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STRUCTURAL OPTIMIZATION IN AUTOMOTIVE DESIGN

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TYPICAL ENGINEERING DESIGN ORGANIZATION

Although mathematical structural optimization has been an active research area for twenty years, there has been relatively little penetration into the design process. Experience indicates that often this is due to the traditional layout-analysis design process. In many cases, optimization efforts have been outgrowths of analysis groups which are themselves appendages to the traditional design process. As a result, optimization is often introduced into the design process too late to have a significant effect because many potential design variables have already been fixed. A series of examples (Ref. 1-6) will be given to indicate how structural optimization has been effectively integrated into the design process (Fig. 1).

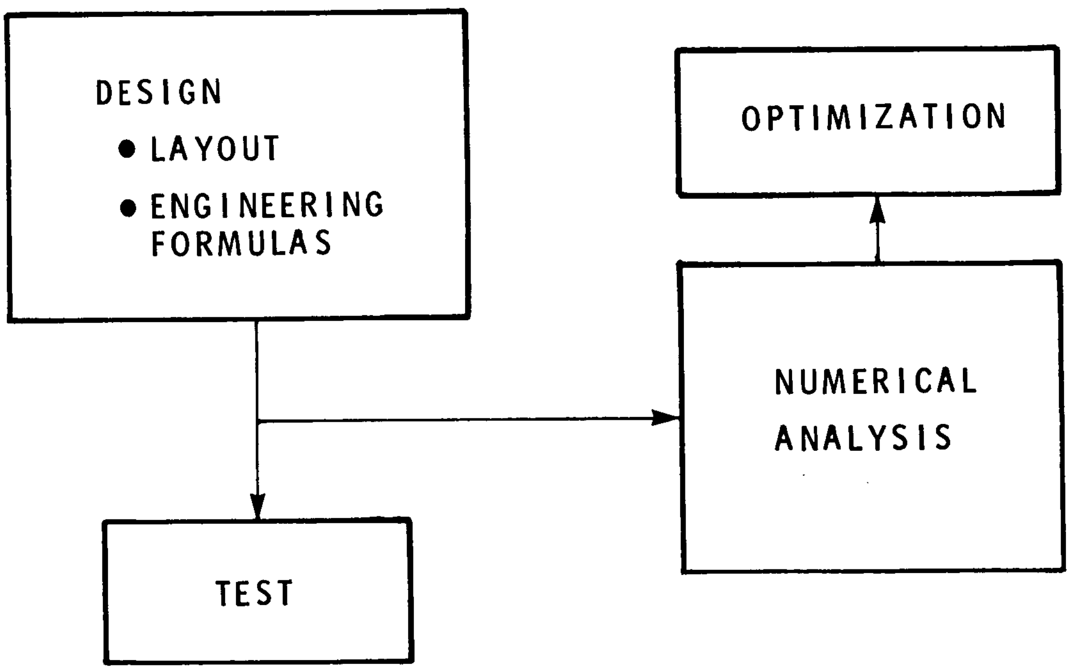


Figure 1

TYPICAL BEAM SECTIONS AVAILABLE IN OPTIMIZATION

The examples in this paper have been obtained with a general purpose structural optimization code developed at the General Motors Research Laboratories which allows both constraint approximation methods and full mathematical programming methods with exact constraint evaluation to be used as required. A feasible directions algorithm is used as the optimizer in both cases. A design library of thin-walled beam elements (Fig. 2) and triangular plate elements (bending and membrane) is available. Multiple load conditions and multiple boundary conditions may be applied and frequency, displacement, and stress constraints may be used.

