design world, there are 4 domains as shown in Fig. 5: client, function, physics, and process. A set in the left domain is satisfied by choosing a proper set in the right domain. Customer requirements (CRs) are a set of ultimate objects of design. Functional requirements (FRs) form a minimum set of independent requirements to achieve CRs. FRs describe the design objects under constraints. Constraints represent the bounds on an acceptable solution. By definition, a constraint is different from FRs in that it needs not to be independent of other constraints and FRs. Design parameters (DPs) are a set of physical embodiments for fulfilling FRs. Process Variables (PVs) are manufacturing methods for realizing DPs. Design process is an interdomain mapping operation. The design equation for the product design is expressed as

$$\{FR\} = [DM]\{DP\} \leftrightarrow FR_i = \sum_j DM_{ij} DP_j \qquad (1)$$

Here {FR} is the functional requirement vector and {DP} is the design parameter vector, and [DM] is the design matrix. To satisfy independence axiom, design matrix should be diagonal, or triangular. If [DM] is diagonal, a FR is satisfied independently by a DP. This design is defined as an uncoupled design. If [DM] is (inverse) triangular, independence of FRs can be assured by adjusting DPs in a particular order. It is called a decoupled design.

Designers can propose several designs, which satisfy first axiom for a given set of FRs. Information axiom allows us to measure the design quality, thereby to select the best design. The information content is directly related to the probability of achieving the FR. Probability for success increases as information quantity for accomplishing the FR decreases. Conversely, infinite information is necessary if the probability for success approaches to zero. In short, information axiom means that the design with the minimum information content is the best one. In the next section, we apply this axiomatic approach to stress relaxation and weight reduction of #-type subframe.

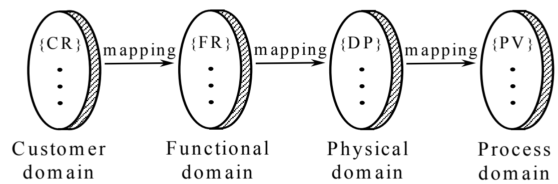


Fig. 5 *Four domains of the design world.*

3.2 DESIGN OF SUB-FRAME THROUGH AXIOMATIC APPROACH

To relax stress of #-type subframe, FRs can be defined as

FR1 = To relieve maximum stress of No. 1
FR2 = To relieve maximum stress of No. 2
FR3 = To relieve maximum stress of center mounting bracket (CMB)
FR4 = To relieve maximum stress of G-point bracket (GPB)
FR5 = To relieve maximum stress of upper member (UM)

It is observed that stress distributions in left and right sides of G-point bracket are at the same level, since the shapes of left and right sides of G-point are alike. Therefore, only "one" FR was allocated to the maximum stress for G-point. The same were center mounting bracket and upper member. At the next stage of axiomatic approach, design parameters (DPs) satisfying those five FRs should be defined, and then optimum DPs satisfying given constraints should be determined. For this, we can first consider the shape of each subpart as a DP. However, shape optimization, being more complicated than parameter optimization, is not yet in the practical stage in spite of its well-established theoretical basis. Concretely, moving boundary condition due to shape change makes its application quite difficult. Algorithms for shape optimization are amply found in the literature, but the reliability, efficiency and accuracy of them seem to need further study (Kwak, 1994). To overcome the difficulty of shape optimization, Kims (1994) defined the shape of engine-mount with several shape parameters. They then determined the parameters so as to minimize the difference between design-specified stiffness and stiffness of shape defined by a set of shape parameters. Kwak *et* *al.* (1995) selected the patch thickness of auto-hood made of sheet molding compound (SMC) as a DP. They then designed a lightest SMC hood with the same stiffness of steel hood. As observed in those studies, defining subpart shape as a design parameter is inappropriate in terms of both information axiom and practical point of view. In this study, to satisfy FRs without changing subframe shape, we therefore define simple DPs as

