NASA SP-5039

TECHNOLOGY UTILIZATION

STRUCTURAL DESIGN CONCEPTS

SOME NASA CONTRIBUTIONS

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA SP-5039

P-76

STRUCTURAL DESIGN CONCEPTS

SOME NASA CONTRIBUTIONS

By L. Albert Scipio
University of Pittsburgh

Technology Utilization Division
OFFICE OF TECHNOLOGY UTILIZATION 1967
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Washington, D.C.

For Sale by the Superintendent of Documents,
U.S. Government Printing Office, Washington, D.C. 20402
Price 70 cents

Library of Congress Catalog Card Number 67–61859

Foreword

The National Aeronautics and Space Administration's work has accelerated progress in many techniques as essential to man's welfare on Earth as to the exploration of space. It has, for example, promoted the rational design of new composite materials for use in its structures. Simultaneously, its researchers and contractors have explored new structural concepts, and used electronic computers to help solve unprecedented design problems.

The Office of Technology Utilization strives to make the results of such work widely available. Prof. L. Albert Scipio of the University of Pittsburgh prepared this report on some of the structural design concepts with which NASA has been concerned. It is addressed to engineers and intended to facilitate their use of findings in the aerospace industry.

GEORGE J. HOWICK, *Director,*
Technology Utilization Division,
National Aeronautics and Space Administration.

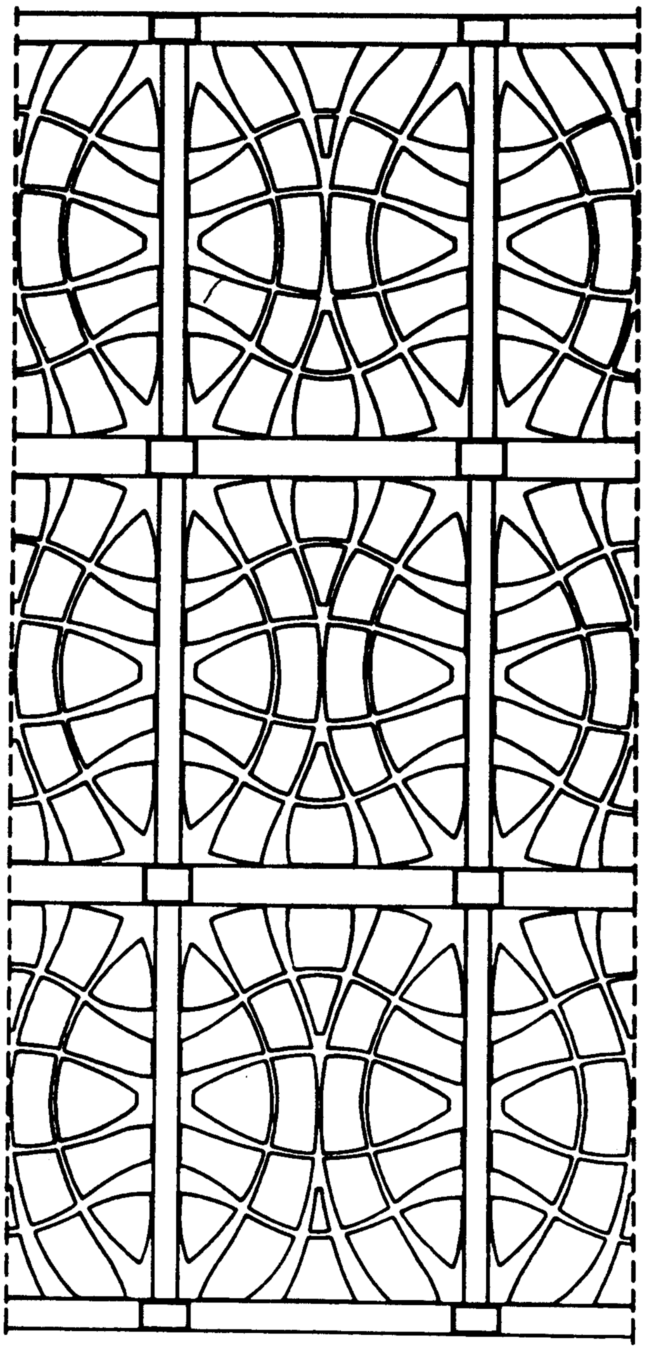
Acknowledgments

More persons and organizations than can be listed here helped the author prepare this survey. NASA personnel whose advice and assistance he would like to acknowledge include Dr. Richard L. Lesher, Assistant Administrator for Technology Utilization; George J. Howick, Director, Technology Utilization Division; and Technology Utilization Officers at the Goddard Space Flight Center, Lewis Research Center, Langley Research Center, and Marshall Space Flight Center. Specialists whose help was especially appreciated include Roger A. Anderson, Assistant Chief, Structures Research Division, Langley Research Center; J. F. Blumrich, Chief, Structural Engineering Branch, Marshall Space Flight Center; T. G. Butler, Head, Structural Analysis Computer Group, Structural Dynamics Branch, Goddard Space Flight Center; and R. H. Johns, Head, Structures Analysis Section, Lewis Research Center.

Organizations that were especially helpful were the American Cyanamid Co.; American Iron & Steel Institute; Fabricon Products; Fibreglass, Ltd. (England); General Electric Co. (Missile and Space Division); Goodyear Aerospace Corp.; Hercules, Inc. (Chemical Propulsion Division); Holiday Manufacturing Co.; Molded Fiber Glass Co.; Morrison-Gottlieb, Inc.; Narmco Research & Development; Owens-Corning Fiberglas Corp.; and the Black-Clawson Co.

Others whose help was essential include Profs. N. Lewis Buck, chairman, Department of Mechanical Engineering, and Allen Kent, director, Knowledge Availability Center, at the University of Pittsburgh.

Contents



An isostatic-ribbed plate can develop "grid action" in any direction. Nervi has used it in floor designs. (*Reprinted by permission of Prentice-Hall, Inc.*)

CHAPTER 1

Introduction

As man pushes back the frontiers of high-speed flight, changes in structural design requirements necessitate the development of new materials. During the last few years, material and structural engineers have solved many formidable problems in ways that have led to significant changes in the design process itself. They have been able to develop new materials and designs without sacrificing structural-weight to gross-operating-weight ratios. In fact, these ratios are generally comparable to, or in many cases better than, those for slower vehicles. This new knowledge and experience are providing a firm foundation for the design of structures to meet individual as well as space requirements.

This survey has three main aims: (1) to identify for those in the field of structural design the contributions of the National Aeronautics and Space Administration (NASA) and the programs it has sponsored; (2) to describe the development of construction materials associated with these advances; and (3) to suggest, by examples, some of the applications in which they may be used. The survey covers structural types (including material systems), structural concepts, and structural design synthesis and optimization. While the analyst may not find this approach as sophisticated as the designer will, he may gain from it some insight into the development of new structural design concepts. The degree to which these and other developments ultimately are utilized commercially depends on the foresight and ingenuity of structural designers.

Selection of materials and structural design to meet specific performance requirements is a complex problem. Each configuration and each part of the configuration must be examined and analyzed to provide the best possible structure for each application. Although we can only scratch the surface of the subject, we have attempted to offer some guidelines for structural designers in material selection, design approach, and optimization procedures for minimum-weight design.

CHAPTER 2

Structural Types

The basic structural element of the modern aircraft, missile, launch vehicle, and spacecraft is the thin-walled shell. Many structural types were proposed and tested to strengthen this shell wall; the more successful ones have become standard for similar types of structures. These include (1) stiffened skin, such as corrugation-skin construction; and (2) composite materials, such as fiber, particulate, flake, filled, and laminate composites. In this chapter, the basic characteristics of these structural types will be examined briefly.

STIFFENED SKIN

One of the first steps in the structural design of a configuration is an investigation of the structural type that will best fulfill the strength requirements. The designer may have several possible choices. Here we will restrict our discussion to designs in which the use of heavy members is avoided. The designer then has the option of carrying the load pressure, bending, compression, and shear by means of semimonocoque structures. In our presentation, the term "semimonocoque" designates a skin structure that is stiffened by a number of reinforcing elements. Two general types may be used, the stiffened skin and the corrugation skin. Quite often both are referred to as "stiffened skin" or "reinforced skin" structures.

Stiffened-skin structures generally consist of reinforcing members that run in one direction only. These members can be attached with rivets, spot or fusion welded, or machined or "chem-milled" integrally with the skin. The bending and compressive loads are carried mainly by stiffeners, whereas the skin supports shear loads and twisting moments. A variation of this concept, which provides bidirectional rigidity, is the grid-stiffened skin. A system composed of stiffeners in two directions may be more efficient than one having either stiffener orientation alone.

The corrugation-skin structure is particularly efficient where the loading is predominantly unidirectional. The corrugation is assembled on the skin by riveting or open spot welding. A variation of this concept, which also provides bidirectional stiffening, is the waffle structure, in which a two-directional pattern is milled on one side of the skin.

Figure 1 (from ref. 1) shows several versions of stiffened-skin construction. Reference 2 discusses the use of stiffened-skin construction in hypersonic vehicles.

Many theories of the semimonocoque have been developed, and the bibliography lists sources of a vast wealth of information. A recent paper by Hoff (ref. 3) deals with new advances in the analysis of semimonocoque structures. Recent developments in the analysis of orthotropic plates and shells are also fairly well documented in such sources.

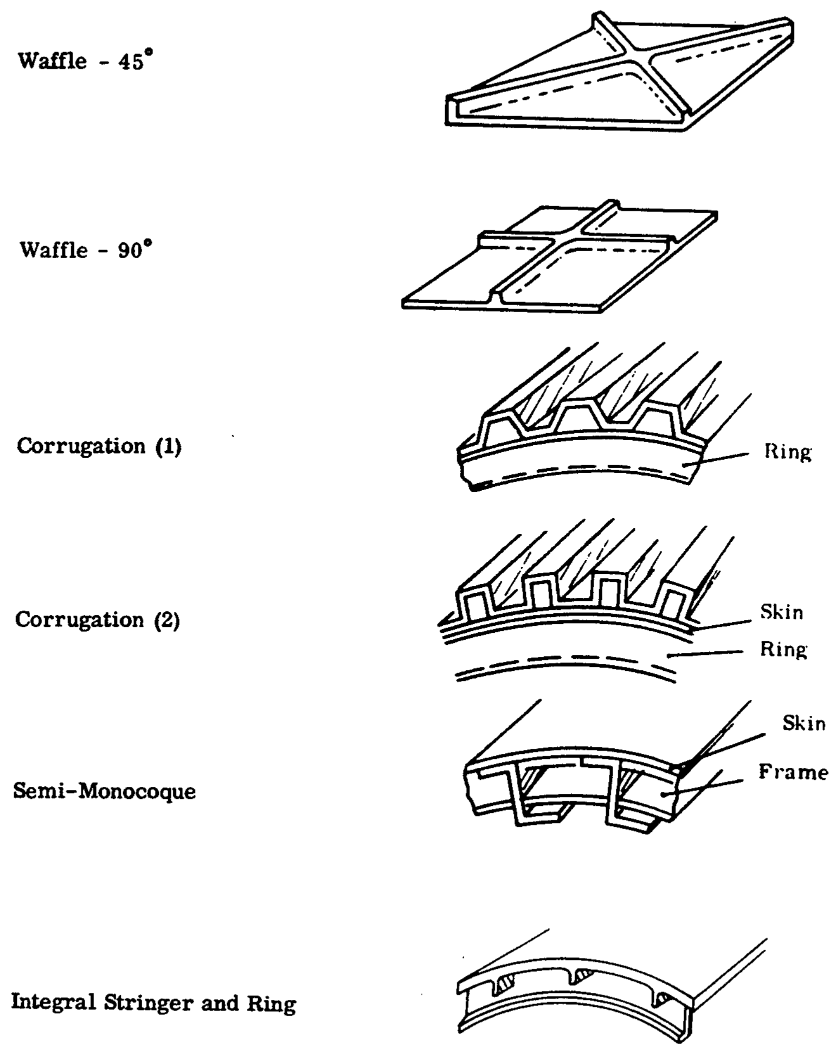


FIGURE 1.—Types of stiffened skin construction.

COMPOSITE MATERIALS

To meet requirements of high-speed aircraft, design engineers were forced to utilize materials up to, and beyond, their practical limits. The material barrier was a major obstacle to further development. To meet the demands, new material mixtures, or composites were developed; these were strong yet light and able to withstand severe temperature and corrosive conditions. Composites, in fact, seem now to represent "dream" materials with tailormade properties.

Composite materials are not new. The Babylonians are credited with discovering that chopped straw added to wet earth extended the life of a wall and enabled it to support more weight. The Egyptians used plaster of paris laced with hair to prepare their tomb walls. What these ancients discovered was that a mixture of materials is often stronger than any of the individual components. This concept is now well established in many technologies.

What do we mean now by a composite material? Definitions in the literature differ widely, as can be seen in references 4 and 5. The most appropriate definition of the composite materials covered in this book is: a mixture or combination of two or more macroconstituents that differ in form and/or material composition and that are essentially insoluble in one another. Although this working definition is not wholly adequate, it takes into account both the composition of the material constituents and the structural form.

A precise definition is difficult to formulate because of a scale factor (ref. 5). At the atomic level all elements are composites of electrons and nuclei; at the crystalline and molecular levels, materials are composites of different atoms; and at successively larger scales, materials may become new types of composites, or they may appear to be homogeneous (ref. 6). In our presentation, we will limit the discussion to the macroscale. Many metallic alloys that are composites of several quite different constituents become homogeneous materials on a macroscale. Specific examples of such materials are dispersion-hardened alloys and cermets. Some engineers will find our definition of composites too broad because it includes several engineering materials that are not usually considered composites, such as concrete, impregnated materials, filled plastics, and precoated materials. All such materials fall within the concept of composites, however, and should be treated as such.

Since our definition does not make a clear distinction between composites and composite structures, some combinations may be considered to be composite structures rather than composite materials. For example, there are differences of opinion as to whether a sandwich should be classified as a structure or a material. Although a precise distinction is extremely difficult to make, the following discussion should be helpful in avoiding confusion.

Composite materials include mill composites: clad metals, honeycombs, nonmetallic laminates, and sandwiches produced in more or less standard lines and suitable for many different applications. On the other hand, we shall call "composite structures" those material systems that are designed and produced for a given application and that are also the finished structure, component, or product itself. Examples of composite structures are rocket nose cones (constructed of several integrated layers), tires (built up of several layers and a fabric-reinforced material), glass-reinforced plastic boats, and filament-wound vessels. Although a finished structure is also an integrated materials system, this does not preclude regarding it as a composite material. In general, structural engineers refer to all structures of complex or heterogeneous construction as composite structures.

The nature of any composite depends on the form and structural arrangement of constituents, which may include fibers, particles, flakes, laminae, and fillers. These structural constituents, shown in figure 2 (from ref. 5), determine the internal character of the composite. Since the structural constituent is generally embedded in a continuous matrix of another material, the matrix is called the "body" constituent. It generally encases the structural constituent, holds it in place, seals it from mechanical damage, protects it from environmental deterioration, and gives the composite form. Not all composites, however, have a matrix. Two or more different materials are sometimes bonded together, as in laminates and sandwiches. These layers form the complete composite.

Composite materials are divided into five basic groups by form of the structural constituents. (See fig. 3 from ref. 5.)

(1) Fiber (or fibrous) composites are composed of fibers in continuous or discrete filaments (called whiskers for their appearance in

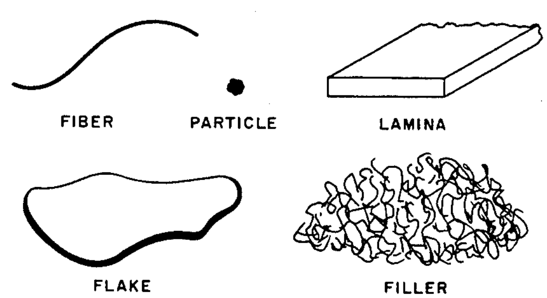


FIGURE 2.—Types of structural constituents. (*Courtesy of* Materials in Design Engineering.)

the production method) embedded in a continuous matrix. Fiber–fiber composites have no matrix.

(2) Particulate composites [1] are composed of minute particles, usually uniformly shaped, embedded in a continuous matrix.

(3) Flake composites are made up of flat particles or flakes, usually of isotropic material held together by an interface binder or embedded in a continuous matrix.

(4) Filled, or skeletal, composites have a continuous three-dimensional constituent which has a random network of open pores or passages, cells, or an ordered honeycomb, filled with another constituent.

(5) Laminar composites are formed by layers of single constituents bonded as superimposed layers.

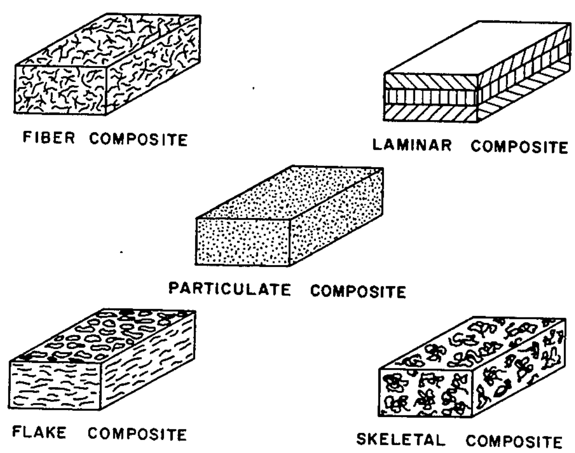


FIGURE 3.—Classification of structural constituents. (*Courtesy of* Materials in Design Engineering.)

Fiber Composites

Fiber composites, particularly the fiber–matrix types, have been of interest to many structural engineers. The forms in which fibers can be employed in composites are numerous and draw on the long experience of textile technology for help and guidance (ref. 6). The simplest and most widely used arrangement is a mat of short fibers

---

[1] Flake and filled composites are sometimes included under particulate composites.

laid down in a random pattern. The material is essentially isotropic in its own plane; however, the strength and elastic modulus are determined by only a small proportion of the total number of fibers, oriented approximately in a certain direction. Highest strength in one direction is achieved when fibers in the form of continuous filaments are laid parallel to each other in a unidirectional pattern. This arrangement produces a high fiber-packed density. Fiber–matrix composites with unidirectional fibers are basically anisotropic. The highest strength is in the direction of the fibers, whereas strength in a transverse direction is essentially that of the matrix. Figure 4 summarizes the orientation, length, shape, and material characteristics of fiber constituents.

Fibers and matrices are available for a wide range of versatile

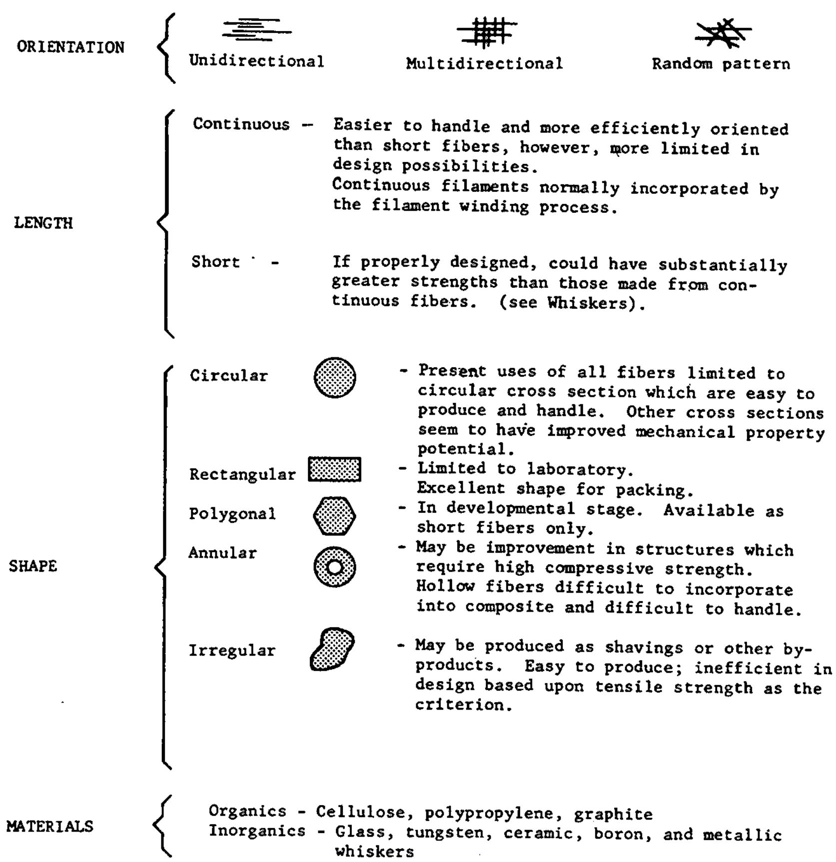


FIGURE 4.—Fibers in composites.

composites. Composites such as glass-fiber-reinforced plastics (GFRP), metal-fiber-reinforced plastics, and asbestos-fiber-reinforced plastics are fiber–synthetic resin combinations. Glass is the most widely used fiber, and synthetic resins such as epoxies, phenolics, and unsaturated polyesters are the most widely used matrices.

