

Agile Modeling of Component Connections for Simulation and Design of Complex Vehicle Structures

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ABSTRACT

Many efficient modeling methods have been developed for analyzing the effects of component-level variations and uncertainties on the system-level response of complex structures. However, relatively little work has addressed the efficient or agile modeling of variations in the connections between components. Such a capability would be useful for simulation (e.g., performing reliability analysis accounting for spot welding variations) and design (e.g., determining fastener locations for up-armor kits) of commercial and military ground vehicle structures. In this work, a component mode synthesis approach to structural modeling is enhanced by also modeling variations in the connections between components. With this framework, changes in the joining or fastening of the components can be considered in a structural analysis or design process. The components are condensed statically or dynamically with all the candidate joining nodes being retained as active degrees of freedom. In addition, this procedure is implemented in a systematic joining topology design optimization procedure for an example case of two structural components welded together. The welding pattern is optimized to improve the structural dynamic performance of the connected structure while satisfying constraints on the total joining area. The joining is modeled with three-dimensional spring elements connected to the candidate nodes, and the intermediate stiffness of the springs are penalized by the Solid Isotropic Material with Penalization (SIMP) modeling. The design variables are the stiffnesses of the springs, which can then be thresholded to determine the optimized welding pattern. All the joining design variables are confined to a single joining component, so only that component is updated during each iteration of the optimization procedure.

INTRODUCTION

In the analysis and design of a complex structure, it is common to take a substructuring approach: the full finite element model is divided into components, and each component is analyzed first and then assembled into a global structural model. The same is true for real design and manufacturing processes for complex structures such as automotive frames and bodies. In this environment, designers are required to choose the locations and methods for joining the parts. Ideally, the joining process would be optimized to meet performance targets such as maximum dynamic and static response levels, while keeping the cost reasonable. This can be quite challenging with even a small number connection points, let alone the thousands of spot welds and other connections that may exist in automotive bodies and other engineering structures. Therefore, it would be useful to develop a systematic design approach to optimizing connection locations.

Joining location optimization could be considered to be a special case of topology optimization [1,2]. However, it differs from traditional component design in that it could involve a large number of components at the system level. Although component-level topology design has been extensively studied throughout last two decades [2], relatively few papers have addressed system-level topology design. Some of these papers are summarized as follows. Chirehdast and Jiang [3,4] and Chickermane [5] addressed the joining topology optimization among components. In Ref. [3], optimum joining locations were obtained to minimize the static compliance of the given substructures. More recently, Zhu and Zhang [6] employed joining topology design with respect to the free vibration properties of fixtures.

While all of the above papers employed spring elements for the joining modeling, Li et al. [7] applied more advanced solid brick elements for the joining and also adopted an evolutionary structural design approach instead of a more traditional sensitivity-based topology optimization method. Although the use of brick elements produces more accurate results, the discretized joining surfaces in the finite element model need to be matched, which can be a difficult challenge in complex structural design.

The goal of this work is to present an efficient framework for topology design of joining locations so that they can be optimized for the static and/or dynamic response of a complex structure. As in the general optimization process, the topology optimization requires iterative evaluations of the finite element model to reach convergence, which can lead to high computational costs. One approach to reduce the computational burden is to parallelize the finite element analysis (FEA) required for topology optimization, which was attempted in Ref. [8]. Another approach is to employ the Craig-Bampton component mode synthesis (CB-CMS) method, which is a substructuring technique that is widely used for its accuracy and efficiency [9]. The CB-CMS method was employed in combination with topology optimization in Ref. [10]. In this work, the CB-CMS method is employed for the joining topology optimization by treating the joining between two components as a third component, which is called a joining component. It will be shown in this paper that—with an appropriate partitioning—all the joining can be confined in one joining component, resulting in fast turn-around time for each optimization iteration.

This paper is organized as follows. First, brief reviews are given on topology optimization and joining modeling, with a focus on spot-welding and the CB-CMS method. Second, the joining optimization problem is posed and the optimizers used for this work are discussed. The design sensitivity analysis is also described for completeness. Next, numerical results are presented for an example structure. Finally, conclusions from this study and suggestions for future research are summarized.

BACKGROUND

TOPOLOGY OPTIMIZATION Since the seminal paper by Bendsøe and Kikuchi [11] in 1988, topology optimization has been successfully applied to a wide range of problems because of its ability to provide initial design topology from the arbitrary fixed design domain. The distinct feature of topology design, as opposed to the size and shape design, lies in adding or removing holes in the design domain as well as considering connections among design domains. These holes are realized by variable distributed micro-holes in the heterogeneous design domain, which can grow big enough in some parts of the design domain to become macro-scale holes in the structure, or topology changes.