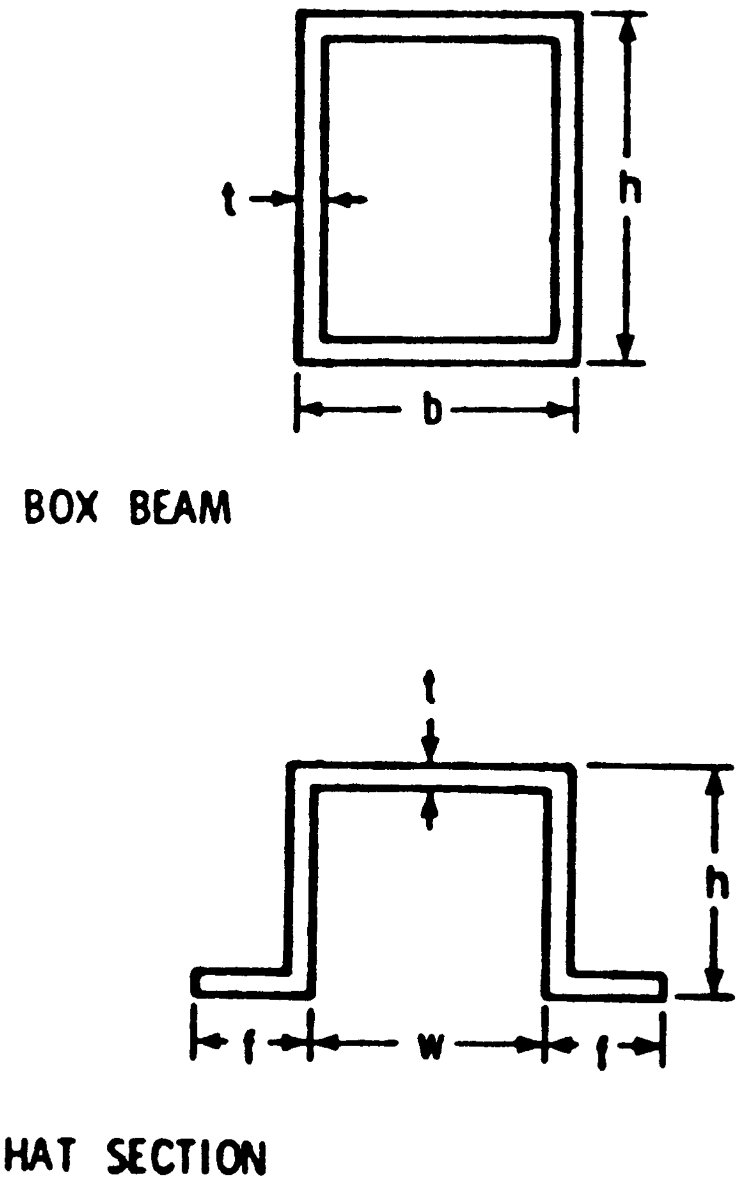
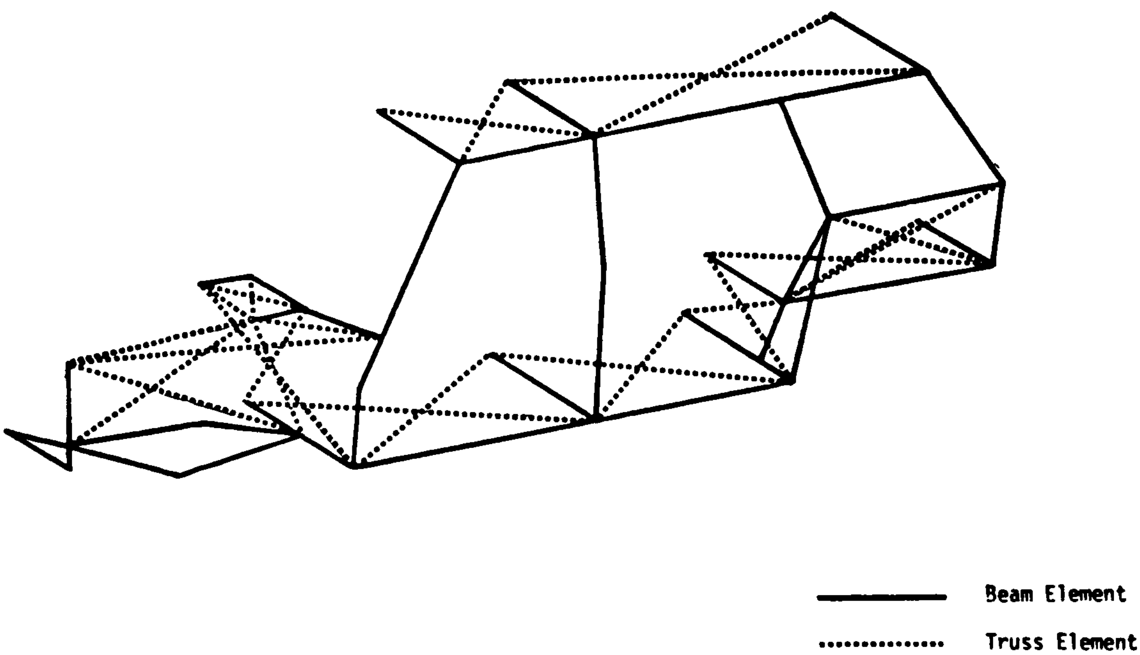