Development of fiber composites for high-temperature service has led to the use of high-temperature-resistant fibers in high-modulus metal matrices (refs. 7, 8, and 9). Alumina- and tungsten-fiber-reinforced silver; and carbon-, graphite-, and silica-fiber-reinforced aluminum are examples. Until recently most of the work was done with strong, stiff fibers of solid, circular cross sections in a much weaker, more flexible matrix (such as glass fibers in synthetic resins). At present, there is considerable interest in hollow, metal and ceramic fibers of noncircular cross sections embedded in stronger, stiffer, and more heat-resistant matrices. Although limited quantities of these new composites are available for high-performance applications, insufficient production and high cost restrict their use. More economical fiber composites, such as glass-fiber-reinforced plastics, are now coming into their own for structural applications where high strength and light weight are desirable.

Although the most commonly used fiber composites may be fabricated by various techniques, we shall limit our discussion to the use of continuous filaments for the fabrication of filament-wound structures and of whisker composites.

Filament-Wound Structures

The modern era in composites began with filament-wound plastics used in glass-reinforced structures such as pressure vessels. Extremely high strength-to-weight ratios are achieved. By exploiting the high strength of continuous fibers or filaments embedded in a matrix of a resinous material (either organic or inorganic), the winding technique is used to direct the structural strength. The resin contains the reinforcement, holds it in place, seals it from mechanical damage, and protects it from environmental deterioration. Rovings are drawn through a resin bath and are wound continuously onto a form, or mandrel, that corresponds in shape to the inner structure of the fabricated part. This winding technique permits orientation of structural strength to resist stress from an imposed load. A wide range of properties can be attained, depending on the filament and resin materials, winding patterns, and configuration of products. Suitable shapes of filament-wound structures include surfaces of revolution or combinations of surfaces that are flat or convex. (See fig. 5 from ref. 10.) Figure 6 (from ref. 11) shows various types of winding patterns.

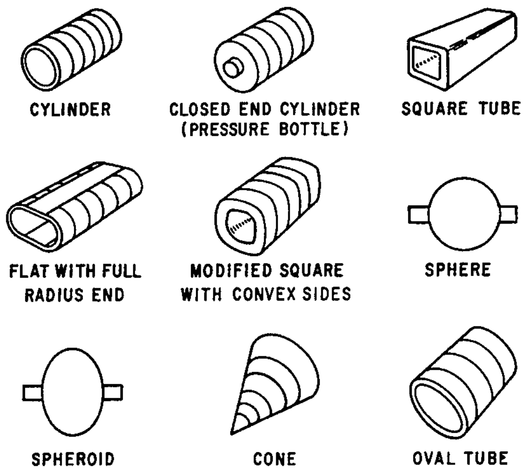


FIGURE 5.—Suitable section shapes for filament winding. (*Courtesy of* Machine Design.)

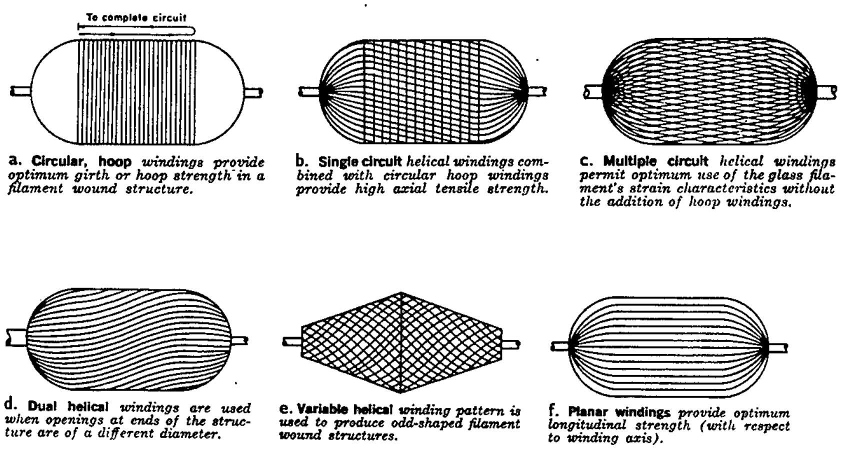


FIGURE 6.—Filament winding patterns. (*Courtesy of* Materials in Design Engineering.)

Whisker Composites

One of the more promising composites under investigation is the whisker composite. Researchers have long been aware of the possibilities of developing materials that nearly approach their theoretical strength. Such materials have been found in the form of small fibers or filamentary microcrystals which are actually fine wisps of material grown like mold cultures. Whiskers are extremely thin (about one-millionth of an inch to a few thousandths of an inch) and have nearly perfect crystal structures, with fewer defects than conventional materials. Consequently, some whiskers have tensile strength above 3 500 000 psi. Their extremely high strength-to-weight and stiffness-to-weight ratios make them particularly attractive reinforcements. Until commercial production is established, however, large quantities of whiskers will not be available for everyday products. One new technique for incorporating whiskers in a metallic matrix is to grow them directly in the metal (ref. 5) by the controlled unidirectional solidification of an alloy.

Among the future applications of whisker composites are whisker-reinforced plastics for bodies and basic structures in automobiles, whisker-reinforced metals for fully cast submarines, and whisker-reinforced concrete.

Analysis of a combination of materials that act in unison or interact is a complex task. The problem is complicated by the presence and interaction of many fibers, different stress levels in these fibers, differences in elastic moduli and Poisson's ratios between fibers and matrix, interaction of fibers and matrix, a boundary layer of indefinite thickness, and variable properties where the fibers and matrix interact (ref. 6). In addition, many matrices behave viscoelastically; that is, their behavior is time dependent. Therefore, a linear relation between stress and strain (Hooke's law) is not valid. Stress concentrations that develop around discontinuities and stress conditions occurring at the ends of fibers and at breaks in fibers also contribute to the difficulties of understanding the behavior of a fiber composite under stress.

For design purposes, a good approximation of fiber-composite properties is often obtained by application of a simple rule that says: the properties of the composite are the sum of properties of the individual components multiplied by their fraction in the total volume. Unfortunately this rule breaks down when the properties are complex functions of fiber geometry, spacing, relative volumes, etc. The determination of properties at right angles to the fiber direction is sometimes based on the weakest link hypothesis. This link is usually the matrix.

Detailed analyses of composites have been given in recent works by Tsai et al. (ref. 12) and Alexander et al. (ref. 13).

Fiber composites are now found in a wide variety of products including nose-cone shields for spacecraft, rocket motor cases, helicopter rotor blades, high-pressure tanks for liquid gases, storage tanks, railway tank cars, automobile fenders, automobile heaters, valves, ball bearings, truck bodies, walls of experimental homes, and concrete forming pans. Table 1 (from ref. 14) lists some present and future uses of fiber composites as primary structural materials.

273-140 O - 67 - 2

TABLE 1.—*Present Development Applications of Whisker Plastic Composites*

Type of composites	Application	Nonproprietary information
Unidirectional: Reinforced plastic wire.	Bulletproof vest, deep-submergence cable.	Proven alinement, 19 million modulus, yarn formation.
Bidirectional: Reinforced plastic laminate.	Space helmets, turbine blades, high-speed centrifuges.	10 million modulus.
Tridirectional:		
Reinforced casting resin.	Dental research [a]	2× better than any other reinforcement.
Supplementary reinforcement—Transfer molding compounds.	Miniature rockets, electronic micromolding.	1.5 v/o addition=20 percent increase in tensile burst strength.
Interstitial reinforced Fiberglas.	Filament-wound deep submergence vessels.	0.65 v/o addition=38 percent increase in interlaminar shear.

[a] Research on reinforcing dental plastic, metal, and ceramic fillings.

Particulate Composites

Particulate composites differ from the fiber and flake types in that the distribution of the particulate constituent is generally random rather than controlled. In some of these composites, the particulate constituent becomes dimensional only on the microscopic scale. These discrete particles are contiguous but insoluble and chemically unreactive with the matrix. The particulate constituent generally contributes strongly to the properties of the composite.

Particulate composites may be divided into several classes: (1) metal in metal, (2) metal in plastic, (3) metal in ceramic (including cermets), (4) organic in organic, (5) nonmetallic in nonmetallic, (6) dispersion-hardened alloys, and (7) self-lubricating alloys. Each of these composite classes is an individual subject beyond the scope of the present treatment, but examples may be found in reference 5. Only three specific types of particulate composites will be discussed here. These are (1) cermets, (2) dispersion-hardened alloys, and (3) self-lubricating alloys.

A cermet is a composite in which ceramic grains are held in a metal matrix in amounts up to 30 percent of the total volume. Cermets are among the most important composites and have a range of properties dependent on the composition and relative volumes of the ceramic and metal constituents. Of the various possible combinations, carbide-based and oxide-based composites are among the most widely used. Examples of carbide-based cermets include tungsten carbide, chromium carbide, and titanium carbide; examples of oxide-based cermets include aluminum oxide and magnesium oxide in chromium.

Dispersion-hardened alloy composites consist of hard, submicron-sized particles dispersed in a softer metal matrix; the particles are usually less than 3 percent by volume. These composites differ from cermets in the smaller size of the particles and by their lower proportion of concentration. Although both the size and the proportion of the total volume of the dispersed particles are small, these particles control the strength properties of the composite. The finer the particle size and the smaller the spacing, the better, generally, the properties (ref. 6).

Self-lubricating alloys are a recent development based on the dispersion of dry lubricant powders in a metallic matrix. These composites include combinations such as molybdenum disulfide or tungsten disulfide in nickel, boron nitride or calcium fluoride in steel, and tungsten diselenide in copper or silver.

Concrete is perhaps the oldest particulate composite. Most recent advances in particulate composites, however, have been related principally to the aircraft industry and the nuclear field. Aircraft builders have used sintered aluminum powder alloys (SAP) for impellers and pistons, and in the nuclear field, cermets are used for applications such as valve seats and bearings. Other structural uses of cermets include high-speed cutting tools, integral turbine wheels, and nozzles. Structural uses of steel-particle-filled plastics include small-lot production tooling. More exacting industrial demands for heat- and oxidation-resistant materials should lead to increased use of metal-in-ceramic particulate composites and the use of metals and nonmetallics in plastics.

By treating the particles as inclusions and applying the theory of stress concentrations (around discontinuities of various shapes), it may be theoretically possible to analyze the resulting structure. Some work has been done and solutions have been obtained for a circular particle embedded in an infinite matrix. As greater numbers of inclusions are considered, this approach becomes impractical, however, and composites consisting of small particles randomly distributed are generally treated as conventional materials.

Flake Composites

Flake composites are still in the development stage. Recently they have received considerable attention for structural use where two-dimensional elements are preferable. The considerable overlap between flakes in the composite can result in an effective barrier against fluid penetration into the matrix as well as reduce the danger of mechanical penetration.

When flakes are embedded in a matrix and made parallel to one another in a plane, they give uniform properties to the composite; however, parallel orientation is difficult to achieve. Flake composites also have special properties due, in part, to their flat shape. They can be packed more tightly than other shapes, thus providing a high percentage of reinforcing material for a given cross section.

Although flakes have good bulk-handling qualities and are relatively inexpensive to produce, desired shapes and sizes are often difficult to achieve. Most metal flakes are aluminum; some silver is used because other metal flakes are difficult to produce. Aluminum flakes can reflect heat, provide a series of protective coatings, and give a metallic luster to coatings and plastic moldings.

Silver flakes are more applicable for composites when high conductivity is needed. For electrical conductivity, flake-to-flake contact is necessary, but this is not easy to achieve without losing good bonding qualities. Silver-flake composites are used for conductive-coating electrical heating elements in equipment subject to low-temperature environments.

Though nonmetallic flakes include both mica and glass, mica is more widely used because of its good heat resistance and dielectric qualities. Mica flakes are not as versatile as glass flakes for moisture barrier and structural applications. With certain binders, however, they can provide a good hermetic seal with vacuum tightness and high-temperature properties. If contoured shapes are desired, flakes can be laminated in several layers, bonded with a resin (such as 5 percent shellac, epoxy, or alkyd), heated until the composite softens, and molded into various shapes.

Glass-flake composites with special strength qualities have been proposed for a variety of structural and nonstructural applications. Less expensive than glass fibers, these flakes have dielectric strength, high heat resistance, and offer a high bending modulus because they are free to bend in only one plane. Structural uses of glass-flake composites include aircraft and missile radomes, battery cases, instrument cases, rocket fins, and rocket motor cases.

Once fabricating difficulties are solved, flake composites will offer attractive possibilities.

As in the case of particulate composites, flakes can be treated as inclusions, and the basic theory of stress concentrations can be applied for general analysis. The practical utility of this approach, however, is extremely limited.

Filled Composites

A filled composite is an open or skeletal matrix containing another material that remains a separate constituent. Both materials and structure can be varied to give a wide range of physical and mechanical properties. Some examples (ref. 5) of what can be done are: (1) improve the function characteristic, (2) increase the strength and ductility of porous metals, (3) prevent leakage of fluids and gases in porous metals, (4) improve the performance of electric contacts, and (5) provide high-temperature properties not obtainable with monolithic materials.

The skeleton may be a solid or a fluid becoming solid during manufacture. The filler may be fluid when introduced into the void structure and may either remain liquid or solidify by cooling. The fluid may be introduced into the structure by a carrier that may be removed, evaporated, or polymerized.

Although the concept of filled composites is old, the full potential of this type of composite was not apparent until the space age. Previously, this technique was used to prevent seepage by filling voids in one material with another. Currently, two types of filled composites are available: (1) porous, or spongelike, structure, and (2) cellular, or honeycomb, structure. Metal, paper, or wax honeycomb filled with a ceramic material; silica honeycomb filled with a ceramic material; and silica honeycomb filled with fiber-reinforced epoxy and silicone rubber are examples of filled cellular composites. Examples of filled porous composites include tungsten or molybdenum impregnated with copper or silver; plastic filled with porous metal such as aluminum, magnesium, or ferrous castings; TFE fluorocarbon and lead-filled bronze; graphite impregnated with lubricating oils; and resin impregnated with aluminum, ceramic, or zirconia foams.

The range of possible uses of filled composites is not fully known. Two current applications are a steel matrix composite containing titanium carbide for making dies, gages, and punches; and a filled honeycomb composite with random network filler metal (ref. 6) for high-temperature aerospace applications.

Laminar Composites

Laminar, or layered, composites are probably the oldest of all the composites; they account for the greatest volume of use and are produced in the greatest variety. They differ in material, form, and/or orientation, as shown in figure 7. Although the great variety of possible combinations makes generalization difficult, we can say that (1) each layer of a laminar composite may perform a separate and distinct function, (2) properties may vary from one side of the composite to the other, and (3) properties of laminar composites tend to be anisotropic.

Many laminar composites are designed to provide characteristics other than strength; for example, improved appearance, protection against corrosion and/or high temperatures, and adjustment for size limitations.

Some types of laminates are: precoated and preplated materials, clad metals, plastic-based laminates, laminated glass, and laminated nylon fabrics. Specific examples of these laminates are alclads; nickel-plated steel for flashlight cases; electrogalvanized steel for roofings; aluminum-clad uranium, molybdenum-clad aluminum (copper, gold, or lead), and silver-clad aluminum for weave guides; lead-clad steel for radiation shielding; glass-nylon fabric for personnel armor; glass-plastic (usually containing two or more layers of glass sheet and one or more layers of polyvinyl butyrol) for safety glass; laminated layers of transparent plastic for light filters; and asbestos or other mineral-based fabrics laminated with silicone matrices for heavy-duty electrical and high-temperature application.

The analysis of laminates can be lengthy and arduous because of the possible combinations of several different isotropic and/or anisotropic materials. Stress distributions depend on the composition and orientation of the individual layers. A laminate may be isotropic, but if its constituents exhibit different moduli of elasticity or different Poisson's ratios, appreciable nonuniform shear stresses may develop and lead to delamination or other serious problems. Consequently, laminates composed of mixed isotropic and anisotropic materials present a difficult analytical problem.

Stress analysis of a laminate subjected to external loads is based on the fundamental assumption that at any point the deformation of all the constituents is the same; that is, strains are equal. Thus, a laminate made of isotropic materials having the same elastic constants behaves like a solid mass of the same material. If the elastic constants are different, the stresses in the constituents are proportional to the elastic constants (elastic or shear moduli). However, even in directions in which no load is applied, stresses in the individual layers may be produced by different transverse contractions or expansions because of different Poisson's ratios; these tendencies are prevented by the bond between layers. Both transverse shears are produced as well as shear in the bond between layers. When isotropic and orthotropic materials are combined, large differential stresses in the laminate as well as large shears in the bonds between layers may develop. Changes in temperature may similarly cause complex stresses (ref. 6). Furthermore, when a layer of material is thin, as in most laminates, the thickness direction is often ignored; even an orthotropic material is treated as a substance that has two elastic moduli, one shear modulus, and two Poisson's ratios associated with the natural axes. For analytical approaches to laminates, the reader is referred to Tsai et al. (refs. 12 and 15), Dong et al. (ref. 16), and the bibliography.

Sandwich Construction

Sandwich construction[2] is a special kind of laminate consisting of a thick core of weak, lightweight material sandwiched between two thin layers (called "face sheets") of strong material (fig. 7). This is done to improve structural strength without a corresponding increase in weight; that is, to produce high strength-to-weight ratios. The choice of face sheet and core materials depends heavily on the performance of the materials in the intended operational environment.

Sandwich composites are often compared to an I-beam with a high section modulus. Because of the separation of the core, face sheets can develop very high bending stresses. The core stabilizes the face sheets and develops the required shear strength. Like the web of a beam, the core carries shear stresses. Unlike the web, however, the core maintains continuous support for the face sheets. The core must be rigid enough perpendicularly to the face sheets to prevent crushing, and its shear rigidity must be sufficient to prevent appreciable shearing deformations. Although a sandwich composite never has a shearing rigidity as great as that of a solid piece of face-sheet material, very stiff and light structures can be made from properly designed sandwich composites.

A useful classification of sandwich composites according to their core properties by respective direction is shown in figure 8 (from ref. 17). To see the core effect upon sandwich strength, let us consider the honeycomb-core and the truss-core sandwich composite. The honeycomb sandwich has a ratio of shear rigidities in the xz and yz

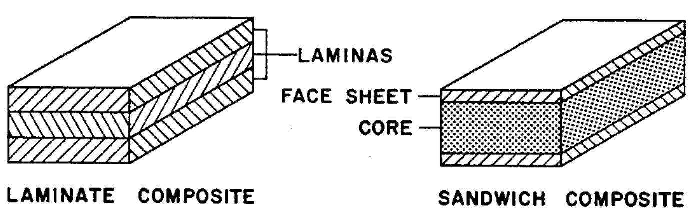


FIGURE 7.—Laminar composites.

---

[2] The ASTM definition: A structural sandwich is a construction combining alternating, dissimilar simple or composite materials, assembled and intimately fixed in relation to each other so as to use the properties of each for specific structural advantages in the whole assembly.