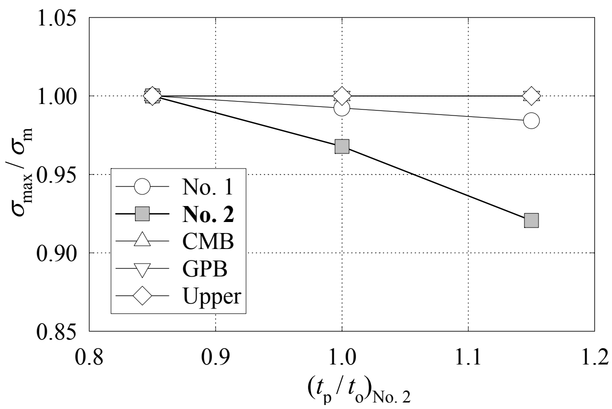
DP1 = panel thickness of No. 1
DP2 = panel thickness of No. 2
DP3 = panel thickness of center mounting bracket (CMB)
DP4 = panel thickness of G-point bracket (GPB)
DP5 = panel thickness of upper member (UM)

The design includes following three constraints.

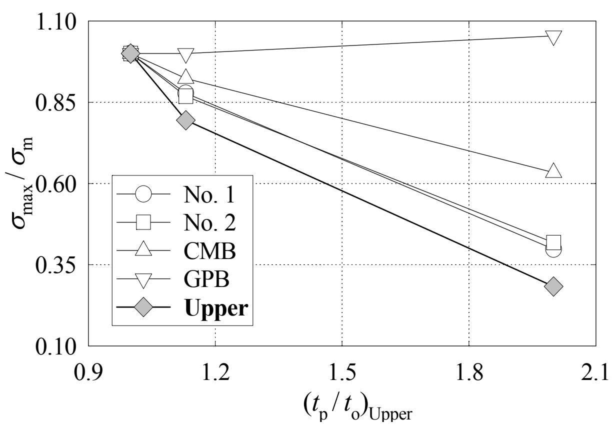
Ct1: maximum stress is less than 80% of panel yield strength
Ct2: $DP_i \geq 1$ mm for formability
Ct3: weight of subframe is less than the as-pre-designed

Table 2 Thickness and maximum stress of each subframe model

Parts	Model subframe		Optimized model	
	Original thickness (mm)	Maximum stress (MPa)	Thickness (mm)	Maximum stress (MPa)
No. 1	2.3	64	1.0	59
No. 2	2.0	71	1.0	43
CMB	2.0	146	1.2	216
GPB	2.6	129	2.0	224
UM	2.3	314	2.8	224



(a)



(b)

Fig. 6 *Variation of maximum stress in each part for changes of thickness of* (a) *No. 2* (b) *UM.*

We first investigate the variation of stress at each subpart, when thickness of a specific part changes while thicknesses of the other subparts are fixed as the pre-designed values in Table 2. Figures 6a-b are the two typical cases among those investigated. Figure 6a shows the maximum stress variations of each subpart when thickness of only No. 2 changes. Figure 6b shows the maximum stress variations of each subpart when thickness of only UM changes. Changed thickness t_p of a specific subpart is normalized with the original thickness t_o of that specific subpart. Maximum stress σ_{max} of each subpart is normalized with a maximum stress σ_m of that subpart (No. 1, No. 2, CMB, GPB, UM) obtained when thickness of a specific part (No. 2 in Fig. 6a; UM in Fig. 6b) has the minimum value. We observe in Fig. 6a that for the thickness change of No. 2, only No. 2 itself shows notable variation in σ_{max}, while other subparts show insignificant variations. In short, thickness of No. 2 affects only σ_{max} of No. 2. The thickness of No. 1 was also observed to affect only σ_{max} of No. 1. The same were CMB and GPB. On the other hand, when thickness of UM increases, σ_{max} of all subparts decreases except GPB as shown in Fig. 6b. This is because UM plays the role of translating driving loads from lower control arm to other subparts. GPB, however, receives loads directly from lower control arm, which results in slight variation of σ_{max} in GPB even with thickness change of UM. Design matrix of Eq. (1) based on these observations comes to an inverse triangular matrix like Eq. (2).