Figure 2

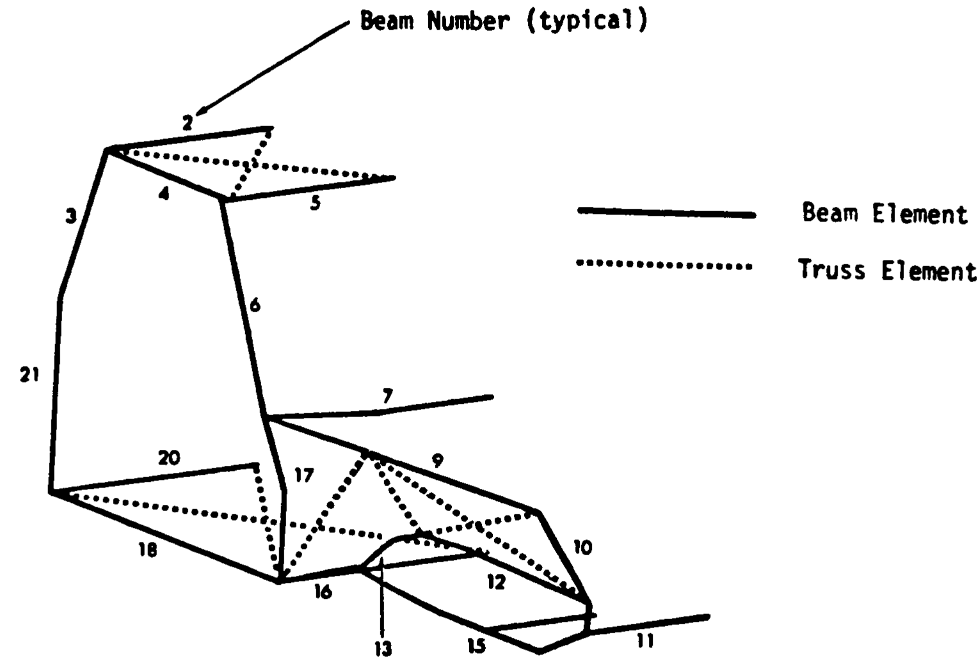
EARLY CONFIGURATION DECISIONS

There are often several competing structural configurations for a major portion of the structure. Rarely are these competing configurations examined on a rational basis. This example examines an optimization study of three configurations proposed for a front structure. The structures were split into upper and lower configurations. Front structure I may be characterized by an upper structure securely attached to the cowl bar and a lower structure comprised of a mid-rail and triangulated lower rail. Structures II and III each have an irregular slanted shear wall for the upper structure and a mid-rail and engine cradle comprising the lower structure, with structure III having an additional under-car longitudinal rail. Each of these front structures was modeled on a common rear structure as shown in Fig. 3. The remaining front structures are shown in Figs. 4 and 5. All structures were subjected to the same set of force load conditions and frequency constraints. In the optimization, all beam cross section dimensions, including widths and heights, were taken as design variables. In addition, beams throughout the structure, not just in the front structure, were allowed to vary. It has been found that relatively simple beam models with truss elements representing the stiffness of critical panels have been sufficient for preliminary design.



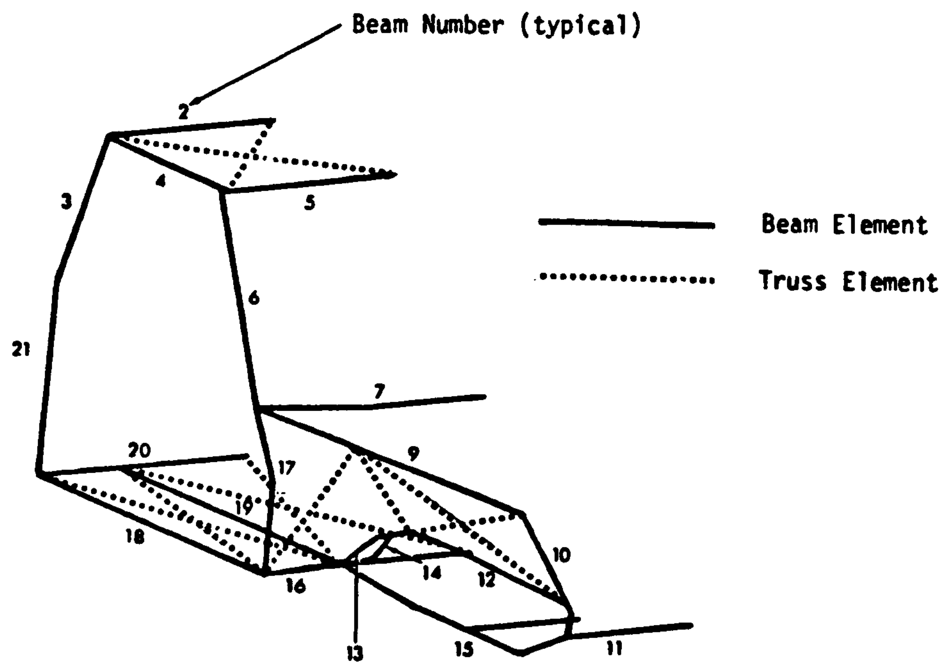
STRUCTURE I

Figure 3



FRONT STRUCTURE II
(Rear structure common with Structure I)

Figure 4



FRONT STRUCTURE III
(Rear structure common with Structure I)

Figure 5

LOAD CONDITIONS AND CONSTRAINTS

It is necessary to include an extensive set of load conditions so that all possible critical load conditions are covered (Fig. 6). Typically, 10-15 loads, including static, inertia relief, and frequency conditions, are used.

Symmetric Load Conditions

- **Jacking (statics)**
- **4 g bump both front wheels (inertia relief)**
- **4 g bump both rear wheels (inertia relief)**
- **1 g brake (inertia relief)**
- **Front bumper (inertia relief)**
- **Rear bumper (inertia relief)**
- **Roof crush (statics)**
- **Cowl crush (statics)**
- **Roof bow (statics)**

Asymmetric Load Conditions

- **4 g bump one front wheel (inertia relief)**
- **4 g bump one rear wheel (inertia relief)**
- **Torsional jacking (statics)**

Frequency Constraints

- **Symmetric - first mode > 18 Hz**
- **Asymmetric - first mode > 21 Hz**

Figure 6

OPTIMUM MASS SUMMARY

The total structural masses for the front end configurations considered are shown in Fig. 7. The lower III/upper I configuration, with a mass of 127.4 kg, was the lightest of the structures. It is interesting to note here that the difference in total mass between the lightest and heaviest of the acceptable designs is only 8.2 kg, approximately 6.5%. Given the apparent differences in the load-carrying capabilities and stiffness characteristics of the various front structures, it would seem that the structure, as a whole, must have been able to compensate for the inherent differences in load-carrying capacity of a particular configuration, resulting in a series of designs having virtually the same total mass but different mass distributions. This indicated that nonstructural reasons could be used to make the final selection. The important consideration here is that all designs met the same load criteria since they were all treated as constraints in the optimization. Thus, by entering the early phase of the design process, important design direction was given by optimization.