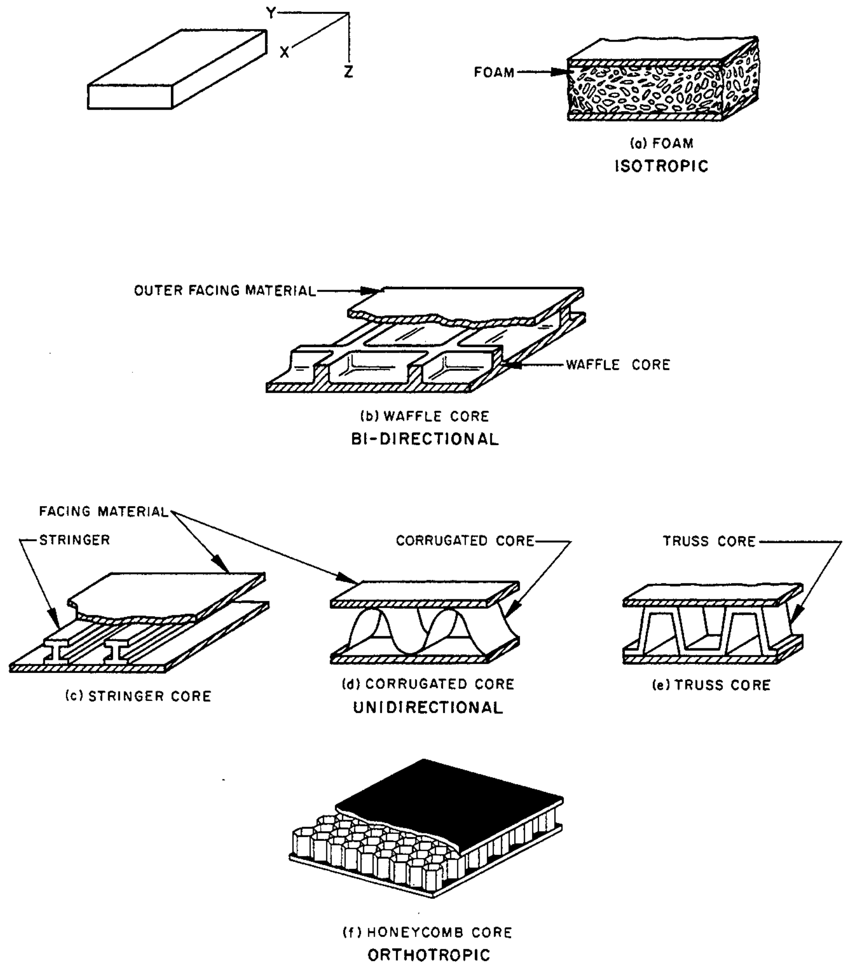


FIGURE 8.—Typical sandwich constructions.

planes of approximately 2½ to 1. The face sheets carry in-plane compressive and tensile loads, whereas the core stabilizes the sheets and builds up the sandwich section. The truss-core sandwich has a shear rigidity ratio of approximately 20 to 1. It can carry axial loads in the direction of the core orientation as well as perform its primary function of stabilizing the face sheets and building up the sandwich section.

CURRENT APPLICATIONS

Figures 9 (from ref. 11), 10, 11, and 12 suggest a few of the many uses for composites. Prior to World War II, sandwich construction of birch face sheets and balsa core was used extensively in the British

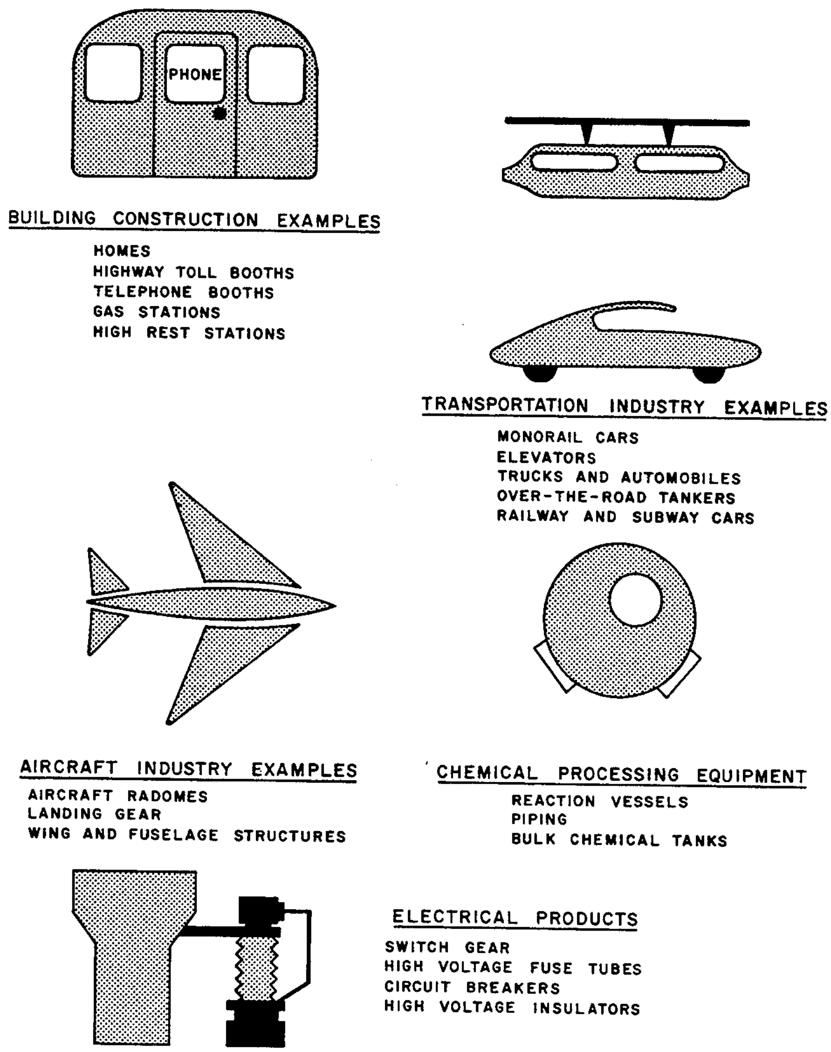


FIGURE 9.—Potential applications of filament winding. (*Courtesy of* Materials in Design Engineering.)

De Havilland Mosquito bomber. Since then a continually growing list of sandwich applications in aircraft includes radomes, fuselages and wings, ailerons, floor panels, and storage and pressure tanks. Because of their dielectric properties, plastic, glass, and fabric honeycomb-core sandwiches are being extensively investigated for use in things such as radar housings and microwave transmission windows. The novel, though costly, design of the B–70 features use of machine-fusion welding for joining brazed, heat-treated, steel-

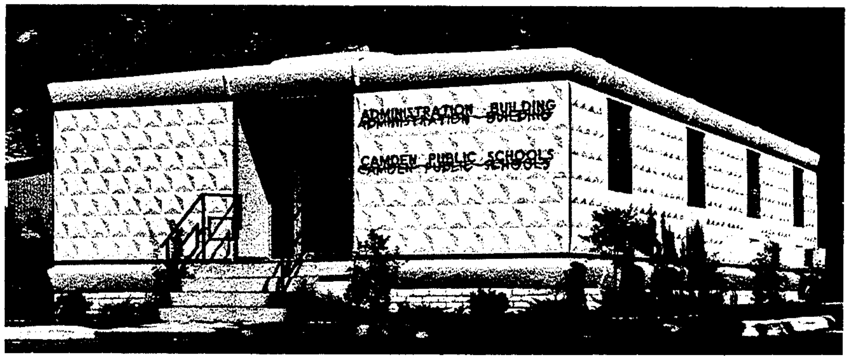


FIGURE 10.—Portable, permanent building, 26 by 56 ft. Exterior is white fiber glass with rigid foam insulation. (*Courtesy of Holiday Manufacturing Co., Division of Holiday Inns of America, Inc.*)

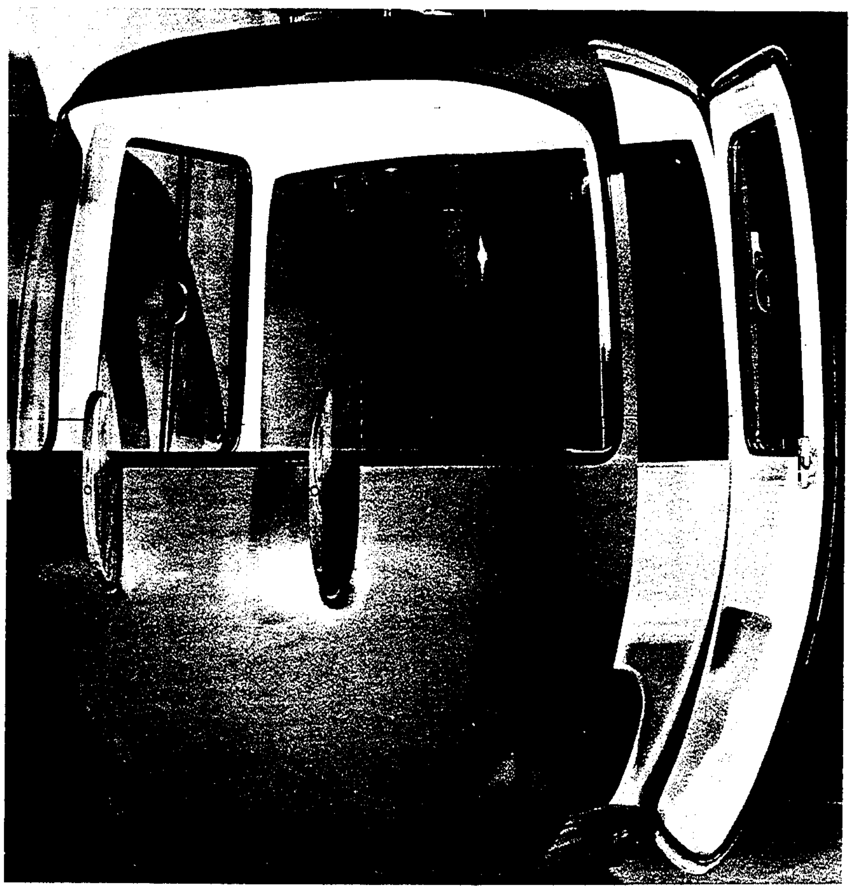


FIGURE 11.—Structural frame for ski lift gondola. (*Courtesy of American Cyanamid Co.*)

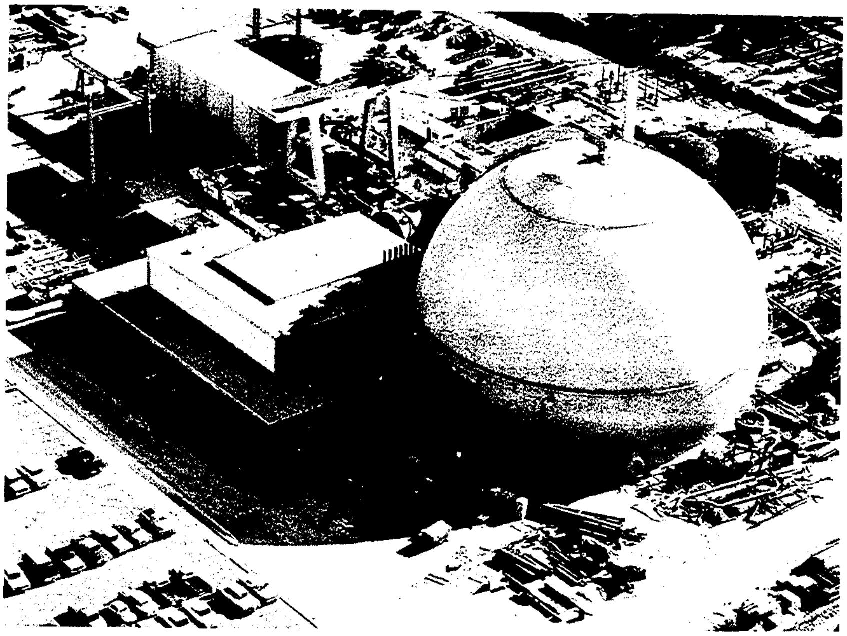


FIGURE 12.—Multilayer wall tank. (*Reprinted from* Oil and Gas Journal, *Jan. 3, 1967.*)

honeycomb panels into a homogeneous, compression-resistant structure with local stresses and shear flow based on minimum strain energy. In a new space-formed system called "Sunflower," the reflector is of honeycomb construction, having a thin coating of pure aluminum protected by a thin coating of silicon oxide to give the very high reflectivity needed for solar-energy collection. The unit is adaptable as a power source for a wide range of Earth, Moon, Mars, and Venus missions. Thirty panels fold together into a nose-cone package in the launch vehicle.

Building Construction

Architects use sandwich construction made of a variety of materials for walls, ceilings, floor panels, and roofing. Cores for building materials include urethane foam (slab or foam-in-place), polystyrene foam (board or mold), phenolic foam, phenolic-impregnated paper honeycomb, woven fabrics (glass, nylon, silk, metal, etc.), balsawood, plywood, metal honeycomb, aluminum, and ethylene copolymer foam. Facing sheets can be made from rigid vinyl sheeting (flat or corrugated); glass-reinforced, acrylic-modified polyester; acrylic sheeting; plywood; hardwood; sheet metal (aluminum or steel); glass-reinforced epoxy; decorative laminate; gypsum; asbestos; and poured concrete.

Civil Engineering

Sandwich construction is now used in bridge decking (stainless-steel truss core), retaining walls, and storage tanks. Structural aluminum honeycomb panels have been used in the construction of the 327-ft-high, 7-million-lb service tower for the Saturn rocket at Cape Kennedy.

Mechanical Design

Foam-core sandwiches have a promising potential in refrigeration and related fields. Early projects involved panels having an expanded styrene core bonded to styrene sheet faces. More recently, two pilot models have been constructed with anodized aluminum faces which offer important cost advantages in tooling. Presently this construction is used in high-temperature furnaces as well as pressure vessels and tanks, particularly cryogenic tanks.

Damped Structures

An increasing number of vibration problems must be controlled by damping resonant response. By using a symmetric sandwich panel with a viscoelastic core, various degrees of damping can be achieved, depending on the core material properties, core thickness, and wavelength of the vibration mode.

Marine Structures

Urethane, expanded styrene, and other types of plastic foam have been used in the construction of small boats. For the same purpose the U.S. Rubber Co. has developed a configuration consisting of an expanded Royalite ABS core with a ply of hard Royalite bonded to each side.

Transportation

Sandwich construction has great promise for transit cases, floor panels, railway cars, and large transportation carrier panels. It has already been used in the design of carriers for experimental rapid transit systems.

Sandwich construction has undergone analytical and experimental investigations that have resulted in a wealth of data. Plantema (ref. 17) has summarized the theory of strength and stability of sandwich-type structures.

The objective of many of the present investigations is to establish rational guidelines for the design and utilization of composite materials for structural applications. More accurate methods of structural analysis are needed to insure the efficient utilization of composites.

Which composite material should be used in a specific design problem? Unfortunately, there is no simple, direct answer. Many important factors contribute to the characteristics, properties, and overall performance of composites. To make the best use of composites, we must be able to characterize each desired property and to assess quantitatively the behavior of the resulting composite. To use composites in design, structural engineers need mathematical expressions and models that adequately predict the behavior of composite materials under various load conditions.

We have briefly examined some of the developments and advanced concepts in materials and types of construction stemming from space-vehicle system research and technology programs. As the deficiencies in our knowledge of the behavior of structures are removed, we can come closer to rational design per se. As production reliability and analytical competence are improved, the behavior of structural elements under various loading will become more predictable. Finally, when production costs are reduced, more extensive industrial use of these developments can be expected.

CHAPTER 3

Selection of Structural Materials and Types

No single known material or construction can meet all the performance requirements of modern structures. Selection of the optimum structural type and material requires systematic evaluation of several possibilities. The primary objective often is to select the most efficient material and configuration for minimum-weight design.

In figure 13 (from ref. 18), materials are plotted according to their strength-to-density ratios. The most commonly used structural materials are clustered in the middle of the figure; those for specialized use (such as fabrics, fiber glass, laminates, and beryllium) are widely dispersed in the lower half of the figure; and the emerging composites containing S-glass, boron, and carbon filaments appear in the upper half. Properties of composite materials are calculated for a planar isotropic layer of filaments in an epoxy-resin matrix. Although this approach is convenient, it is not appropriate for all structures.

Figure 14 (from ref. 19) shows the lightest forms of construction that will carry pressure and bending loads without cylinder buckling or material yielding. The forms considered are filament wound, sandwich, stiffened skin, and simple isotropic walls. Regions to the right of the indicated boundary (slant line) in the chart are those of higher bending moment; that is, those where structures must support compressive and shear stresses efficiently. Consequently, material stiffness is an important requirement. While sandwich constructions of various types have a wide range of application for most loading parameters, the extremes of pressure and bending moment require filament-wound and isotropic constructions, respectively. Filament-wound construction is superior to other structural types for pressure-vessel applications, except when the applied bending-moment index is also large; then the conventional isotropic shell is lighter. For lower or zero internal pressure, all types of wall construction have a range of efficient application, depending on the magnitude of the bending-moment index.

Figure 15 (from ref. 20) is a weight comparison of several types of construction for cylinders as a function of loading intensity for

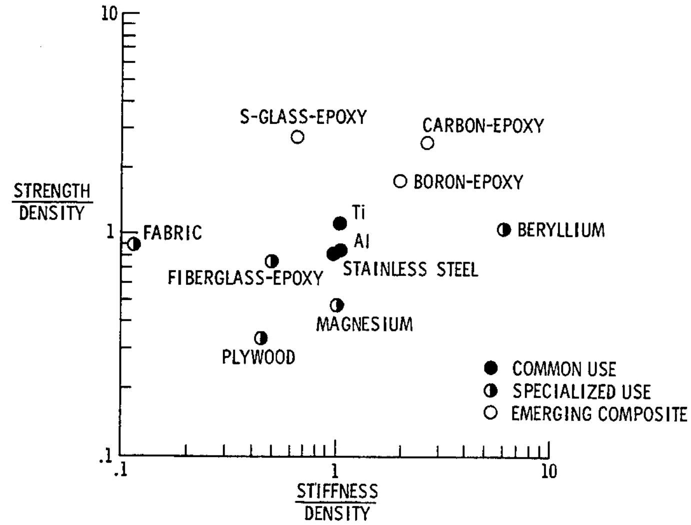


FIGURE 13.—Materials comparison. (*Reprinted from Proceedings AIAA/ASME 7th Structures and Materials Conference.*) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

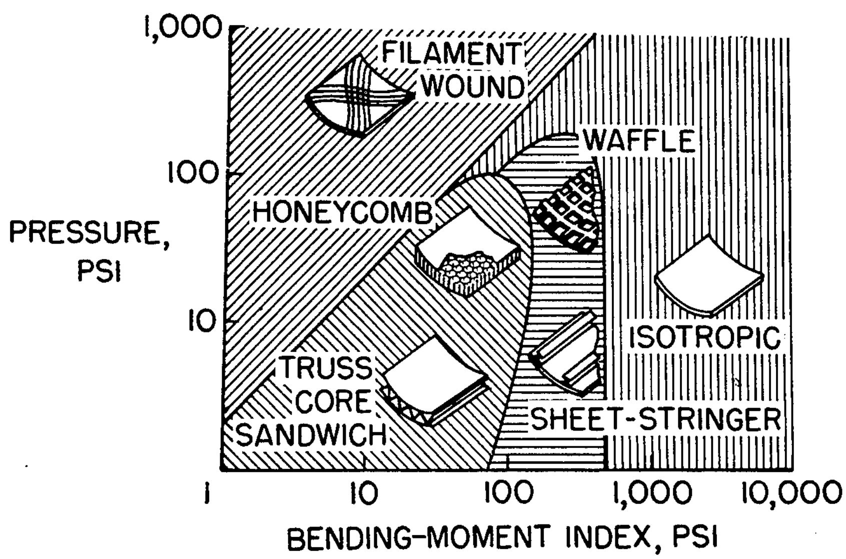


FIGURE 14.—Least-weight construction regimes. (*Reprinted from* Astronautics and Aeronautics.) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

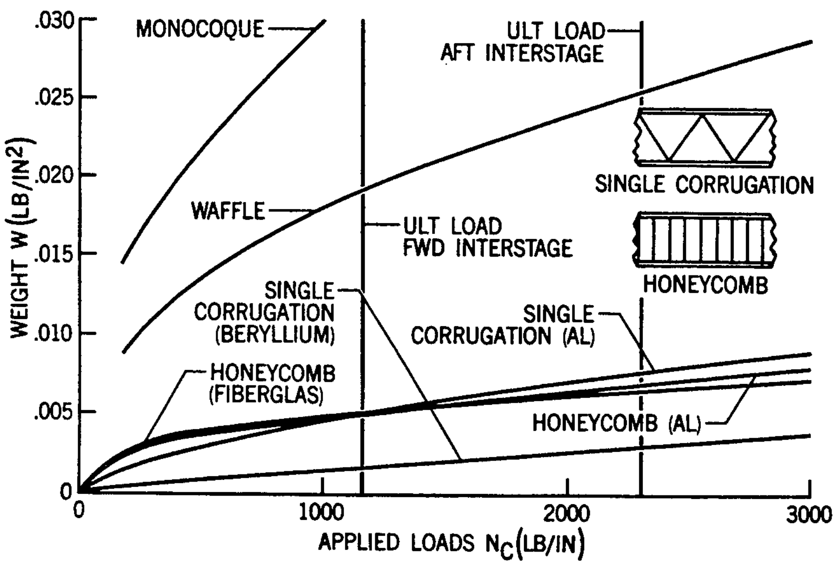


FIGURE 15.—Comparison of minimum-weight designs for structural systems and materials in an axially loaded cylinder.

designs involving combined axial compression and bending. Figure 16 (from ref. 21) presents data for 10-ft-diameter cylinders with local compression load capability of 400–1600 lb/in., which is within the appropriate range for the early orbiting laboratories. Four types of construction were considered: sheet-stringer (stiffened skin), waffle, corrugated truss-core sandwich, and honeycomb sandwich. A preliminary estimate of wall gages required for these load levels indicated that integrally milled waffle plate could be eliminated from the study. In theoretical axial strength alone, the honeycomb wall is the lightest, followed by truss core and sheet stringer. Since weight comparisons are very close, they are very sensitive to design ingenuity.