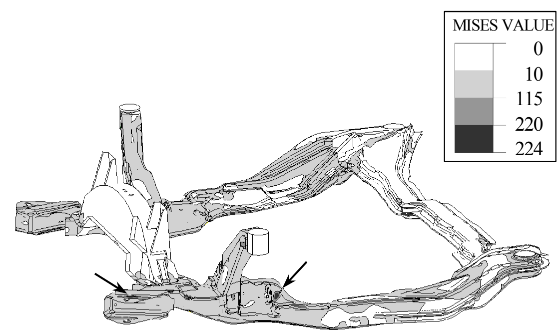


Fig. 7 *Equivalent stress distribution in the subframe model with optimized part thicknesses at sudden brake.*

$$\begin{pmatrix} FR1 \\ FR2 \\ FR3 \\ FR4 \\ FR5 \end{pmatrix} = \begin{bmatrix} X & O & O & O & X \\ O & X & O & O & X \\ O & O & X & O & X \\ O & O & O & X & O \\ O & O & O & O & X \end{bmatrix} \begin{pmatrix} DP1 \\ DP2 \\ DP3 \\ DP4 \\ DP5 \end{pmatrix} \quad (2)$$

Here X and O mean that DP do and do not affect FR, respectively. In a rather complicated design with many FRs and DPs such as Eq. (2), investigation of sensitivity of FR with respect to each DP is the most essential work. This is because design matrix sets the sequence for determining the optimum values of design parameters. To determine DPs satisfying FRs in a design equation having inverse triangular matrix like Eq. (2), each DP should be defined one by one in reverse order, that is, from DP5 to DP1. We determine optimum panel thickness satisfying Ct1 via simpler 2nd order equation as follows.

$$C_o + C_1 t + C_2 t^2 = \sigma_{\max} / \sigma_y \tag{3}$$

Here $t = t_p / t_o$ (= ratio of changed thickness to original thickness of a specific subpart), σ_{max} is maximum stress at the specific subpart, σ_y is yield strength, and C_o, C_1, C_2 are unknown constants. To determine these constants in Eq. (3), three σ_{max} values for 3 thickness values of the specific subpart are needed. Values of σ_{max} are obtained by FE analyses for 3 different values of thickness of that subpart. By substituting 3 values of thickness and corresponding 3 values of σ_{max} into Eq. (3), 3 simultaneous equations for unknown constants C_o, C_1, C_2 are obtained. Solutions to the simultaneous equations obtained by LU decomposition method are (C_o, C_1, C_2) = (3.24, -2.85, 0.71). When maximum stress of UM is 80% of yield strength, Eq. (3) then gives $t_p \,|_{Upper}$ = 1.23 $t_o \,|_{Upper}$ = 1.23x2.3 = 2.8 mm. FE analysis with $t_p \,|_{Upper}$ = 2.8 mm gives maximum stress of subframe as 224 MPa. This differs only 1% from 222 MPa (= 80% of yield strength of SAPH38P, which validates the approach by Eq. (3). Determining DP4-DP1 in the same manner, we obtain the thickness of each subpart as summarized in Table 3. When thickness of a specific subpart was varied with thicknesses of other subparts fixed, it was observed that maximum stress at No. 1 and No. 2 were always much lower than yield strength. Therefore, minimum thickness (1mm) allowed for formability is selected as optimum thickness of No. 1 and No. 2. Figure 7 shows the equivalent stress distribution by FE analysis at a sudden brake condition for #-type subframe model consisting of optimum thicknesses of subparts. Table 2 also shows the flatted stress distribution that is an indication of enhanced structural efficiency.

Table 3 Weight and maximum stress for each sub-frame model

Model type	Weight (kg)	Maximum stress (MPa)	Ratio of stress (σ_{max}/σ_y)
Model subframe	24.15	314	1.13
Subframe model with optimized part thicknesses	20.23	224	0.81

4 SUMMARY

Table 3 compares the weight and maximum stress of as-pre-designed #-type subframe and the one with optimized subpart thicknesses. The subframe with optimized subpart thicknesses accurately decreases maximum stress to the preset value (= 0.8 σ_y). Stress ratio is the ratio of maximum stress to yield strength of SAPH38P (277 MPa). It is noteworthy that enhanced subframe model also gives the weight reduction effect of about 3.92 kg (16%). Manufacturer had relaxed the maximum stress to the same level (= 0.8 σ_y) with an experience-based patch of 3.89 kg. Compared with this patched one, enhanced subframe model gives a significant weight reduction of 7.81 kg (28%). The stress relaxation methods presented in this work can be applied to the other multi-shell structures.

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