Front Structure Configuration	Total Mass (kg)
1. Lower III / Upper I	127.4
2. Lower III / Upper II	132.7
3. Lower II / Upper II	135.2
4. Lower II / Upper I	135.6

Figure 7

ROCKER SECTION STUDY

As the design progresses, nonstructural decisions begin to dictate the shapes of various structural members. While the shapes of these members should be influenced by the earlier optimization study, often the nonstructural influenees prevail. This effect can be evaluated as shown in Fig. 8. In this case, the proposed rocker section was replaced in the model and only the thickness was allowed to vary in this section. In addition, the rest of the design variables in the remainder of the structure were also allowed to change. The proposed irregular section produced a mass penalty of 4.51 g. This was deemed severe enough to attempt a redesign of this component. Again, this information is difficult to obtain without an optimization capability.

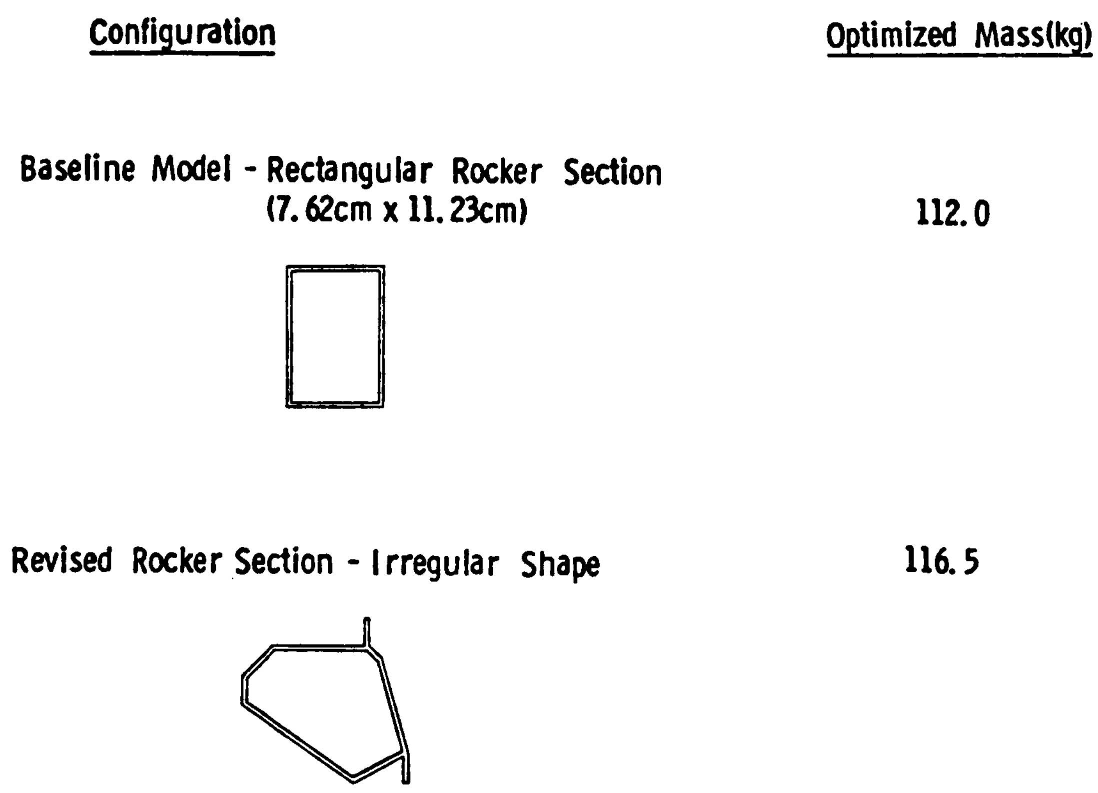


Figure 8

HOOD STRUCTURE OPTIMIZATION MODEL

As a final example, we will take the design of a secondary structural component of a typical construction in which the inner structure is primarily a beam structure and the outer is a plate structure (Fig. 9). This detailed model clearly would occur later in the design process, as opposed to the simpler models shown in the other two examples.

For this particular study, the outer structure was assumed to be of constant thickness. Each of the inner beams was assumed to be a channel section of constant thickness and size. The heights of all beams were set at 2.5 cm.

Two load conditions were used for this study. The first assumed the hood was supported on three of its four support points, and a deflection constraint of 2.0 cm was placed on the fourth point under a dead weight load. This load was the estimated final mass of the hood uniformly distributed on all nodes. The second load condition was the hood in its fully supported condition with a 75 kg load distributed over the center portion. Each load condition required a separate boundary condition set.