Figure 17 (from ref. 22) shows a comparison of the structural index curves for variously constructed cylindrical shells subject to axial compression. The curves include isotropic, Z-stiffened, waffle, truss-core sandwich, and multiwall sandwich cylinders. The multiwall sandwich consists of layers of dimpled and flat sheets welded at the crests of the dimpled sheets. Structural index curves for flat plates of various construction subjected to edge compression are shown in figure 18 (from ref. 22). The curves shown in both figures are based on formulae given in reference 22 and citations made in this reference. The curves

273-140 O - 67 - 3

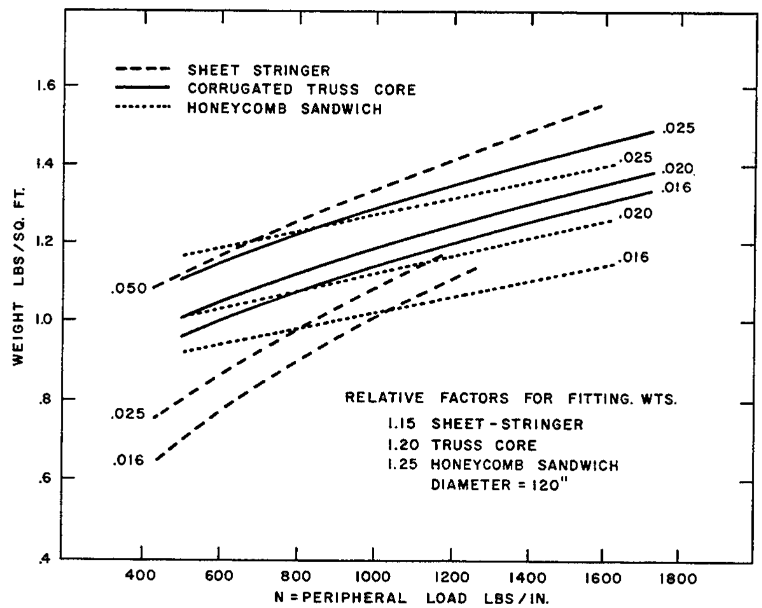


FIGURE 16.—Compartment wall weights.

$$\frac{E}{\sigma_{cy}}=115 \quad \text{and} \quad \frac{E}{\sigma_{cy}}=153$$

are related to the original report and are included here merely for reference. P_1 is compressive load per unit width; N_x, compressive buckling load per unit circumference in the longitudinal direction; b, width of plate; R, radius of cylinder; $\bar{t}$, equivalent thickness of material; E, modulus of elasticity; ρ_c, density of core; ρ_t, density of face sheets; η, plasticity reduction factor; and σ_{cy}, compressive yield stress.

One of the major reasons for combining inorganic fibers and inorganic matrices is to achieve high-temperature performance not possible with organic materials. One of the new and promising composites under investigation is metal reinforced with alumina whiskers. Figure 19 (from ref. 5) shows the tensile strength of various classes of whisker-reinforced materials at elevated temperatures. So far, strengthening metals with whiskers at elevated temperatures has only been demonstrated in the laboratory.

Theoretical whisker-plastic laminates are compared with present bidirectional materials in figure 20 (from ref. 14). The calculated strength is based on the assumption that the whiskers are nonwoven, biaxial, felt impregnated with epoxy resin, and high pressure molded.

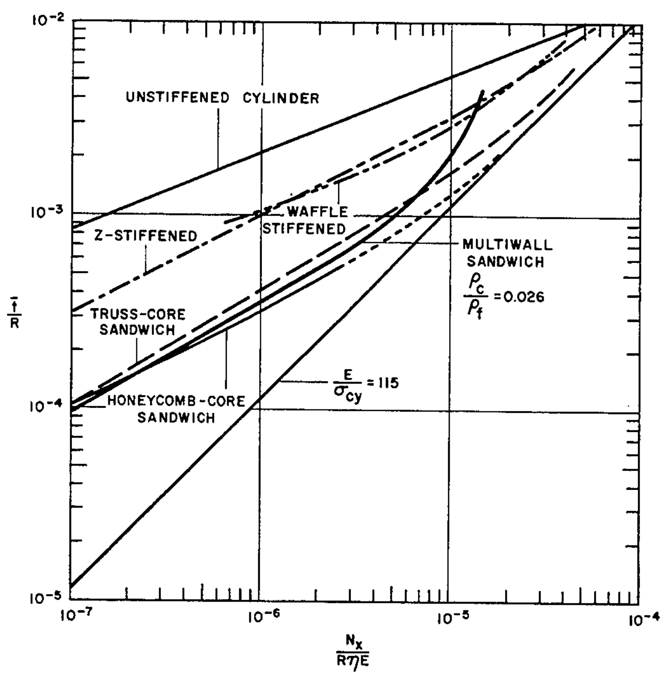


FIGURE 17.—Structural index curves for cylindrical shells of various constructions subjected to axial compression.

Under these assumptions, the curves show that the whisker composite would offer an order-of-magnitude increase in strength.

Figure 21 (from ref. 23) summarizes the evaluation for pressure vessels. On the horizontal scale, the uniaxial material tensile-strength–weight ratios are indicated. For glass filaments, this ratio is based on the strength of the rovings. The vertical scale represents the uniaxial structural strength-to-weight ratios that can be achieved by application of the best current technology. Although filamentary composites appear to be superior to monolithic construction, it must be noted that the configuration efficiency coefficient must also be considered when evaluating overall pressure-vessel efficiencies. Consequently, figure 21 does not permit a direct comparison of the relative efficiencies of materials as used in pressure vessels. The values in figure 21 are based on short-time-load applications at room temperature; cryogenic and elevated temperatures and other environmental factors can change the relative efficiencies of monolithic and filamentary materials substantially.

Figure 22 (from ref. 23) depicts the overall efficiencies of monolithic and filamentary materials for membrane-type pressure vessels.

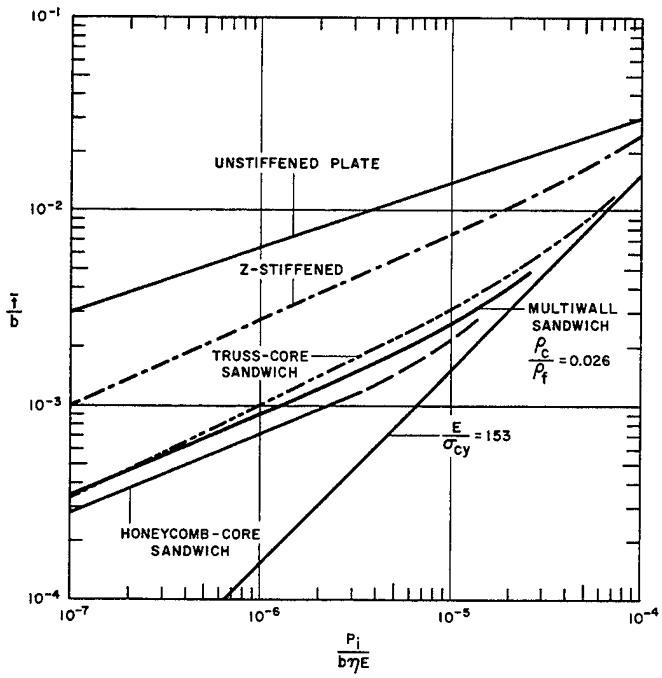


FIGURE 18.—Structural index curves for flat plates of various constructions subjected to edge compression.

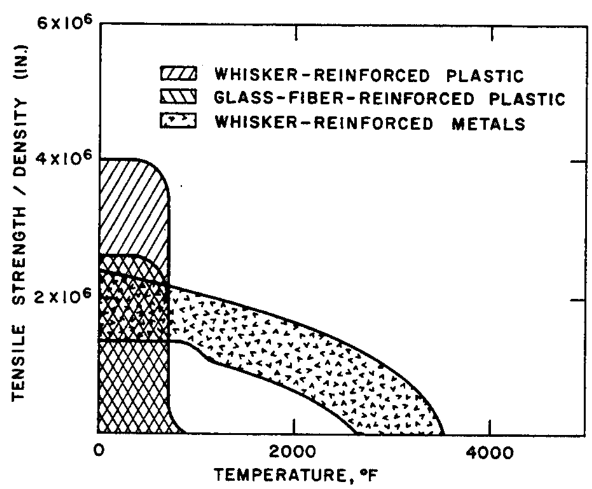


FIGURE 19.—Strength versus temperature for various fiber composites. (*Courtesy of* Materials in Design Engineering.)

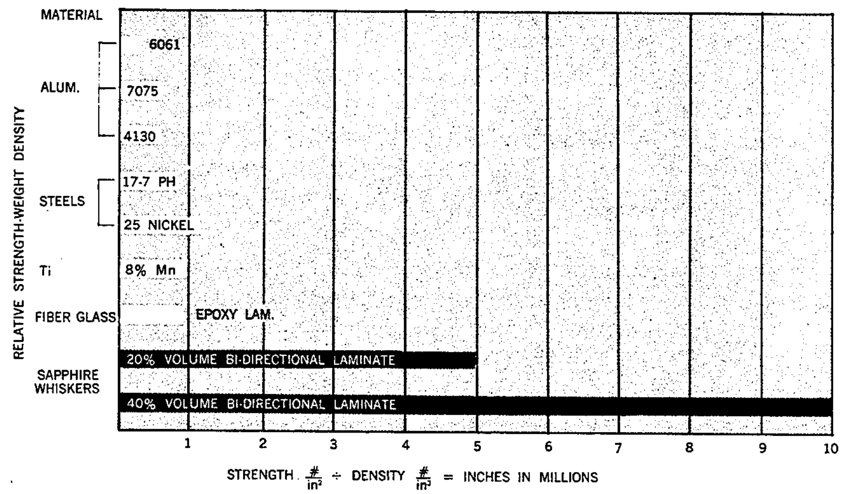


FIGURE 20.—Strength density of various bidirectional structural materials. (*Reprinted by permission of the publisher, F. D. Thompson Publications, Inc., from* Research/Development Magazine, *March 1966.*)

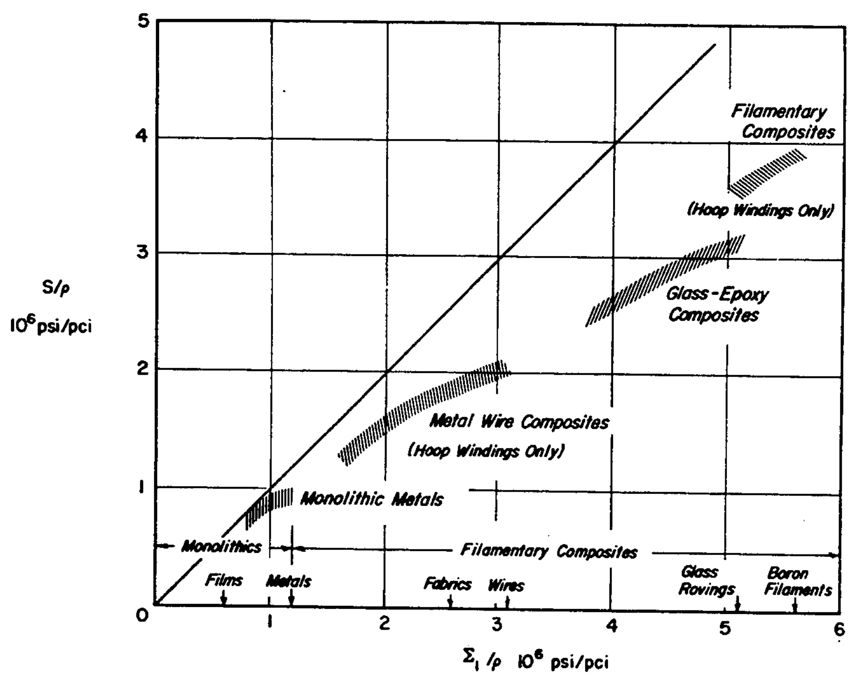


FIGURE 21.—Comparative structural efficiencies of various materials in pressure vessel applications at room temperature.

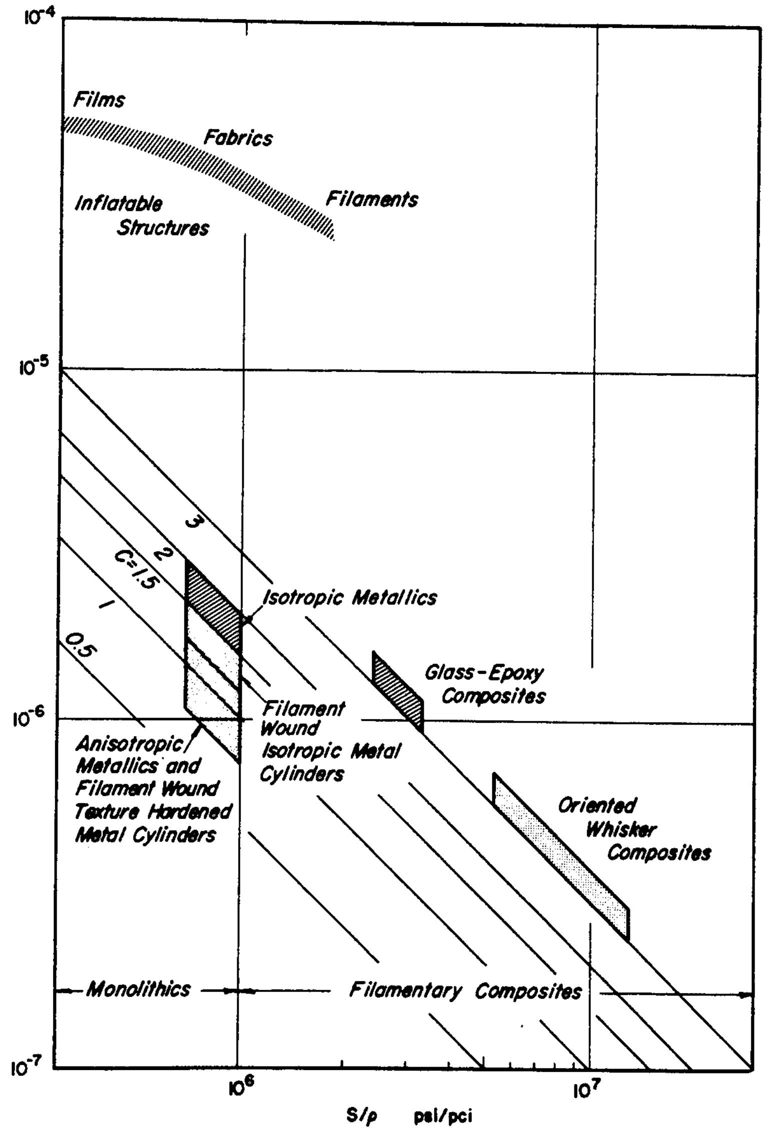


FIGURE 22.—Overall membrane efficiencies of pressure vessels at room temperature.

It is based on two efficiency factors: (1) structural efficiency coefficient, C, and (2) material efficiency parameters, (S/ρ); where C is a nondimensional function of the configuration and material-failure law; ρ, the density in pci; and S, the structural strength in psi. The crosshatched regions in figure 22 represent materials that have been utilized in full-scale aerospace-production components. One should remember that the aerospace environment encompasses temperatures other than room temperature, on which figure 22 is based.

The data of Brewer and Jeppeson (ref. 24) indicate that inflatable structures as a class are inherently much less efficient than metallic and glass-epoxy composites. Isotropic metallics are not as efficient as glass-epoxy composites, when properties are compared at room temperature. Under the best circumstances for each, a weight-saving potential of approximately one-third can be attained with the glass–epoxy composite.

For other materials concepts that have not, as yet, reached the aerospace production stage, filament-wound isotropic metal cylinders represent an inherent improvement over monolithic isotropic metallics. At room temperature, however, the glass-epoxy composites still appear to have an advantage. On the other hand, anisotropic metals, as opposed to currently used materials, can represent a significant weight-saving potential. This potential depends strongly on the degree of anisotropy that can be achieved with high-strength metals and the configuration of the pressure vessel. This is also true for filament-wound, texture-hardened metal cylinders.

An important improvement in overall efficiency appears possible with oriented whisker composites. However, on the basis of the analysis used herein, the potential of such composites appears to be far less dramatic than predicted by Hoffman (ref. 25). In fact, only the low-density whiskers, such as graphite and aluminum oxide, appear to be attractive when used in the form of oriented whisker composites.

In selecting a material and design for specific performance requirements, each configuration (and each part of the configuration) must be examined and analyzed to provide the best possible structure. Ultimately, materials design will be integrated into structural design as an added dimension. Since the selection of the configuration requires consideration of the environment, rigidity requirements, fabricability, smoothness, and reliability, a detailed analysis is needed to provide a valid basis for selection. In chapter 5, we will consider the interplay among design, structures, and materials as well as the general aspects of design synthesis and optimization.

CHAPTER 4

Structural Concepts and Applications

In chapter 2, we emphasized advances in strengthening materials for structural applications that have resulted in part from aerospace requirements. We now turn to some of the recent developments in structural concepts, their uses, and general types of construction.

LAMINATION

Structural types may be used singly or in combinations, depending on the functional requirements of the object to be constructed. For example, for ordinary performance a pressure vessel may be made from a monolithic material, but when weight is a critical factor, it can be made from a filament-wound design. As components of liquid-hydrogen flight vehicles, vessels must withstand extremely high temperatures for long periods of time without serious loss of structural integrity. Composite laminates permit multifunctional constructive systems that have this capability. This new structural concept involves layers of either monolithic or composite materials. Three examples developed for application in reusable structures are: hot monocoque, insulated, and multiwall designs.

Hot Monocoque

Figure 23 shows a hot-monocoque structure for a hydrogen tank which operates near equilibrium temperature and supports applied load. The interior systems, consisting of an aluminum waffle-plate tank with reinforcing rings, are isolated from the exterior load-bearing, or primary, structure by insulation and by a carbon dioxide purge system. Panels of fibrous insulation are bound to the outside, with carbon dioxide filling the voids in the insulation between the tank and the outer structure. The primary structure is a corrugation-stiffened panel of a high-temperature superalloy with transverse rings for additional support.

Insulated Design

The insulated structure concept seen in figure 24(a) is composed of a superalloy heat shield for temperatures up to 1800° F, fibrous high-temperature insulation, a primary structure, cryogenic insulation, and a fuel-tank structure. The temperature of the primary structure is

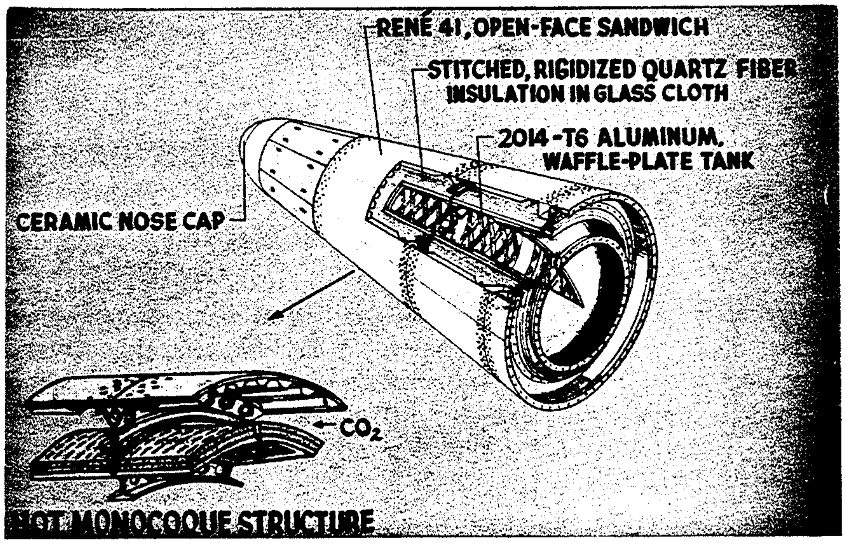


FIGURE 23.—Structural research model.

partly dependent on the thickness of the cryogenic and fibrous insulations. This type of construction requires essentially three leaktight shells: (1) the internal hydrogen tank, (2) the primary structure which precludes liquefaction of air that enters the cryogenic insulation area, and (3) the heat shield which prevents trapping and freezing of moisture within the fibrous insulation area.