There are two basic stages to topology optimization:

- parameterizing the design domain
- posing and solving the optimization problem

In terms of parametrization of the design domain, the homogenization method and the Solid Isotropic Material with Penalization (SIMP) method are widely recognized. In the homogenization method, design variables are assigned to control the size and the orientation of the representation of each hole in a finite element in the design domain. By the homogenization method, the finite element with a hole can be interpolated into appropriate elastic coefficients and densities. As a shortcut of the homogenization, the SIMP method was proposed by Bendsøe [12] to replace the size and orientation variables of the holes with the density variable of the finite element in the design domain. In the SIMP method, the densities are assumed to be related to the elastic coefficients in a power law that can control the stiffness of the material. Thus, it can be used directly as an effective design variable instead of using geometric variables to account for the hole size and orientation. Despite the lack of rigorous derivation from physical material, the SIMP method was shown to be reliable in producing a solid-void structure if the penalizing power law satisfies the Hashin-Shtrikman bounds that describes the possible limits of the two-phase isotropic material [13]. For its simplicity of implementation and smaller number of design variables, the SIMP method was adopted for this study.

To solve optimization problems, some specific methods for topology optimization have been developed that can handle a large number of design variables with several constraints. Among them, the Method of Moving Asymptotes (MMA) [14, 15] and the Optimality Criterion (OC) method [11, 16] are broadly utilized for their efficacy and generality. The MMA method is based on the convex approximation method [17–19] with the advanced feature of setting asymptotic moving limits to approximation variables. One the other hand, the OC is the well-known Karush-Kuhn-Tucker (KKT) condition in conjunction with the dual method. In this work, the OC method is used for static design optimization and the modified OC method is used for dynamic design optimization.

COMPONENT MODE SYNTHESIS In this paper, the Craig-Bampton component mode synthesis (CB-CMS) method [20] is used to improve the efficiency of the structural analysis required for each iteration of the optimization process. First, the full finite element model is divided into substructures (components). Then, the physical degrees of freedom (DOF) of each component's mass and stiffness matrices are partitioned into interior and interface DOF:

$$\mathbf{M}_i = \begin{bmatrix} \mathbf{m}^{AA} & \mathbf{m}^{AO} \\ \mathbf{m}^{OA} & \mathbf{m}^{OO} \end{bmatrix}_i \quad \mathbf{K}_i = \begin{bmatrix} \mathbf{k}^{AA} & \mathbf{k}^{AO} \\ \mathbf{k}^{OA} & \mathbf{k}^{OO} \end{bmatrix}_i \quad (1)$$

where superscript A denotes “active” DOF of the interface with connected components, and O denotes “omitted” DOF of the interior. The component modes in CB-CMS consist of a set of fixed-interface normal modes Φ_i^N and static constraint modes Φ_i^C ,

$$\Phi_i^N = \text{eigenvector}(\mathbf{m}_i^{OO}, \mathbf{k}_i^{OO}) \quad (2a)$$

$$\Phi_i^C = -\mathbf{k}_i^{OO-1} \mathbf{k}_i^{OA} \quad (2b)$$

By the component mode sets of each substructure, Eq. (1) can be projected into component modal space as

$$\hat{\mathbf{M}}_i = \begin{bmatrix} \mathbf{m}^C & \mathbf{m}^{CN} \\ \mathbf{m}^{NC} & \mathbf{m}^N \end{bmatrix}_i \quad \hat{\mathbf{K}}_i = \begin{bmatrix} \mathbf{k}^C & \mathbf{k}^{CN} \\ \mathbf{k}^{NC} & \mathbf{k}^N \end{bmatrix}_i \quad (3)$$

The assembly of the substructures is similar to the finite element assembly process, which gives the name of “superelement” for the component. The assembled matrices are written as

$$\mathbf{M}^{CMS} = \begin{bmatrix} \sum_{i=1}^{n^{ss}} \mathbf{m}_i^C & \mathbf{m}_1^{CN} & \mathbf{m}_2^{CN} & \cdots & \mathbf{m}_{n^{ss}}^{CN} \\ \mathbf{m}_1^{CN^T} & \mathbf{m}_1^N & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{m}_2^{CN^T} & \mathbf{0} & \mathbf{m}_2^N & \cdots & \mathbf{0} \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ \mathbf{m}_{n^{ss}}^{CN^T} & \mathbf{0} & \mathbf{0} & \cdots & \mathbf{m}_{n^{ss}}^N \end{bmatrix}$$

$$\mathbf{K}^{CMS} = \begin{bmatrix} \sum_{i=1}^{n^{ss}} \mathbf{k}_i^C & \mathbf{0} & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{0} & \mathbf{k}_1^N & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & \mathbf{k}_2^N & \cdots & \mathbf{0} \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ \mathbf{0} & \mathbf{0} & \mathbf{0} & \cdots & \mathbf{k}_{n^{ss}}^N \end{bmatrix} \quad (4)$$

where the subscripts refer to individual substructures and n^{ss} is the total number of substructures.