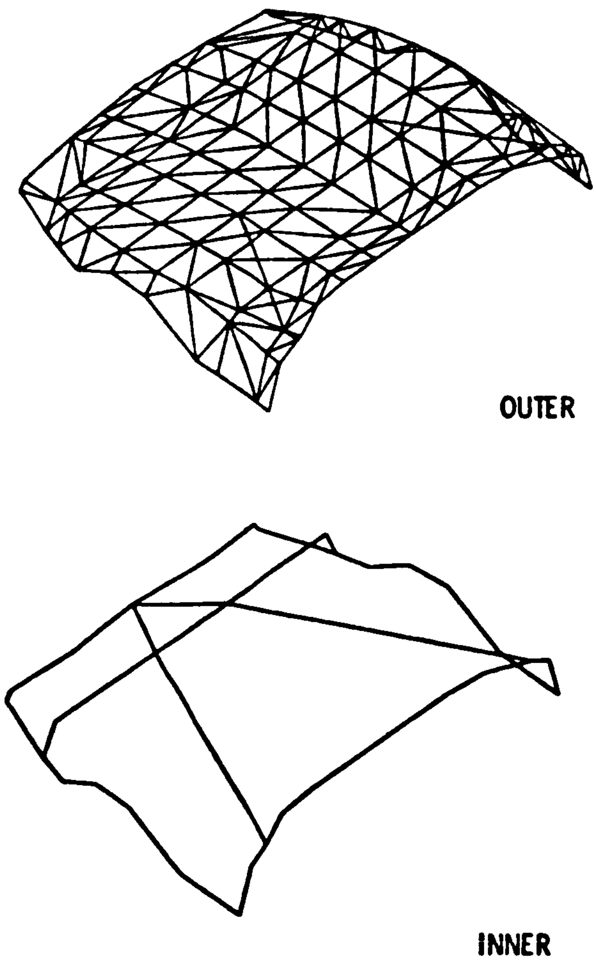
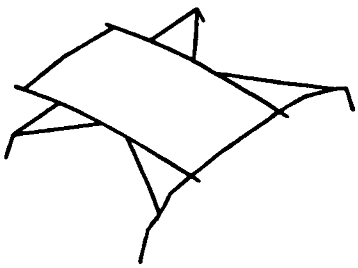


Figure 9

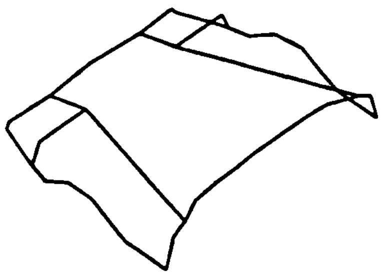
HOOD STRUCTURE INNER CONFIGURATION

Three different stiffener patterns were optimized as shown in Fig. 10. As might be expected, the more triangulated structure required the lowest mass. In this design, the minimum width of the beam section was allowed to be a very small number (0.15 cm). As the width of the channel section approaches this number, the section approaches a blade type of stiffener, typical of molded SMC structures or a hem flange or turned edge in steel. As can be seen from Fig. 11, beams 3 and 4 reached this condition. Since beam 3 is on an edge, this suggests a turned edge would be sufficient. In this example, more detailed information about the final structure is being obtained.

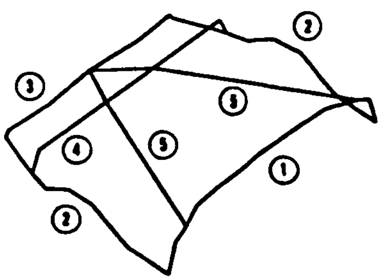
Total Mass



28.9 kg



28.2 kg



22.6 kg

Figure 10

Mass = 22.6 kg

Beam	t	width	flange	height
1	.076	1.14	.05	2.5
2	.076	1.21	.05	2.5
3	.076	.36	.05	2.5
4	.076	.33	.05	2.5
5	.076	1.31	.05	2.5
skin	.076			

skin mass = 17.3 kg
Beam mass = 5.3 kg

Dimensions in cm

Figure 11

BOUNDARY ELEMENTS

Ultimately one would like to merely describe the function and limitations of the structure in some conceptually convenient terms and then allow the computer to automatically make adjustments in some way to produce a best design. This process will require the implementation of a boundary-based description of the problem as opposed to a nodal description as used in typical finite element analysis programs. Since the design process will be under the control of an optimization program, the analysis mesh must continually be generated as the design changes. In addition, it is necessary to guarantee the continuing accuracy of the analysis as the design changes. These considerations suggest the integration of a boundary-based automatic mesh generation scheme with adaptive mesh refinement techniques and structural optimization to produce an effective shape optimization program.

A mesh generator for multi-connected, two-dimensional regions which requires only boundary information was chosen. This information is initially a continuous description which is then discretized. The algorithm then distributes points uniformly throughout the region and connects them to form triangles. An averaging form of smoothing is applied to produce triangles of roughly uniform shape. The problem can then be described in terms of a set of boundary design elements, each of which has associated with it a set of design variables (Fig. 12). As the design changes, the new mesh can be generated from the new boundary description.