Multiwall Design

The multiwall design, shown in figure 24(*b*), is unique because the thermal-protection and load-carrying functions are performed by one integral component. The design consists of a sandwich of alternating layers of flat and dimpled sheets joined by welds at the dimples. The insulating effect is produced by the multilayer reflective sheets when the spaces between these layers are evacuated. The inner layers form both the primary load-carrying structure and the tank wall. Because large temperature differences through the wall thickness are a major problem, the potential of this concept is limited, first, by manufacturing difficulties and, second, by possible thermal stresses inherent in its complex design. A multifunctional, multilayer laminate, nevertheless, has been successfully used in a rocket-nozzle design to withstand 6800° F.

Laminates have also been used in filters, printed circuitboards, and skis.

Several layers of felts or other fibrous materials can be bonded by

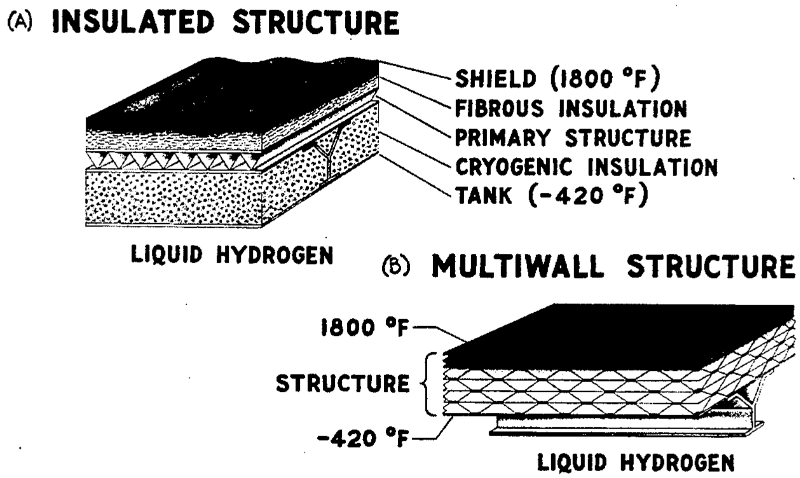


FIGURE 24.—Cryogenic tankage for hypersonic aircraft.

interlocking the fibers. Furthermore, layers of different fiber systems with varying pore sizes, densities, and thicknesses can be bonded together to form a filter laminate, in which each layer can separate particles by specific sizes. Recently a printed circuitboard consisting of a layer of silicone rubber bonded between two layers of glass-reinforced-epoxy laminates was introduced. The glass–epoxy layers are clad with copper to provide good electrical conductivity; the silicone rubber gives damping power; finally, the glass–epoxy adds strength, rigidity, and insulating properties.

A new ski design uses a seven-layer laminate shown in figure 25. After a layer of wood-particle board is bonded between two aluminum strips, the aluminum strips are bonded to two strips of high carbon steel to provide camber and flexure. Lastly, cotton fabric layers are applied to the aluminum to increase its bond strength to the wood and the steel. The top, bottom, and sides are each bonded to a layer of phenolic plastic.

Many other design problems can be solved with plastic laminates bonded to organic or inorganic materials. Potential advantages of choosing materials for specific purposes include: better strength-to-weight ratios, increased rigidity and strength for soft sealing materials, dimensional stability over a wide temperature range, improved bearing surfaces and fabrication characteristics, greater range of frictional and electrical characteristics, higher resistance to corrosion and chemicals, and reduced costs.

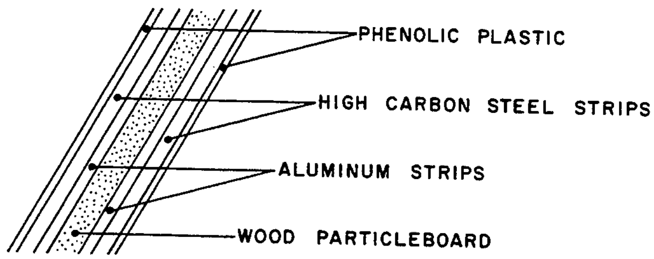


FIGURE 25.—Cross section of ski.

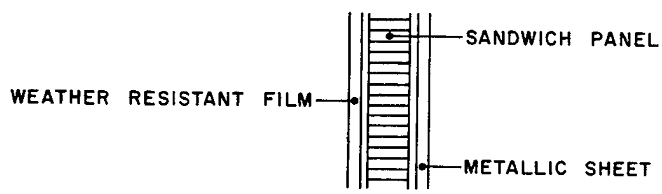


FIGURE 26.—Wall section.

Composite laminates have been shown to provide almost limitless design possibilities and versatility. An example of a future commercial application is a structural wall (fig. 26) that can be used widely in the construction industry. From left to right, the layers consist of (1) a film which serves both for weather protection and decoration, and (2) a sandwich panel, bonded to a metallic sheet, for load support and insulation; this, in turn, provides for radiant heating and cooling. Another layer of fluorescent material could be added for lighting (p. 118 of ref. 5).

FILAMENT-OVERWRAPPED PRESSURE VESSELS

Although glass-fiber composites are excellent for many structural applications, their use in pressure-vessel applications is limited. Johns and Kaufman (ref. 26) of NASA have described cylindrical cryogenic pressure vessels made by wrapping glass fibers around a metallic vessel in such a way that the metal acts as an impervious liner as well as supports a large part of the pressure load. In overcoming the yield strain difference between the glass fibers and metal, the glass fibers may be prestressed to put the metal into precompression.

The prestressing problem must be carefully considered. Although prestressing by pretensioning is generally desirable during winding, pressurization may be necessary, depending on the amount of prestressing required. To prevent damage during winding, a number of glass fibers, such as S–HTS glass, can be wound at about 25 percent of their ultimate load. If the vessel is to be used in either high-temperature or cryogenic environments, the difference in the thermal expansion coefficient of the filamentary and the metallic materials must be taken into account in prestressing. For example, an aluminum cylinder wrapped with S–HTS glass at room temperature with near-maximum prestrain will lose most of the prestress at cryogenic temperatures. Cases of this type require special winding techniques to obtain the necessary prestrains without filamentary damage.

In the course of the work described by Johns and Kaufman (ref. 26), aluminum cylinders were wound with sufficient glass filament to carry about half the hoop load at burst pressure, as based on uniaxial tensile properties. Because the metal and filaments reach their ultimate strengths simultaneously, this amount of fiber-glass-reinforced plastic is referred to as optimum. These cylinders were designed to have a one-to-one biaxial stress field at burst pressure, with the filaments being uniaxially wound.

A number of small overwrapped cylindrical pressure vessels were tested to burst. (See fig. 27 (*a*), (*b*), (*c*), and (*d*), from ref. 26.) The 2014–T6 aluminum tubing was wrapped with S–HTS glass impregnated with epoxy resin to form a layer of fiber-glass-reinforced plastic. Most of the vessels were pressurized to burst. In the optimum design, the metal is designed to be in a one-to-one stress field at burst pressure, where the failure orientation in the metal is not readily predictable. The fracture usually originates in the metal; the failure is either circumferential or longitudinal, or often both. When less than the optimum amount of glass has been used, the fractures seem to originate in the glass almost as often as in the metal.

When cylinders having optimum amounts of glass were tested at room temperature, as shown in figure 27(*a*), some of them failed without the glass breaking because of circumferential stress in the metal. In these cases, the resin had crazed during straining, allowing the pressure to escape when the aluminum failed and leaving the glass intact. When the tests were repeated with liquid nitrogen, the aluminum failed because of longitudinal stresses, as shown in figure 27(*b*). In some cases, the failure produced a sawtooth pattern; in others, a smooth pattern. Tests conducted on cylinders in liquid nitrogen with 90 percent of the optimum amount of glass indicated that the glass ruptured first, allowing the aluminum to bulge because of plastic flow. (See fig. 27(*c*).) Failures during tests conducted on both types of vessels in liquid hydrogen were catastrophic, as shown in figure 27(*d*). In similar experiments 2014–T6 aluminum cylinders wrapped with S–HTS glass proved to be as much as 50 percent more efficient than homogeneous 2014–T6 aluminum cylindrical pressure

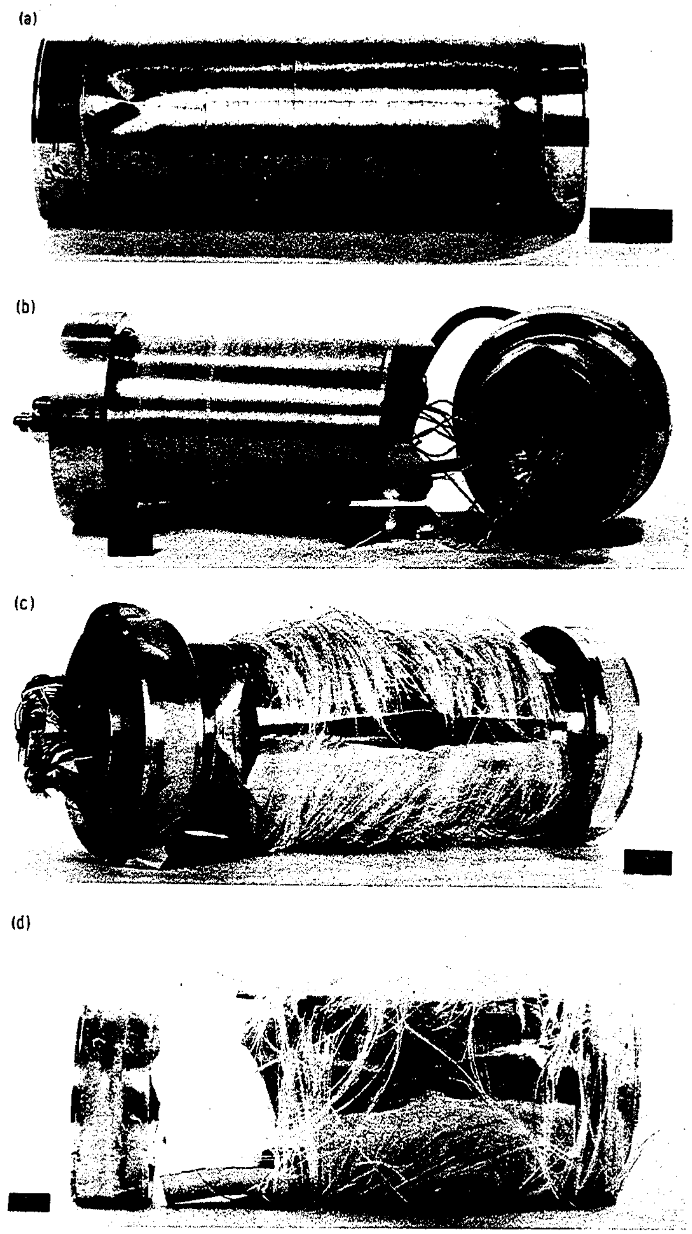


FIGURE 27.—Failures of 2014–T6 aluminum pressure vessels overwrapped with S–HTS glass: (*a*) 70° F, optimum overwrap; (*b*) −320° F, optimum overwrap; (*c*) −320° F, 90 percent of optimum overwrap; and (*d*) −423° F, optimum overwrap. (*Reprinted from Proceedings, AIAA/ASME 7th Structures and Materials Conference.*) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

vessels, and consequently, more efficient than spherical pressure vessels.

The greatest potential use for overwrapped tanks is as high-pressure containers since minimum thickness requirements based on fabrication and handling considerations usually predominate in low-pressure applications.

SEGMENTATION OF TANKS

Because enormous propellant tanks are needed for large launch vehicles, engineers have made radical changes in tank shapes and construction. The size of elliptical bulkheads such as those used on conventional tanks became critical because of the length of the launch vehicle. The usual bulkheads would not only add to the length but also create stability problems due to the increased skirt length, which is the peripheral section between tanks. When diameter increases, the thrust load and geometry require such increased skin gages and stiffener sizes for the skirts that machined integral panels are eliminated.

To solve this problem, NASA-Marshall investigated new concepts for large vehicle propellant tanks. Of these concepts, three are described below: (1) the multicell tank, (2) the semitoroidal tank, and (3) the flat-bulkhead tank.

Following a suggestion made by Professor Oberth some 40 years ago, NASA-Marshall conducted a detailed study of segmented designs which have been used quite successfully in large storage tanks for many years. One type of segmented tank is the integral cluster, scalloped, or multicell configuration.[1] The use of the multicell configuration instead of the conventional cylindrical pressure vessel is an innovation developed for launch systems. A 10-lobe version of the multicell design (ref. 27), shown in figure 28, is composed of thin-walled, partial-circular, cylindrical shells and radial webs. The partial cylinders that form the tank periphery and the radial webs may be of unstiffened, stiffened, or sandwich construction. The radial webs extend from a center tube to the juncture of two outer wall sections and then longitudinally between cell and closure bulkheads. Bulkheads are partial cones connected to the partial cylinders by spherical sectors. Extended and partial Y-sections are used as attachments for cylinder-web-bulkhead junctures and cylinder-spherical skirt junctures along the periphery of the cross section, respectively.

One advantage of the multicell configuration over the conventional

[1] The multicell configuration is no longer an isolated concept, but is now considered to be a tank with low-profile bulkheads.

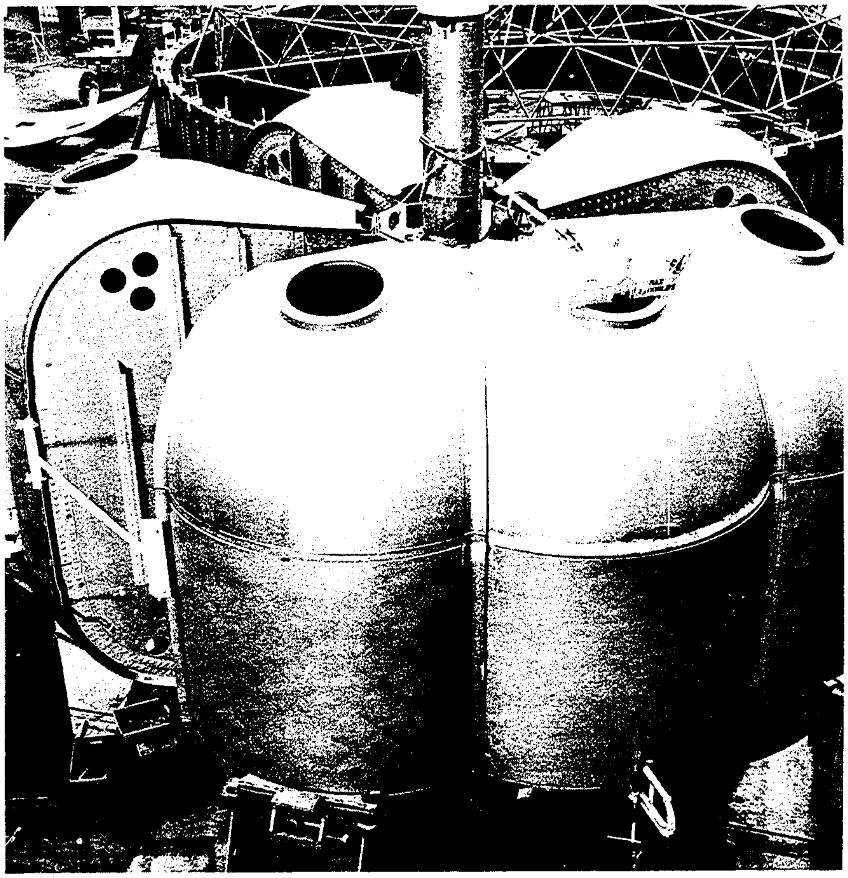


FIGURE 28.—Ten-lobe multicell tank.

pressure vessel design is the reduction in bulkhead depth. As figure 29(*a*) shows, the bulkhead of the multicell structure is relatively flat. The multicell design not only permits a reduction in overall missile length by decreasing the length of the tanks but greatly shortens the space between the tanks themselves. The radial webs, used most efficiently as part of the basic structure, eliminate the need for baffles to reduce sloshing. The multicell construction provides a flexibility that no other configuration can offer; namely, it distributes basically needed material for a given pressure vessel into both the outer shell and the internal tension wall system. Furthermore, it offers flexibility in selecting tank diameters and bulkhead arrangements and makes it possible to use existing facilities for manufacturing sections of a multicell vehicle. Blumrich (refs. 27, 28, and 29) and Wuensher and Berge (ref. 30) of NASA, among others, have been associated with this launch vehicle design, and their reports include excellent discus-

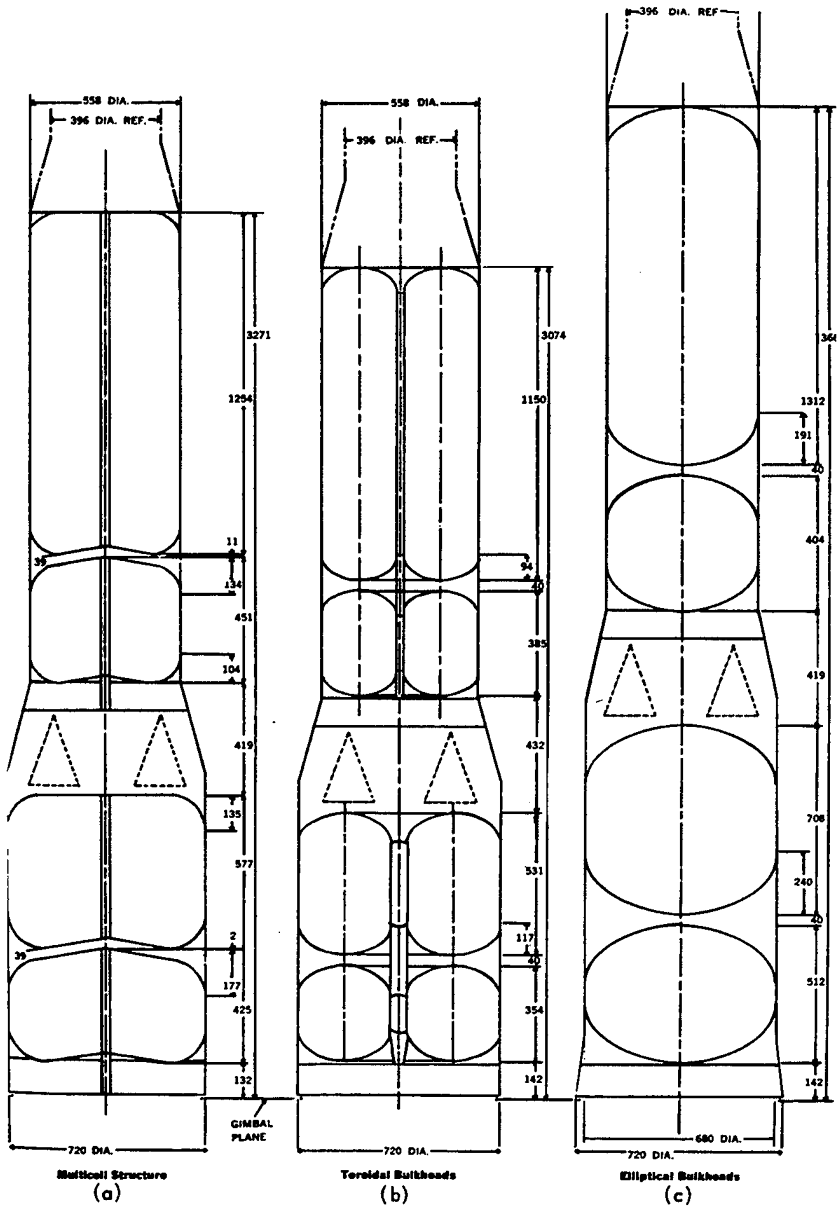


FIGURE 29.—Large-size first- and second-stage structural systems (dimensions in inches). (*Reprinted from* Astronautics and Aeronautics.) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

273-140 O - 67 - 4

sions of the design and development of manufacturing techniques. The analysis of multicell structures also has been treated recently by Blum (ref. 31) and Wilson et al. (ref. 32).