JOINING MODELING METHODS Although modeling the joining/fastening of components remains perhaps one of the most improvisational tasks in structural FEA, proper modeling methods for joining have been extensively studied. Although joining could refer any type of part-assembly process (welding, fastening, riveting, etc.), spot-welding is considered in this section because of its prevalence in the automotive body engineering community.

Depending on the required accuracy and complexity of the problem at hand, an appropriate joining modeling strategies can be adopted. Several joining modeling techniques were proposed in literature [21–25] as shown in Fig. 1. For the fatigue analysis based on the local stress and strength of the materials, both the fine mesh on the structure and the accurate joining modeling as Fig. 1(c), 1(e) are required. For the dynamic or NVH analysis of a complex structure, a moderate level of accuracy and complexity is required, which leads to the simple solid modeling of the joining as shown in Fig. 1(c), 1(d). For a design optimization problem especially for the preliminary design, the simple rigid bar modeling as shown in Fig. 1(a), 1(b) is preferred [3, 5, 23, 26] because the joining modeling can be easily catered for the iterative updating, which should be equipped for the design optimization.

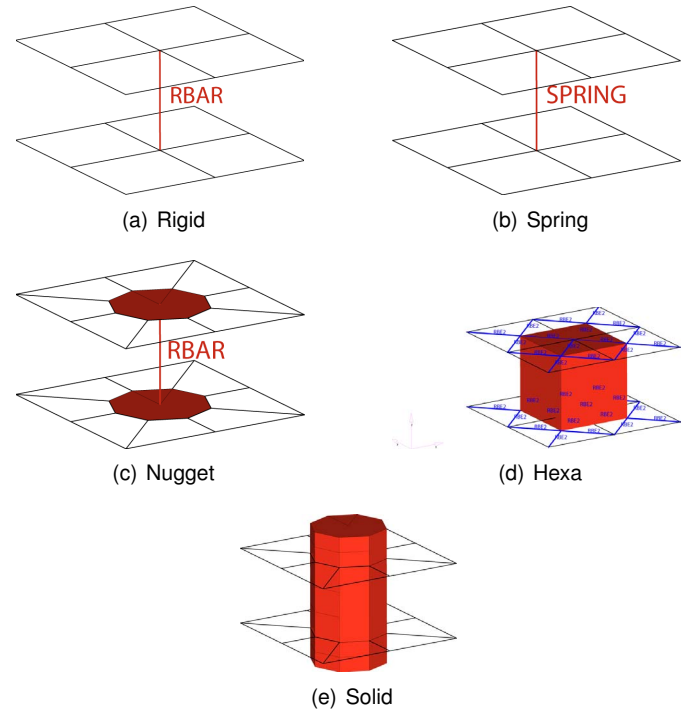


Figure 1: Some of the welding modeling practices used in structural FEA. The spring connection shown in (b) is used throughout this work.

In this work, each spot-weld connection is modeled as a three-directional spring to enable the simple and automatic updating of the joining in the global model. Let the stiffness of the joining be $\mathbf{K}_s = \text{Diag}([k_x \ k_y \ k_z])$, where $k_{x,y,z}$ denotes stiffness of the spring along each cartesian coordinate. Note that the three joining stiffnesses at one point in the structure tend to be correlated. In this work, it is assumed that $k_x, k_z = \mu_k k_y$. Thus, the joint modeled as three springs can be described as

$$\mathbf{K}_s = \begin{bmatrix} K_s & -K_s \\ -K_s & K_s \end{bmatrix} \quad (5)$$

where $K_s = k_y \text{Diag}([\mu_k \ 1 \ \mu_k])$

JOINING DESIGN PROBLEM STATEMENTS

In this section, the joining design statements are formulated both for static and dynamic cases. They are based on the the minimization of the static compliance and its extension on dynamic compliances [16, 27].