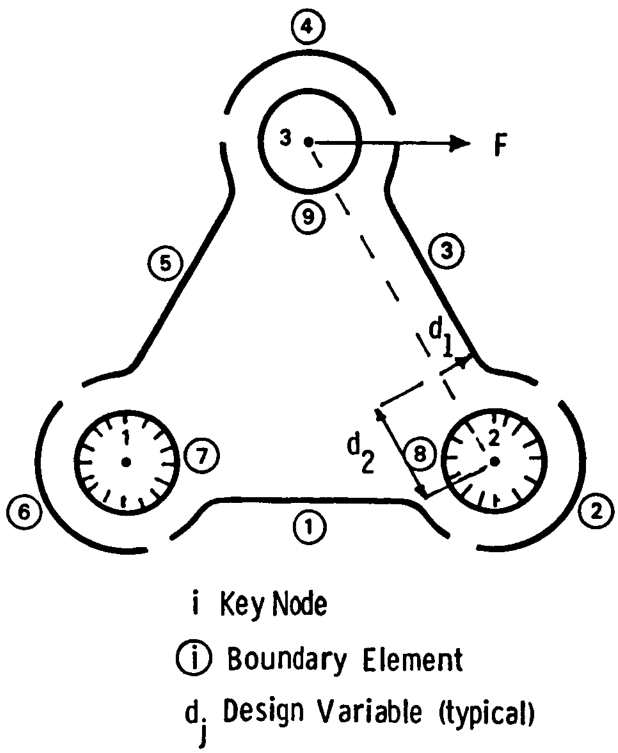


Figure 12

c-3

MESH REFINEMENT

When finite element analysis is used for a fixed configuration optimization, the integrity of the model is assured at the start of the optimization and is assumed to remain acceptable throughout the design process. However, when the design process is changing the shape of the part and the shape and location of cutouts, this assumption is no longer valid. One way of handling this problem is to use the concept of adaptive mesh refinement. In this concept, information from one analysis is used to identify regions of the finite element mesh which need further refinement. This refinement can take the form either of adding additional elements in the area to be refined or of increasing the order of the existing finite elements. The mesh refinement approach has been chosen since it can be used with existing elements and does not require the formulation of new finite elements. In addition, it can be effectively integrated with the mesh generation scheme described earlier since it merely involves the addition of more points to be triangulated. Regions of refinement are based on strain energy density (SED) gradient contours. Typical contours and a refined area are shown in Fig. 13.

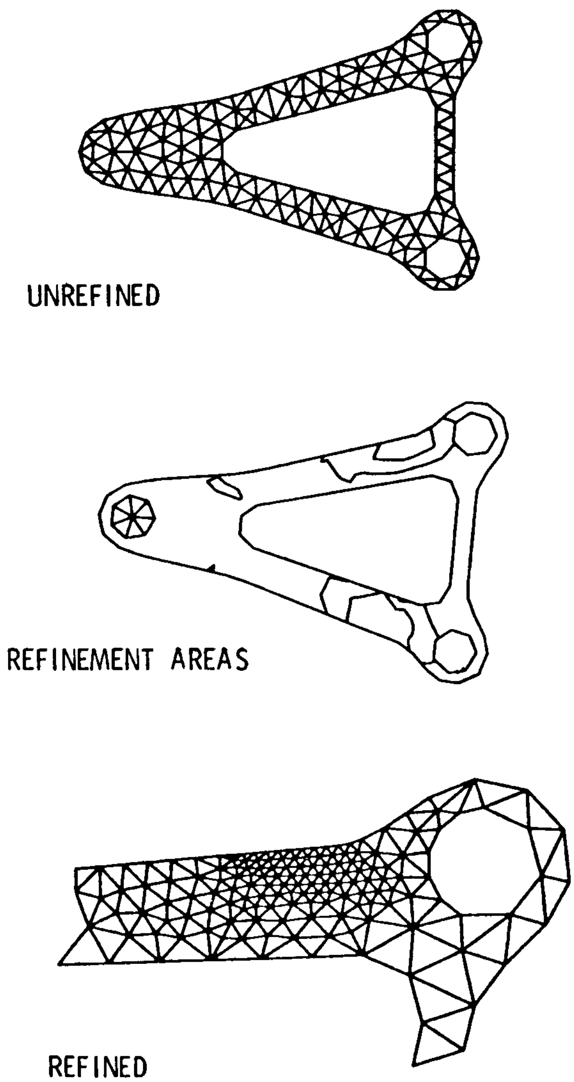


Figure 13

NONPLANAR STRUCTURES

It is convenient to think of three distinct forms of nonplanar thin structures. The first of these structures, for example, can be described by a mathematical transformation from a simple flat surface into a cylindrical surface. Secondly, the surface may take the form of a general shallow shell which may not be obtained from a simple mapping relationship but can be obtained by projection. Thirdly, the structure may be made up of several segments which may be either planar or one of the two previously mentioned forms (Fig. 14). In each of these forms, the ideas discussed in Ref. 5 can be used in the planar form to describe the segments, generate the mesh, and carry out the refinement.