Another principle under investigation is the semitoroidal tank, shown in figure 29(*b*), in which two ellipses smaller than those in 29(*a*) form a bulkhead. In figure 29 (from ref. 29), three design concepts are compared: the multicell, semitoroidal, and elliptical bulkhead. Features of the semitoroidal tank are: (1) supports between the tanks and between rear tank and thrust structure; (2) the centerpost which has a diameter determined by the acceptable thickness of the adjacent bulkhead portion; and (3) a connection from the bulkhead to the centerpost. The tank is supported on the thrust structure by the tail section which has either radial beams or at least one member extending through the centerpost of the vehicle to pick up the load. If the material is too thick, it is not possible to make a tangential connection from the elliptical bulkhead to the centerpost. For structural and manufacturing reasons, a conical transition between bulkhead and centerpost seems to be preferable. The advantages of the semitoroidal design include: (1) reduction of stage and vehicle lengths, and (2) elimination of deep elliptical bulkheads because the new design permits the use of separate tanks, with some additional reduction of stage length.

A third principle under investigation is the flat-bulkhead concept (shown in fig. 30 from ref. 27), so named because of the overall appearance of the design. The concept is that of a segmented tank employing several of the principles already discussed under multicell tanks. Further tests are being conducted on a model of the flat-bulkhead concept to determine its structural integrity. Resulting data may be used to compare tank designs and determine preferability.

ISOTENSOID STRUCTURES

Design problems involving filamentary-matrix construction are simplified if the direction of loading is confined to the principal directions of stress and shear stresses in the matrix are avoided. When shear stresses can be prevented or offset, conditions such as those found in so-called isotensoid structures (ref. 5, p. 126) are produced.

In isotensoid structures, the filaments (in filament-wound structures) are oriented so that they are equally stressed and provide resistance in the principal stress directions in proportion to the magnitude of principal stresses. Because this technique allows circumferential stresses to be twice as great as axial stresses, it is excellent for cylindrical pressure vessels in which about half the fibers

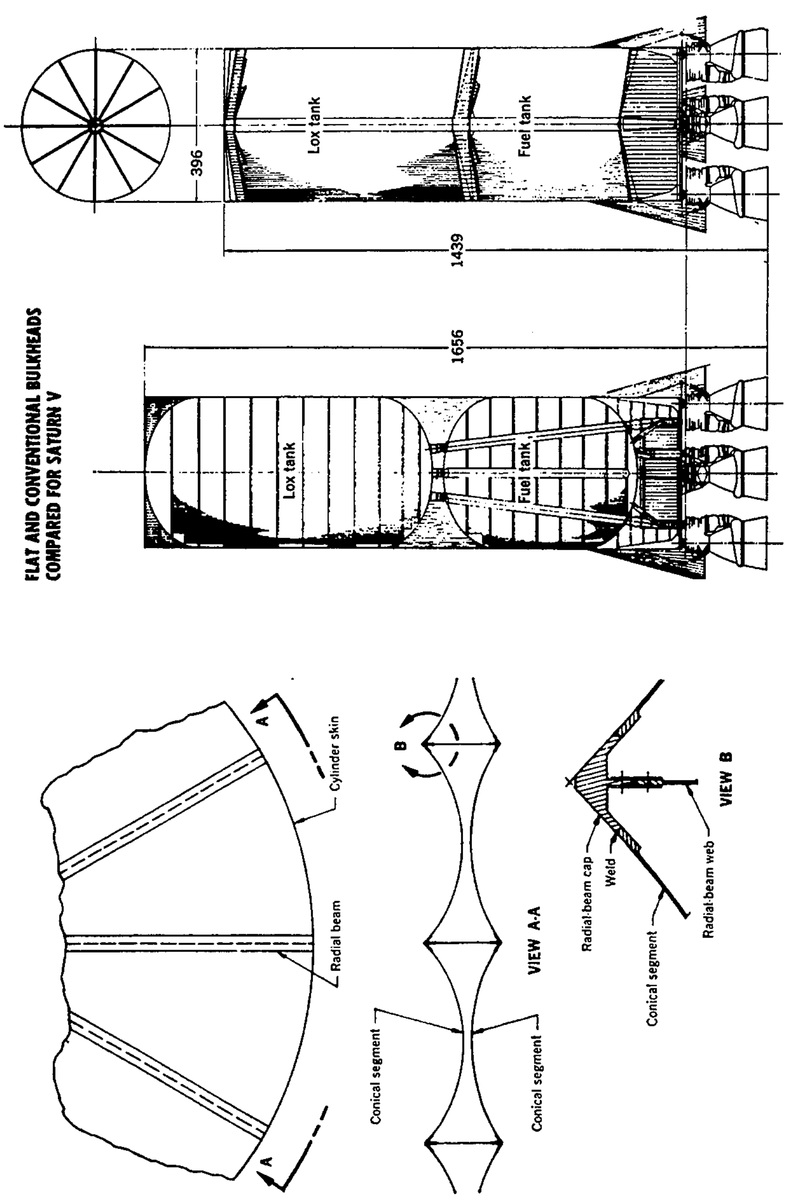


FIGURE 30.—Flat-bulkhead schematic. (*Reprinted from* Astronautics and Aeronautics.) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

are wound axially and half circumferentially. For the preferred helical winding pattern, the filament is wound using an angle to produce its tensile resistance so as to give a desired 2-to-1 ratio when resolved in the two directions of principal stress. Under design conditions, however, such a layer of filament will produce considerable shear stresses in the matrix. Therefore, a second layer is wound in the opposite direction (a reverse wind) to offset the shear stresses.

Comparisons of practical design parameters of the cylindrical and spherical shells have shown that a sphere is a more efficient strength-to-weight ratio pressure vessel. Glass-fiber-reinforced plastics have been found to be the best basic constituents.

The isotensoid design is based on the concept of designing an equal and uniform tension in each fiber. Levenetz (ref. 33) showed that certain modifications of the spherical shape can improve the efficiency of a vessel. He designed the winding pattern of the fibers to maintain unidirectional loading and uniform tension. The geometry of this modified sphere is called oblate spheroid, ovaloid, or ellipsoid, as shown in figure 31. This efficient type of pressure vessel is characterized by a short polar axis and a larger perpendicular equatorial diameter. The head shape is determined by an elliptic integral. The only parameter is the ratio of the central opening to the vessel diameter, which determines the variations of the winding angles, with fibers oriented toward the polar axis. The angle of the fibers with the polar axis depends on the polar openings (end closures).

Composite stresses of 200 000 psi have been reported in rocket cases of this configuration. Recent research at NASA Langley Re-

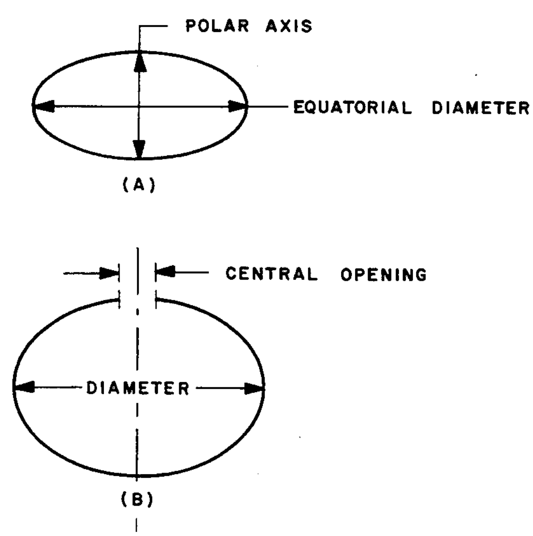


FIGURE 31.—Isotensoid configuration.

search Center (ref. 34) and in industry on the isotensoid concept of filament winding has resulted in extremely high strength-to-weight ratio rocket-motor cases. The determination of the coordinates for isotensoid pressure vessels is given by Zeckel (ref. 35) and in other references.

TENSION-SHELL CONFIGURATION

The tension-shell configuration, sometimes called the "Langley tension shell," is a reentry vehicle design developed for the Voyager mission to meet minimum weight and high-drag profile requirements. Since high-drag profiles generally have a wide base as compared to axial length, buckling is the usual mode of failure of such a configuration.

In figure 32 (*a*) and (*b*), a conventional design is compared with tension-shell design. In the tension-shell configuration in figure 32 (*b*), the meridional contour flares outward toward the base. The payload is so attached to the shell that the inertial forces, resulting from the deceleration of the payload, develop an axial tension in the flared portion of the shell. Although the external aerodynamic pressure tends to produce circumferential compression in the shell, this compression is more than compensated for by the circumferential tension produced from a combination of the axial tension and the flared portion. In turn, this tension-shell configuration reduces the buckling tendency of the shell. Nevertheless, the crushing action of aerodynamic forces must be resisted even in the tension shell with a compression-resistant spherical nose segment at the forward end and a compression-resistant ring at the aft section. Papers by Anderson et al. (ref. 36), Halberg (ref. 37), and Levy and Hess (ref. 38) give detailed discussions and analyses of the tension shell.

TENSION-STRING STRUCTURE

Alai (ref. 39) has reported a new ultralight, high-drag concept called the "tension-string structure," which is a variation of the tension shell. This configuration, lying between that of the sphere cone and a tension shell, is produced by using high-strength filament materials in tension, as shown in figure 33 (from ref. 39). The basic elements are a strong forebody and shield, nose cap, equatorial ring, afterbody, central support, and payload. Figure 34 (from ref. 39) shows a completed model.

A forebody is generated by straight strings in tension, arranged to form a curved (ruled) hyperboloid surface. A specified amount of pretension is applied to each string during winding to assure tension under all loading conditions specified by the mission. The strings

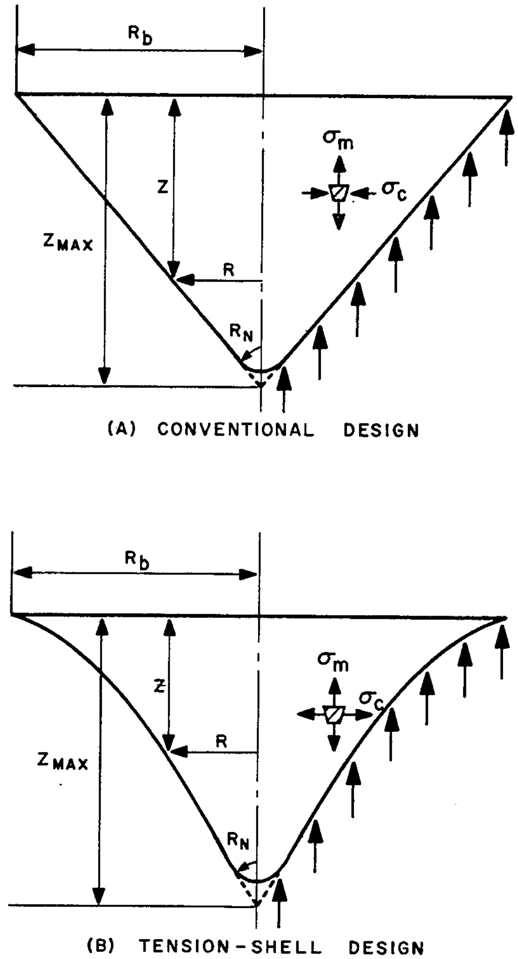


FIGURE 32.—Tension-shell structure.

generate a surface of revolution of negative Gaussian curvature. The finished shape is obtained by applying an elastomeric-shield material over the filaments. The nose cap may be either blunt or pointed, wound along with the forebody, or solidly integrated with the payload. The equatorial ring, one of the primary compression elements, is restrained (bonded) by the forebody and afterbody strings, stabilizing the ring laterally while providing resistance against lower-mode overall buckling.

The afterbody, wound as a continuation of the forebody, is shaped like a truncated hyperboloid. The exact configuration, however, is

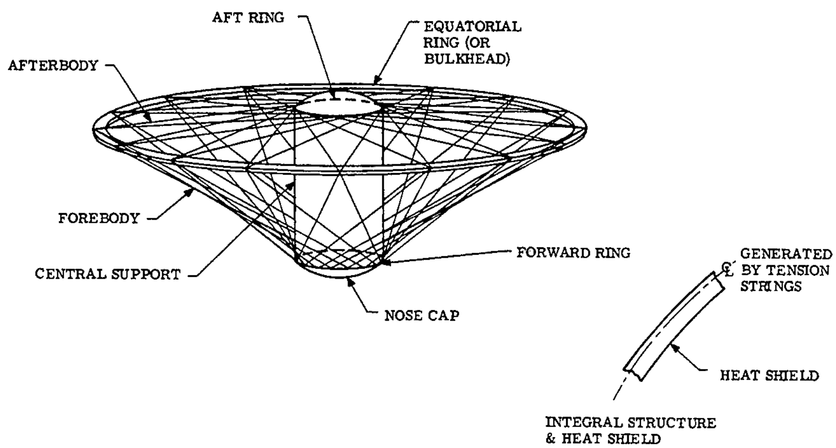


FIGURE 33.—Tension-string structure. (*Reprinted from Proceedings, AIAA/ASME 7th Structures and Materials Conference.*) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

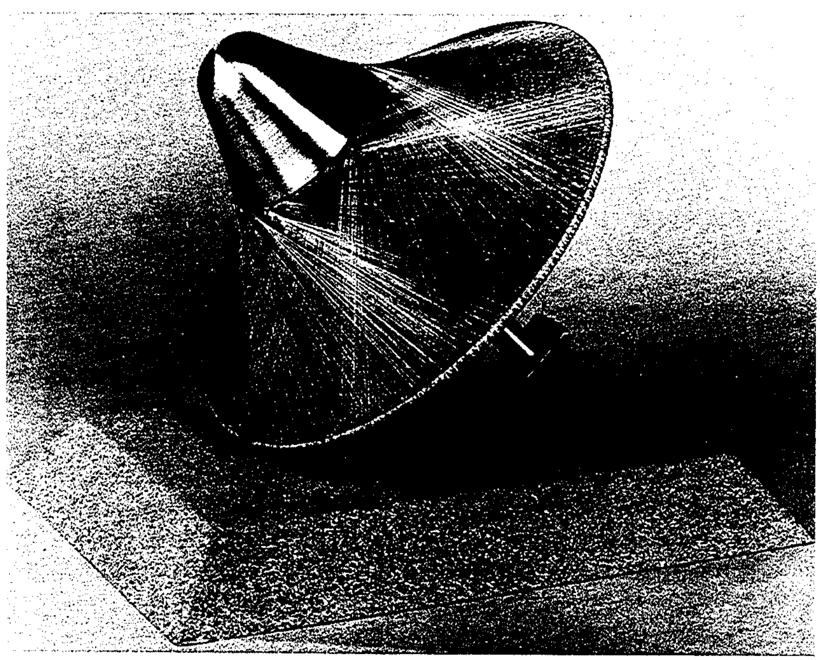


FIGURE 34.—Tension-string model. (*Reprinted from Proceedings, AIAA/ASME 7th Structures and Materials Conference.*) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

arbitrary and depends ultimately on vehicle requirements rather than on aerodynamic considerations.

The central support, the other primary compression element, may be any shape required. In figure 34 it is shown as a cylinder supporting the payload and fore-and-aft rings, to which forebody and afterbody strings are bonded.

EXPANDABLE STRUCTURES

The size and mass of missile payloads will always be restricted. At present, there are two approaches to transporting large structures into space in small, lightweight packages. One approach is space construction, requiring prefabricated sections to be launched, rendezvoused, and assembled in space. The second uses a structure that can be expanded from a small to a large volume. Figure 35 (from ref. 40) is a step-by-step illustration of the ejection, erection, and rigidification of an antenna dish. There are four basic types of expandable structures: (1) inflatable, (2) chemically rigidified, (3) unfurlable, and (4) elastic

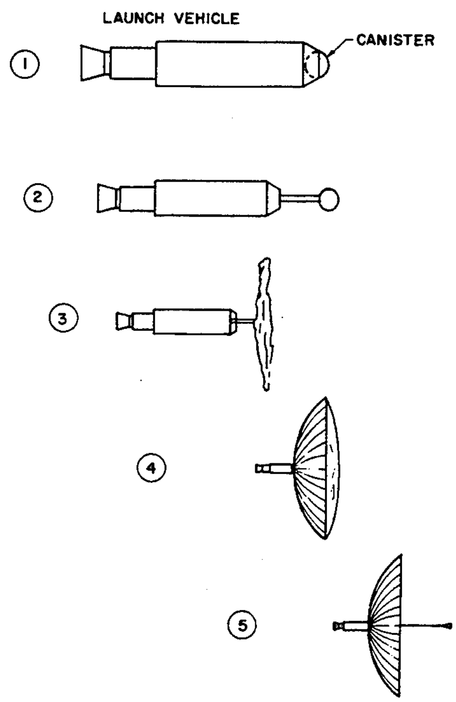


FIGURE 35.—Ejection, erection, and rigidification of antenna dish. (*Reprinted from Proceedings, AIAA/ASME 7th Structures and Materials Conference.*) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

recovery structures. Seven basic techniques (see fig. 36 from ref. 41) are available to the structural engineer to bring about the expansion and rigidification of flexible materials. Brink et al. (ref. 42) and Schuerch and Schindler (ref. 43) have analyzed the foldability of expandable structures.

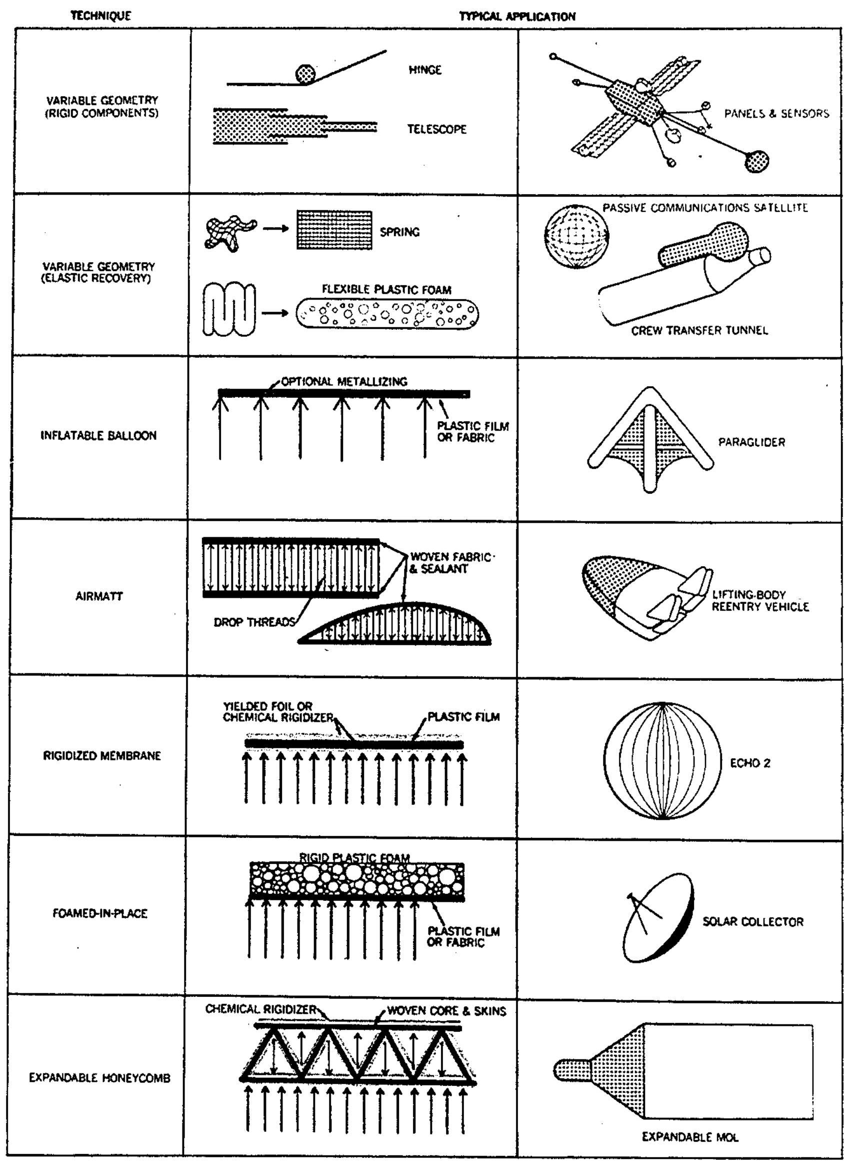


FIGURE 36.—Expandable structure deployment technique and applications. (*Courtesy of* Space/Aeronautics.)

Inflatable Structures

"Inflatable structures" are defined as fabric or film envelopes that maintain structural integrity by internal pressurization. Generally they are spherical, cylindrical with dome ends, or toroidal shells. An example of this concept is the inflato plane (designed by Goodyear Aircraft Co. for the U.S. Army), an aircraft with a 28-ft wingspan, that weighs only 290 lb. A variation of the principle is found in the Airmat (also developed by Goodyear), in which varying lengths of integrally woven drop threads create cross-sectional shapes, such as flat panels or airfoils. (See fig. 37 (*a*), (*b*), and (*c*) from ref. 41.) Potential space applications include extensions for space stations, space furniture, and rendezvous docks.