STATIC PROBLEMS First, consider the minimization of the internal strain energy of the structure under static load. Because the internal strain energy is equal to the external work done on the structure, it is equivalent to minimize the external work, which is also defined as the compliance of the structure. Therefore, the minimization formulation for the discretized domain can be written as

$$\begin{aligned} \min_{\mathbf{U}, \mathbf{E}_e} \quad & \mathbf{f}^T \mathbf{U} \\ \text{subject to:} \quad & \mathbf{K}(\mathbf{E}_e) \mathbf{U} = \mathbf{f}, \quad \text{and} \quad \mathbf{E}_e \in \mathbf{E}_{ad}. \end{aligned} \quad (6)$$

Two fields of interest are the displacement $\mathbf{U}$ and the stiffness $\mathbf{E}$. $\mathbf{E}_{ad}$ denotes the set of admissible stiffness tensors and should be defined appropriately to be consistent with the topology domain, which is the distinct feature of Eq. (6) from size and shape optimization problems.

According to the SIMP method, the elasticity tensor in the design elements are written as

$$\mathbf{E}_e = \rho_e^p \mathbf{E}_0, \quad p > 1. \quad (7)$$

With the active volume constraints on the usage of the materials, the intermediate value of ρ ($0 < \rho < 1$) is penalized compared with 0, 1 design in terms of obtaining the same stiffness ($\sim \rho^p$) by setting the parameter $p > 1$.

In the context of SIMP modeling Eq. (7), the intermediate joining stiffness of Eq. (5) can be penalized for the density ρ to attain the 0, 1 design. Thus, the stiffness matrix of the joining in SIMP becomes

$$\mathbf{K}_e = \rho_e^p \begin{bmatrix} K_s & -K_s \\ -K_s & K_s \end{bmatrix} = \rho_e^p \mathbf{K}_s^0 \quad (8)$$

Because it is assumed that the ratio of the minor directional stiffness with respect to the main directional stiffness is fixed throughout the optimization process, one density variable is assigned to one joining. The total stiffness of the structure which is the sum of the stiffness matrix for the structure without connections ("ground structure") and the joining stiffness matrix:

$$\mathbf{K}_{st} = \mathbf{K}_{ground} + \sum_{e=1}^{n_{var}} \rho_e^p \begin{bmatrix} K_s & -K_s \\ -K_s & K_s \end{bmatrix} \quad (9)$$

where $\sum_{e=1}^{n_{var}} \mathbf{A}_e$ denotes the usual CB-CMS global matrix assembly process [20], which is used to obtain the global joining stiffness matrix from the local submatrices. In the joining optimization, instead of placing a constraint on the total volume of the materials, the number of connections among components can be limited as

$$\sum_{s=1}^N \rho_s = \mathbf{V} \leq \mathbf{N} \quad (10)$$

For the multiple load cases, the response can be evaluated for each load case and summed into a single objective function. By substituting $\mathbf{K}$ with $\mathbf{K}_{st}$ and the constraint on the number of connections, Eq. (6) becomes

$$\begin{aligned} \min_{\rho} \quad & c(\rho) = \sum_{load=1}^{l_n} \mathbf{f}^T \mathbf{U} = \sum_{load=1}^{l_n} \mathbf{U}^T \mathbf{K}_{st} \mathbf{U} \\ \text{s.t.:} \quad & g(\rho) = \sum_{e=1}^{n_{var}} \rho_e - \mathbf{N} \leq 0; \quad 0 < \rho_{min} \leq \rho_e \leq 1 \end{aligned} \quad (11)$$

where $\mathbf{U}$ should be obtained by static FEA such that $\mathbf{K}_{st} \mathbf{U} = \mathbf{f}_{ext}$.

DYNAMIC PROBLEMS For dynamic optimization, the focus of this study is the topology design optimization of the structure under external harmonic forcing. This topic has been studied by Ma et al. [16], among others. In Ref. [16], the dynamic compliance was defined first and the modified OC method was proposed. The updating scheme for the modified OC method was shown to be guaranteed to work regardless of the sign of objective sensitivity. Following their work, in this paper the objective to be minimized is defined as the absolute value of dynamic stiffness in the frequency range of $[f_{low}, f_{high}]$ for multiple load cases as

$$\min_{\rho} \int_{f_{low}}^{f_{high}} \sum_{load=1}^{l_n} |\mathbf{f}^T \mathbf{U}(\rho)| df \quad (12)$$

where $\mathbf{U}$ and $\mathbf{f}$ are complex for the general case of a damped system. With Eq. (12), the dynamic topology optimization statement becomes

$$\begin{aligned} \min_{\rho} \quad & \int_{f_{low}}^{f_{high}} \sum_{load=1}^{l_n} |\mathbf{f}^T \mathbf{U}(\rho)| df \\ \text{s.t.:} \quad & g(\rho) = \sum_{e=1}^{n_{var}} \rho_e - \mathbf{N} \leq 0; \quad 0 < \rho_{min} \leq \rho_e \leq 1 \end{aligned} \quad (13)$$