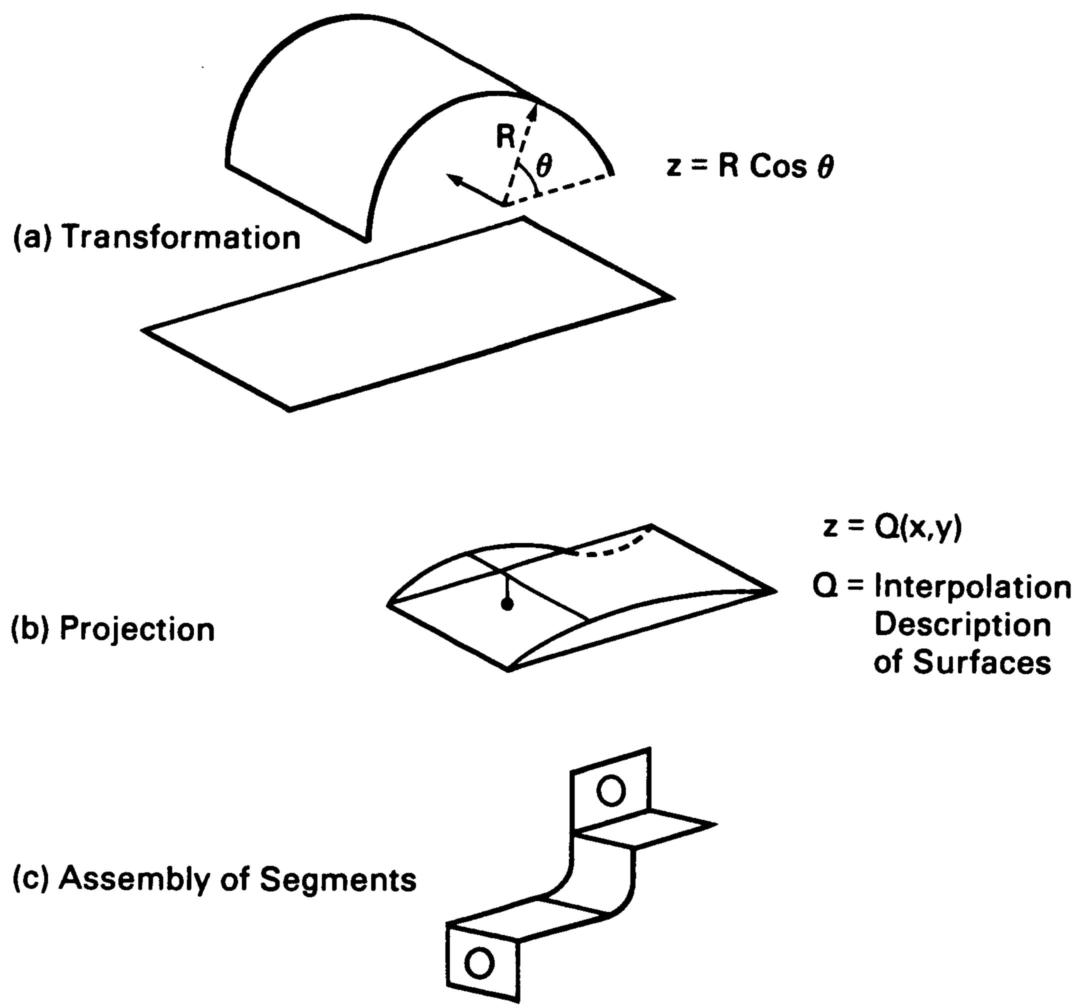


Figure 14

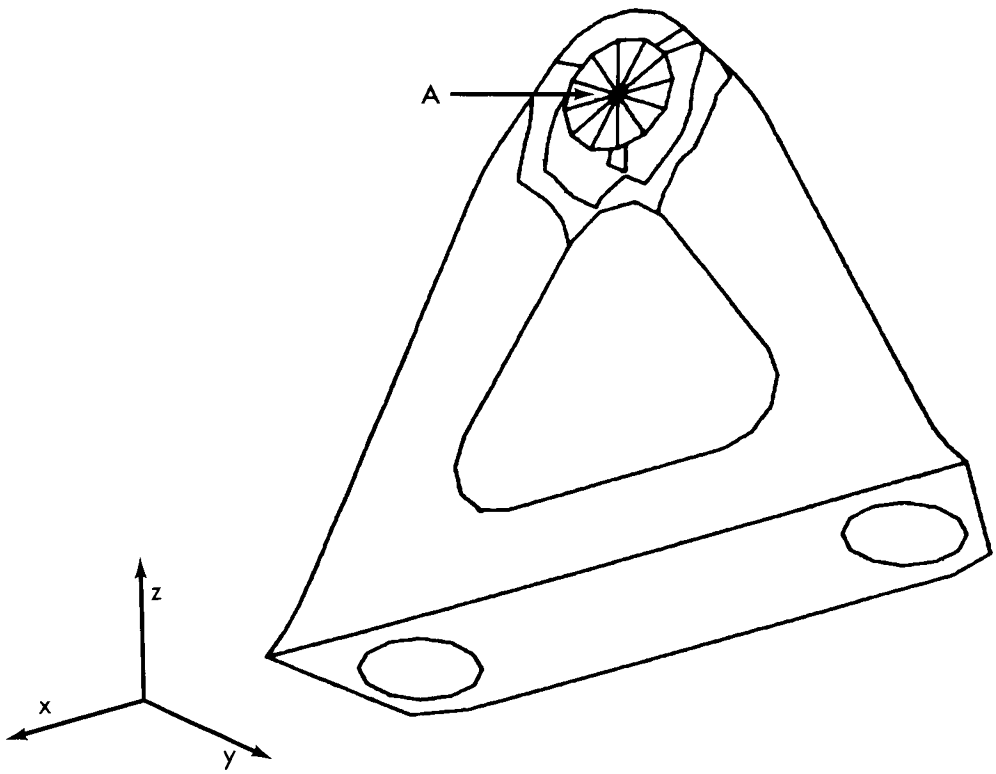
FOLDED PLATE EXAMPLE

An example of a plate folded through a 90° angle is shown in Fig. 15. A static loading of 400 N is applied to point A normal to the plane of the triangular segment, thus causing bending moments in the plate. After the structure has been triangulated, it is rotated as required.

Eleven design variables control the shape of the plate. The outer edge of the lower segment is the double cubic shape design element type with four design variables. Each of the sloping outer edges of the upper segment is a double cubic but with only two design variables each. The size of the triangular interior cutout is controlled by the location of the key nodes. The z-coordinates of all the nodes and the x-coordinates of the two bottom nodes are variables. The variables are appropriately linked to yield a symmetric design. The material thickness was also allowed to vary but remained at minimum gage throughout the design.

The stress in the structure was constrained to be everywhere less than the yield stress. In addition, geometric behavior constraints were imposed to limit the minimum distance between boundary segments to be less than 0.29 cm.