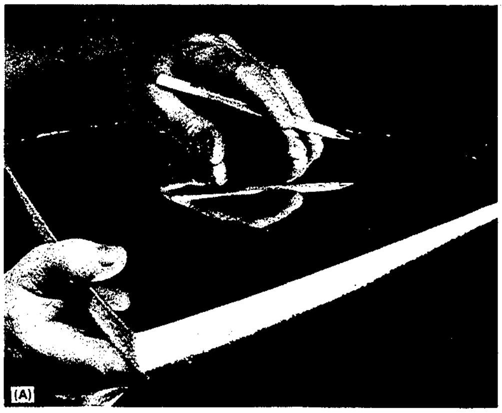
Chemically Rigidified Structures

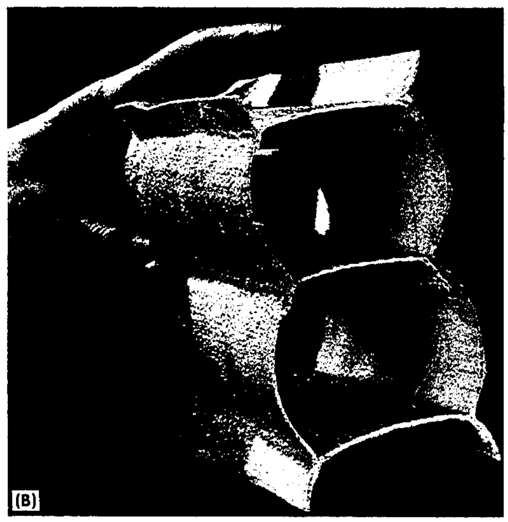
Deployed structures given rigidity through a chemical reaction are made by impregnating a film fabric with a rigidifying resin. Woven, fluted, or corrugated sandwiches also may have application in these structures to provide buckling resistance under bending loads. Three basic types of chemical rigidification systems are: (1) plasticizer boiloff, (2) gas catalyst, and (3) radiation-cured systems.

The best plasticizer boiloff system utilizes gelatin and water as a plasticizer. Gelatin rigidification systems were developed by the U.S. Air Force, Swift & Co., Monsanto Chemical Co., and Hughes Aircraft Co. for the Air Force. Such a system permits a structure to be deployed, rigidified, and tested on Earth, then plasticized and packaged for launch. Gelatin can be replasticized repeatedly by exposure to humidity. In the space environment, vacuum causes water migration out of the structure and produces rigidity.

Gas-catalyzed urethanes, developed by Archer Daniels Midland Corp. for the U.S. Air Force, are cured into rigid plastic upon exposure to water vapor. Another example, developed by National Cash Register Co. for the Air Force, is a fast-reacting system that utilizes a vinyl monomer and an amine catalyst. The catalyst, locally introduced by a fine spray or gas, will automatically cure and propagate throughout the entire structure. Both of these gas-catalyst systems require refrigeration to extend storage life.

Radiation-cured systems employing epoxy and polyester resins have been developed by Hughes Aircraft Co. for the Air Force and NASA. These systems require exposure to solar radiation for initiation and/or continuation of the rigidification process. A foam-in-place rigidification system was once considered for space use, but tests of several versions for the Air Force and NASA showed it to be impractical because of its complexities and poor strength-to-weight ratios.





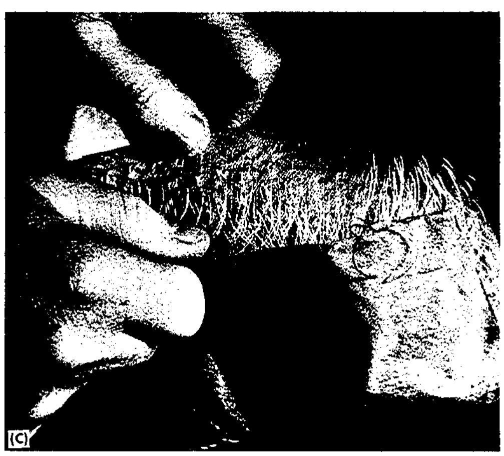
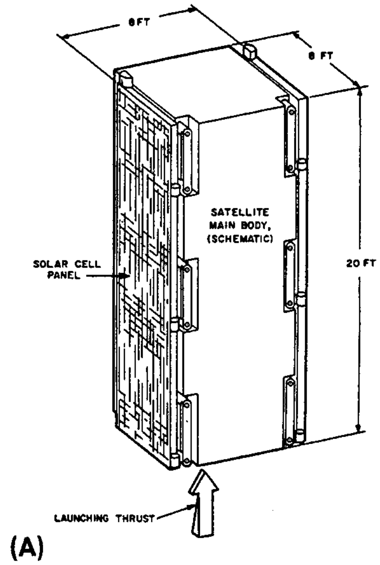


FIGURE 37.—Three expandable structure sections: (*A*) segment of rigidified-foam solar reflector which would be activated in space; (*B*) rigidified honeycomb section; and (*C*) sample of airmat. (*Courtesy of* Space/Aeronautics.)



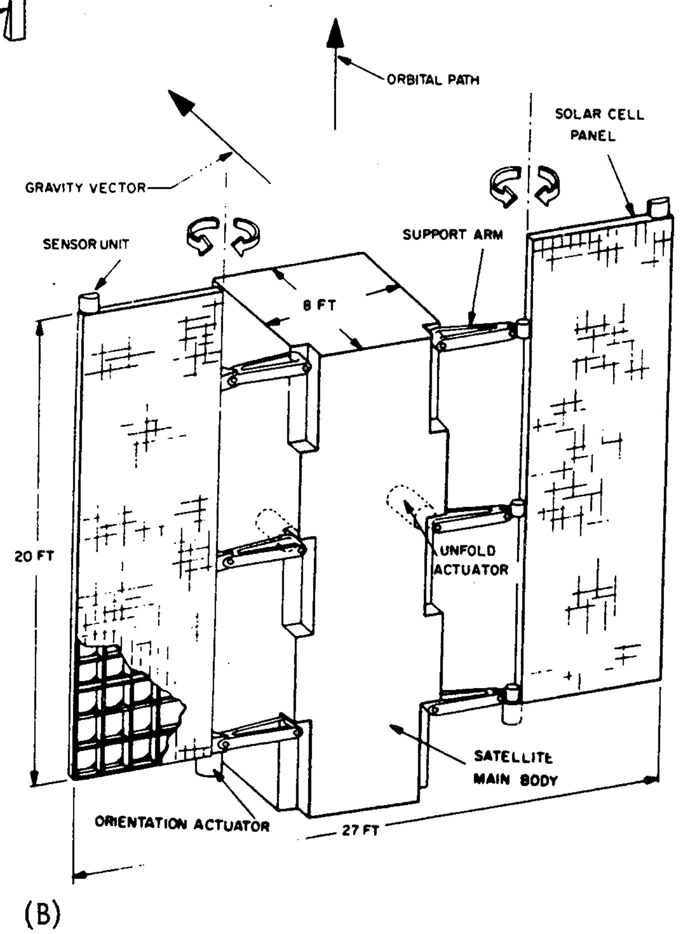


FIGURE 38.—Solar cell array: (*A*) stowed position for launching, and (*B*) operating position. (*Courtesy of Radio Corporation of America.*)

Unfurlable Structures

Unfurlable structures are defined as bodies that are mechanically deployed by hinges, sliding sections, and telescoping members. Conventional materials often can be designed to fold mechanically into a component package. This concept and its variations have been widely used by aerospace designers because conventional mechanical design procedures can be applied with high reliability. Figures 38 (*a*) and (*b*) and 39 (from ref. 44) are designs used for solar cells.

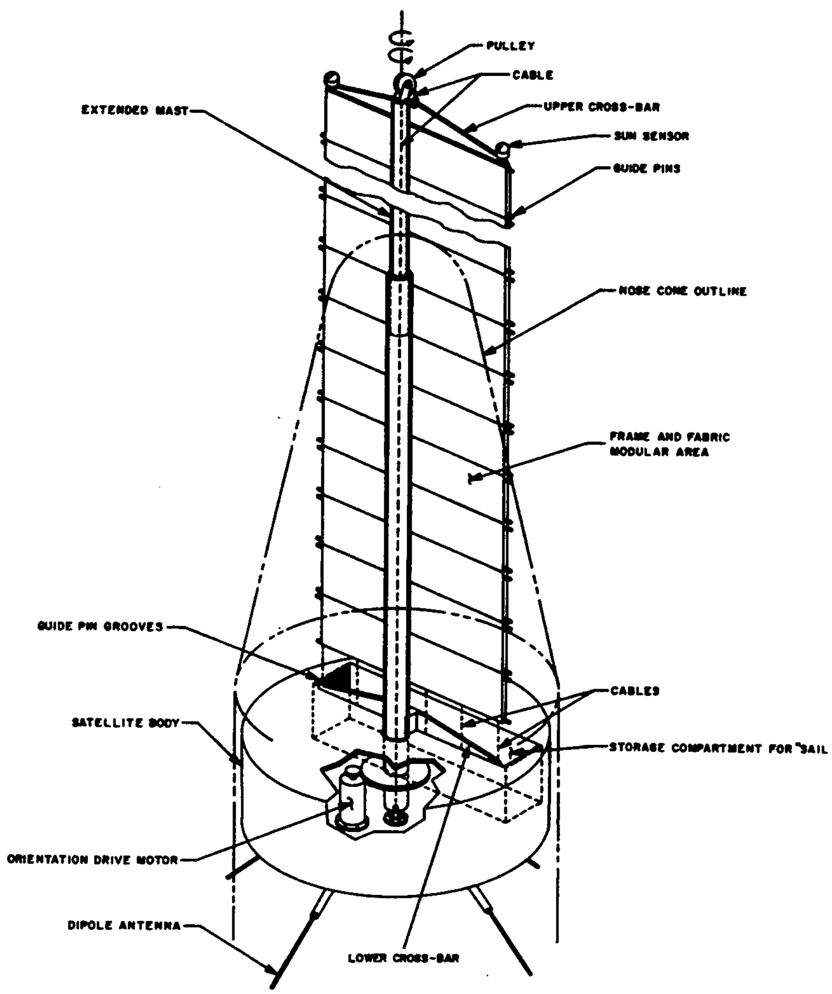


FIGURE 39.—Unfurlable sail type of solar collector. (*Courtesy of Radio Corporation of America.*)

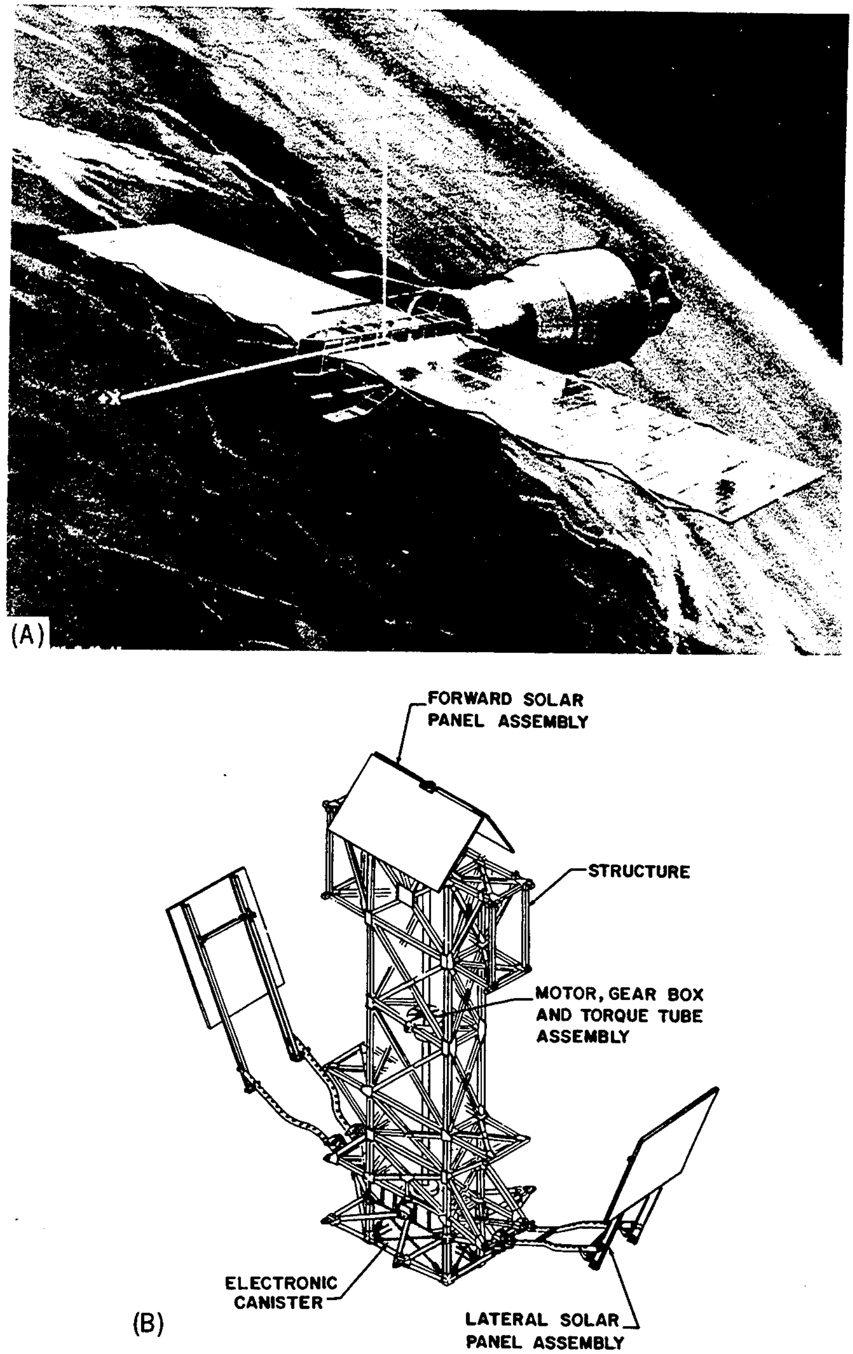


FIGURE 40.—Pegasus satellite and its center section: (*A*) showing orientation of axes, and (*B*) center section.

The Pegasus Satellite (Fig. 40 (*a*) and (*b*) from ref. 45), formerly called the Micrometeoroid Measurement Capsule, was an excellent example of this type of structure. A modular winged satellite launched for meteoroid detection in near-Earth orbits, it was developed by Fairchild Hiller Corp. for NASA under the supervision of Marshall Space Flight Center. Deployment of the detection panels was accomplished by a mechanical unfolding stress structure, giving a wingspan of 96 ft.

Elastic Recovery Structures

Both the Air Force and NASA have launched programs to develop expandable structures for aerospace applications, such as solar collectors (Geophysics Corp. of America, Viron Division; National Cash Register Co.; Goodyear Aerospace Corp.); reentry applications (Goodyear Aerospace Corp.); space stations (Goodyear Aerospace Corp. for the Air Force); expandable airlocks, crew tunnels, and hangars (Goodyear Aerospace Corp., Geophysics Corp. of America, Viron Division, for the Air Force and Narmco Division of Whittaker Corp. for NASA); and lunar shelters (Goodyear Aerospace Corp. for NASA). A recent report by Brink et al. (ref. 42) contains a detailed discussion of the development, feasibility, and applicability of the elastic recovery concept to expandable structures.

These structures are defined as those that utilize the basic elastic properties of the materials to deploy and provide limited structural rigidity. Such a structure is packaged by compressing and folding it into an extremely small container. Upon release from the container, the stored potential energy of the material is sufficient to expand and rigidify the structure.

The STEM boom of De Havilland Aircraft Co. is an excellent example of this concept. Circular in cross section, the boom is slit along its entire length so that it can be rolled up on a drum that flattens the cross section. Upon extension of the boom, the flat ribbon springs back into its original circular cross section.

Figure 41 (from ref. 41) shows properties and applications of the expandable-structure techniques, and figure 42 (from ref. 41) compares structural-merit to packaged-to-deployed volume ratio.

BUILDING CONSTRUCTION APPLICATIONS

Pneumatic Structures

Experience with novel structures in space may facilitate the use of such new ideas on Earth. Within the atmosphere, an inflatable structure is commonly referred to as a pneumatic structure; this has become an important concept for commercial structures. Membranes pre-

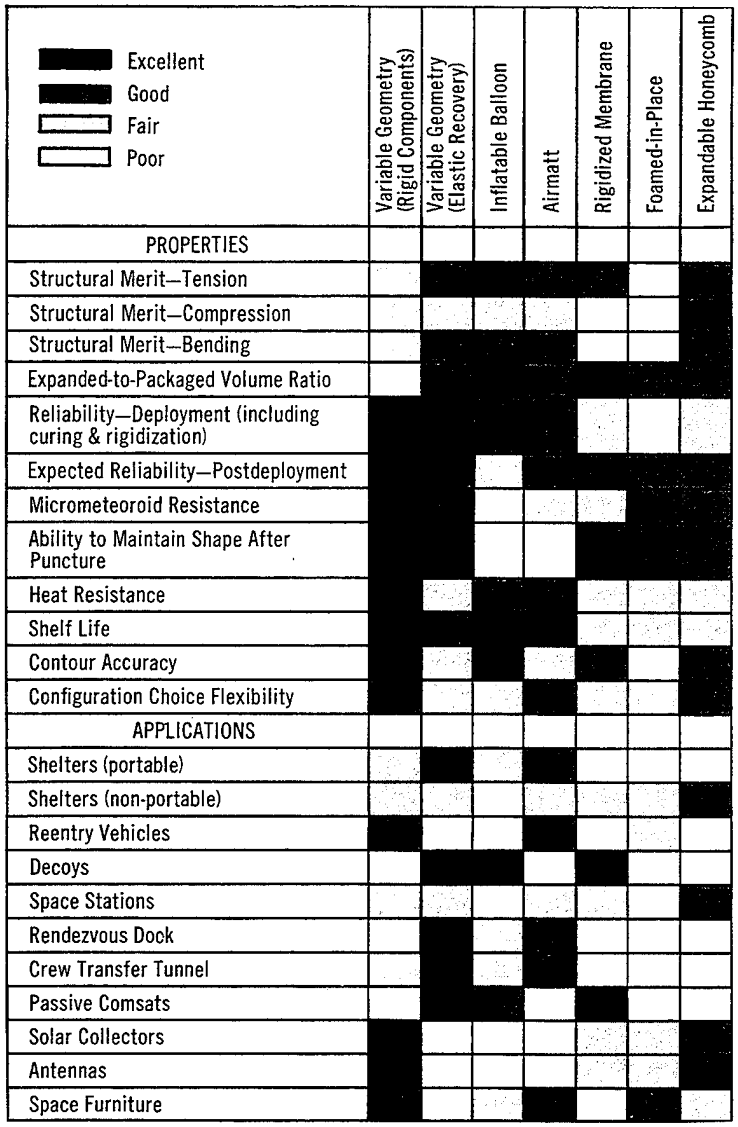


FIGURE 41.—Properties and uses of expandable structures. (*Courtesy of* Space/Aeronautics.)

stressed by internal pressure completely enclose a volume or a number of separate volumes. The membranes are very thin stressed skins, generally built from sheet metal, fabrics, or fiber-reinforced plastics. They are so thin that, for all practical purposes, they cannot resist compression, bending, or shear, but only tension. Figure 43 (from

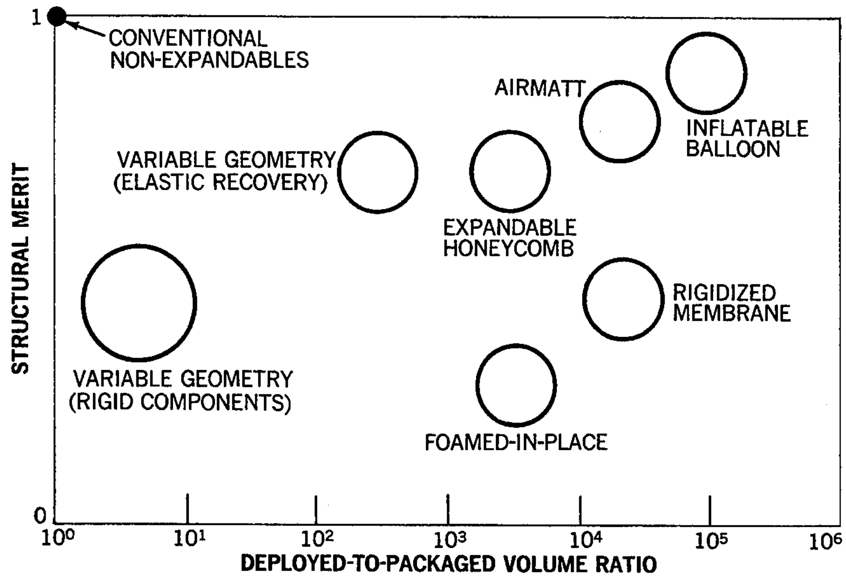


FIGURE 42.—Comparison of structural merit to deployed-to-packaged volume ratio of expandable structure techniques. (*Courtesy of* Space/Aeronautics.)

ref. 46) illustrates an advanced application of membrane structures called a "wavehall."