A plot of mass versus optimization step number is shown in Fig. 16. Plots of the initial and final designs are shown in Figs. 15 and 17 with the strain energy difference contours showing the areas which were refined in the design. The size of the triangular cutout was limited by stress constraints. The boundaries along the folded edge, however, were controlled by the geometric behavior constraint which limits how close two edges may be to each other.



INITIAL DESIGN

Figure 15

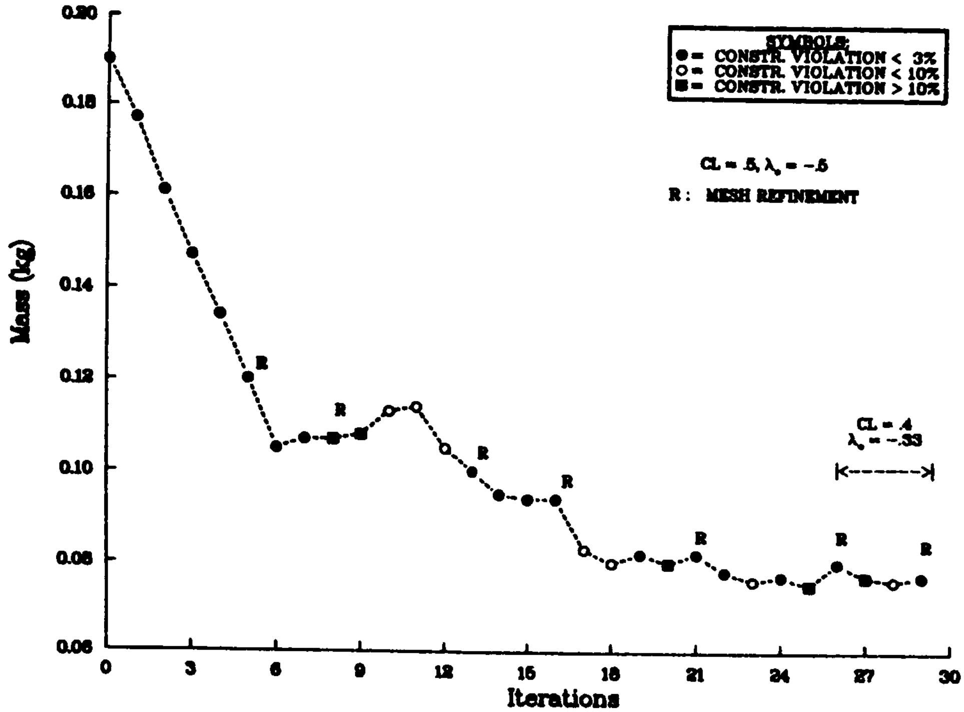
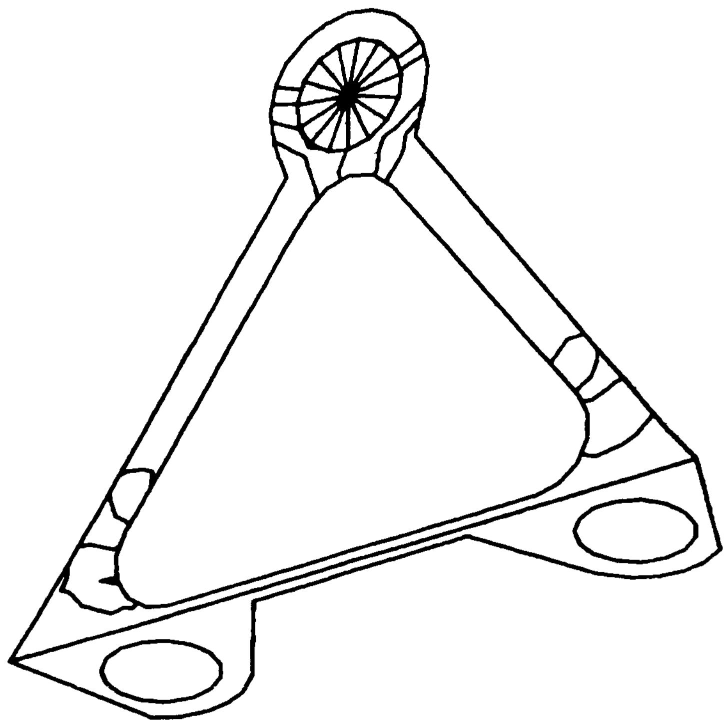


Figure 16



FINAL DESIGN

Figure 17

OBSERVATIONS

1. The mathematical tools exist to develop an effective structural optimization program. These tools may have to be developed for a particular industrial situation.

2. Optimization can be most effective if it is initiated in the preliminary design phase with simple models when the critical parameters of the design can be most affected. This requires an easily used optimization program.

3. An organization arrangement where optimization is introduced through an analysis group which is appended to the traditional design and test organization will probably not be successful because by the time optimization is applied, few design freedoms will be available.

4. The finite element model used must be accurate and the load conditions and constraints must be carefully chosen. Therefore, the user must possess the same universality of view required of the traditional engineering designer with the appreciation of the numerical aspects required of the finite element analyst. This combination of skills is not evident in either distinct group, and it will be necessary to provide a thoughtful learning environment to produce engineers who can effectively use these new tools.

5. The approach taken in the shape optimization in which the finite element model is generated from a design description of the part suggests a direction which will resolve some of the concerns described above.

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