Structural membranes may be classified as either anticlastic or synclastic, depending on the curvature of the surface. "Anticlastic" means having opposite curvatures; i.e., having the center of principal radii at different sides of the observed tangent plane. A soap bubble and certain pneumatic structures are synclastic surfaces. The rubber raft is an excellent commercial application of this principle.

Concepts of an optimal structural form for two-dimensional components have long been in common use for the arch and suspension-cable structures. The extension of these concepts to curved surfaces or three-dimensional structures can be characterized by the structural membrane or minimum structure. Otto and Trostel have developed "sail-shells," stiffened pneumatic surfaces, and other tensile-stressed forms (ref. 46), which are based on the premise that structural form is determined by the equilibrium of forces rather than the geometry. In certain applications, a reversal of the tensile-stressed structure seems to result in a parallel reversal of stresses from tension to compression.

Pneumatic-structural systems, proposed by Lanchester in 1917, have made practical great large-span domes and hydraulic structures such as large-span dams. Figure 44(*a*) (from ref. 47) illustrates a pneu-

273-140 O - 67 - 5

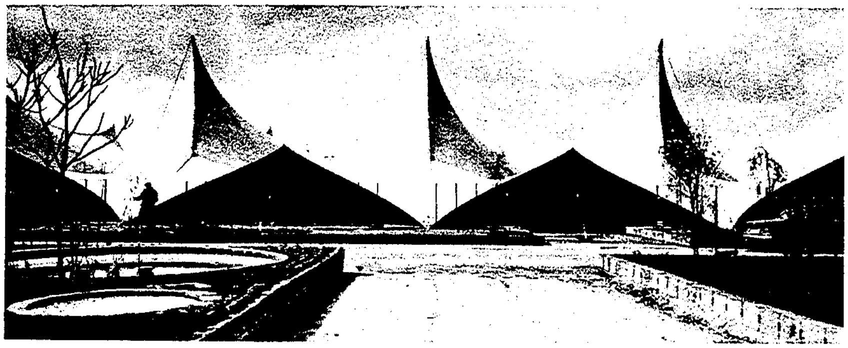


FIGURE 43.—A wavehall membrane structure.

matic-roof application with thin plastic membranes inflated by a small pressure to create stable domes for swimming pools or other installations. An overpressure of only one-tenth or two-tenths of a psi is sufficient to hold up such a structure. Regular doors can be used, since, even if they are opened frequently, the loss of pressure in the large enclosed volume is negligible. Such losses, of course, are replaced intermittently under control of a pressure gage.

Koch and Weidlinger designed a balloon roof (fig. 44(*b*)) for a summer theater in the form of a lens inflated by a pump to an overpressure of 10 psf. And some 24 years ago, Wallace Neff began to produce stiff shells called "igloo houses" by pouring concrete over rubber balloons (fig. 44(*c*)). An inflated balloon supports a reinforcing steel mesh which is sprayed with a 1-in. layer of concrete from a concrete gun. After the concrete has hardened, the form is deflated and pulled out of the house through the door opening. The igloo house was invented by Neff and designed by Elliott Noyes and Mario Salvadori.

In 1962, in Essen, Germany, a pneumatically stressed balloon skin was sprayed with plastic from the inside to produce a weatherproof, insulated shell. The finished wall was a 20-mm-thick sandwich construction produced of glass fiber, polyester, and Perlite. After the polyester had set, the inside pressure (55 mm H_2O) was lowered and openings were cut into the shell. In this construction a translucent skylight can be made. Another example in figure 45 (from ref. 48) illustrates a step-by-step version of this same technique. In this case a plastic film was inflated and held taut by compressed air while a 2-in. coat of urethane foam was sprayed onto the inside of the film. After drying, the shell was covered with metal reinforcing bars and openings were cut into the shell. The outer surface was then sprayed with 3 in. of concrete to make a durable, well-insulated, contemporary design.

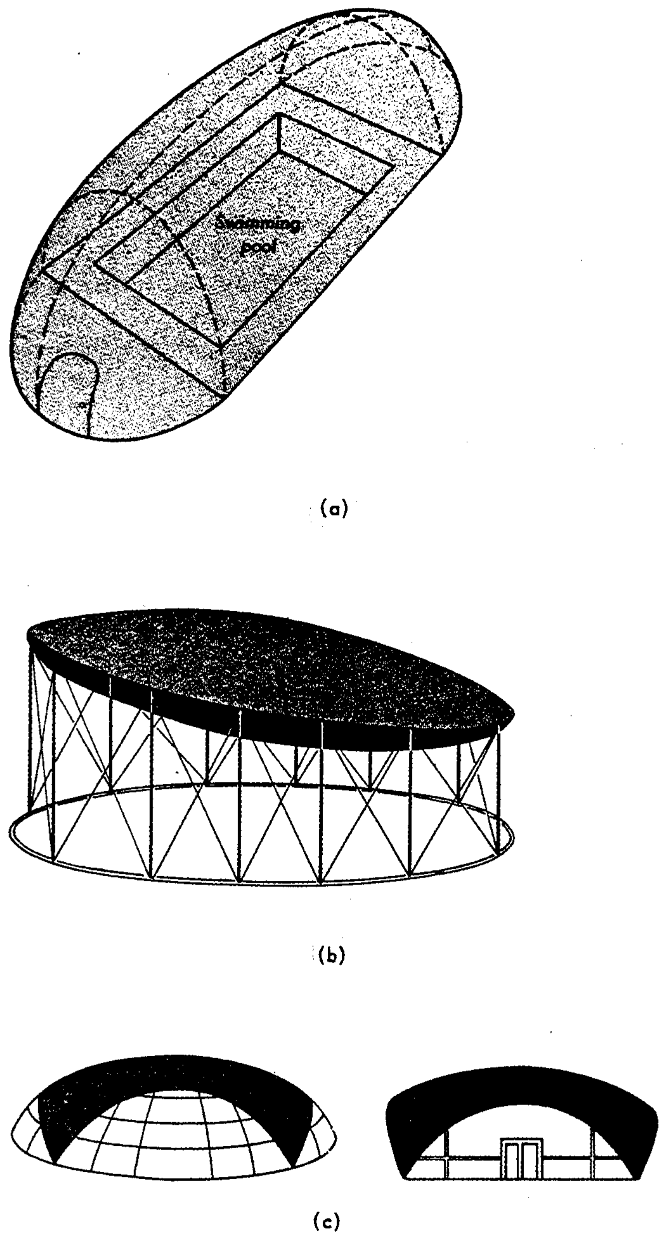


FIGURE 44.—Modern enclosures: (*a*) pneumatic roof, (*b*) balloon roof, and (*c*) igloo (balloon) house.

Lattice Shells and Domes

Lattice shells can have very high efficiency, use little material, and are stiff against buckling. Although their forms look complicated, they are very similar to their natural counterparts. (See fig. 46 from ref. 46.)

In figure 47 (from ref. 46), a method of forming a continuous lattice is shown. On a formwork (*A*), a steel wire fabric (*B*) and a second fabric (*C*) are placed in layers connected with wire ties (*D*). Between the ties are placed rubber balloons (*E*), which are blown up to prestress the wire fabric. After the balloons are deflated, only small voids remain; these are filled and stiffened with cement, concrete, or resins to form a continuous lattice shell. Finally, the rubber balloons are removed.

Shell systems may also be constructed from thin laths; in these systems a flat square grid is deformed into a spatially curved surface. Some examples are shown in figures 48 and 49 (from ref. 46). Such a shell may be erected over any arbitrarily chosen planform, such as one- or two-sided curvatures in dome or saddle form. Elastic, thin, flexible profiles of wood, aluminum, or steel are best suited as building materials for this adaptable method. Lattice shells can be easily erected and dismantled, and their form can be changed without destroying the structure.

Space-frame domes permit the spanning of large distances with relatively less material than required by other methods. In the geodesic dome designed by Fuller (ref. 49), the triangle and the pentagon are used in subdivision of bars of equal length (fig. 50). It is called "geodesic" because the vertexes of the curved figures that form its structure mark the arcs of great circles, known in geometry as geodesics. Radar domes built this way have withstood winds up to 150 mph and temperatures ranging from far below zero to over 150° in direct sun. The U.S. Pavilion at Canada's Expo '67 is an example of the geodesic dome.

The technology developed for space-frame domes with fiber-reinforced plastic skins can be interpolated into a folded-plate design for use in flexible and demountable dome structures. Although used mainly as roof structures, these plates may also be used as vertical walls to resist both vertical and horizontal loads. Combinations of folded-plate roofs and walls have been used to enclose large spaces. These plates may be ribbed, curved, or sandwiched for strength and rigidity (fig. 51).

Figure 52 is a proposed design employing the shatterproof sheet-panel concept. These panels are now available in a wide variety of sizes, colors, and light-transmission properties.

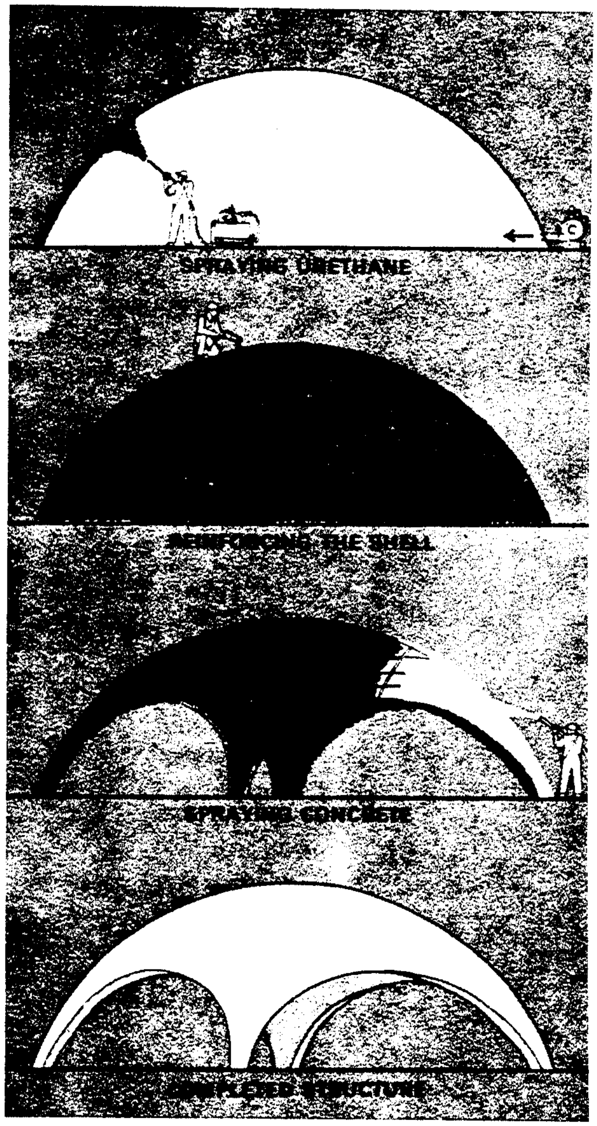


FIGURE 45.—Plastic building shell.

Figure 53 illustrates the formed-skin concept. A pavilion, constructed in 1950 for the U.S. Exhibition in Moscow, had a roof composed of a 20-ft-high, umbrellalike cluster of 16-ft-diameter hexagonal canopies supported on hollow columns. For translucency, as well as strength in shape, glass-fiber, mat-reinforced plastic was used as forming skin. Thickness ranged from $\frac{1}{16}$ to $\frac{1}{4}$ in., depending on the

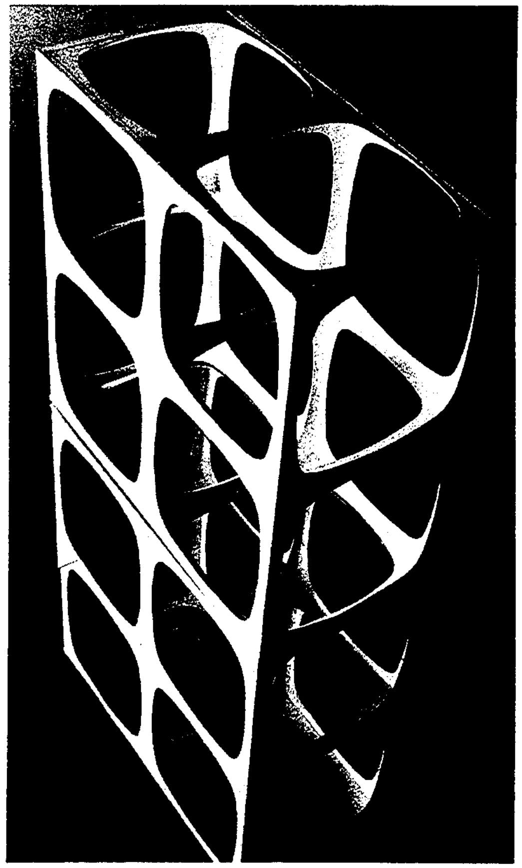


FIGURE 46.—Two-layered lattice shell made of sheet elements.

number of layers of material. The formed-skin concept, utilizing compound curvatures and shapes, is almost unlimited as a design approach.

Large spherical domes over radar installations (radomes), built on the principle of thin plastic membranes, do not interfere with the reception and transmission of electromagnetic beams. One recent type of dome structure for a ground-based radome uses foamed plastics which have very low electrical-loss characteristics. The edges of foam panels can be joined together with foam plastic or adhesive bonding to provide a uniform wall radome with a minimal effect upon transmission losses. This design may have wide uses, particularly for higher frequency applications.

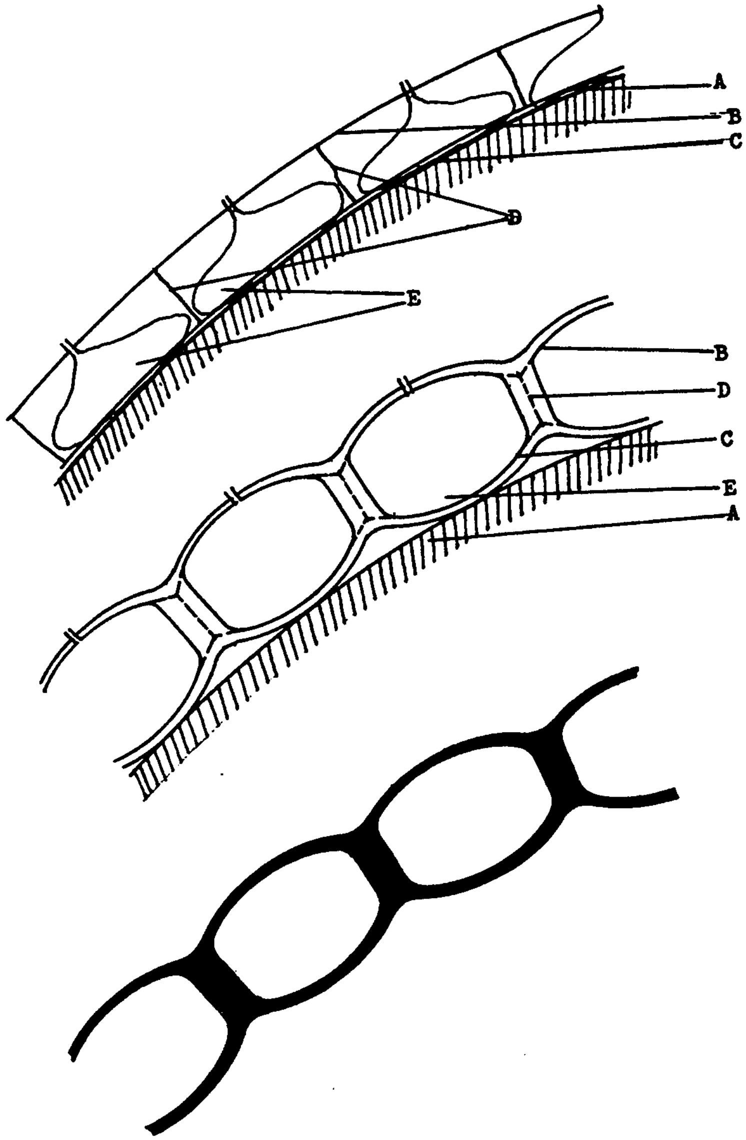


FIGURE 47.—Method of constructing a space structure.

The "House of the Future" at Disneyland uses self-supporting shells of fiber-glass-reinforced plastic in a unitized design that permits both brightness and stiffness. The 8- by 16-ft cantilevered, hollow-structure monocoque shells comprising the structure are glass-fiber

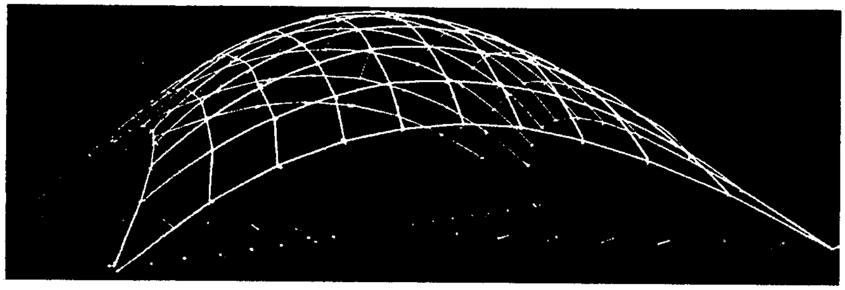


Figure 48.—Chain model.

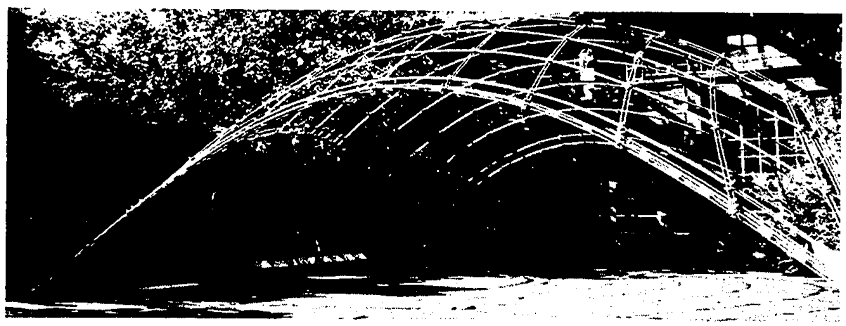


Figure 49.—Finished dome.

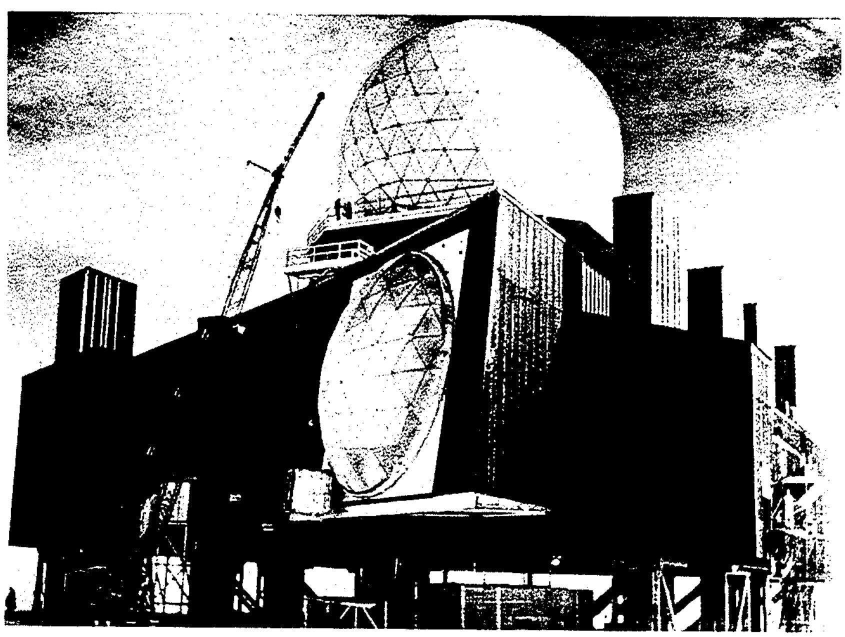


Figure 50.—Space-frame dome structures. (*Courtesy of Owens-Corning Fiberglas Corp.*)

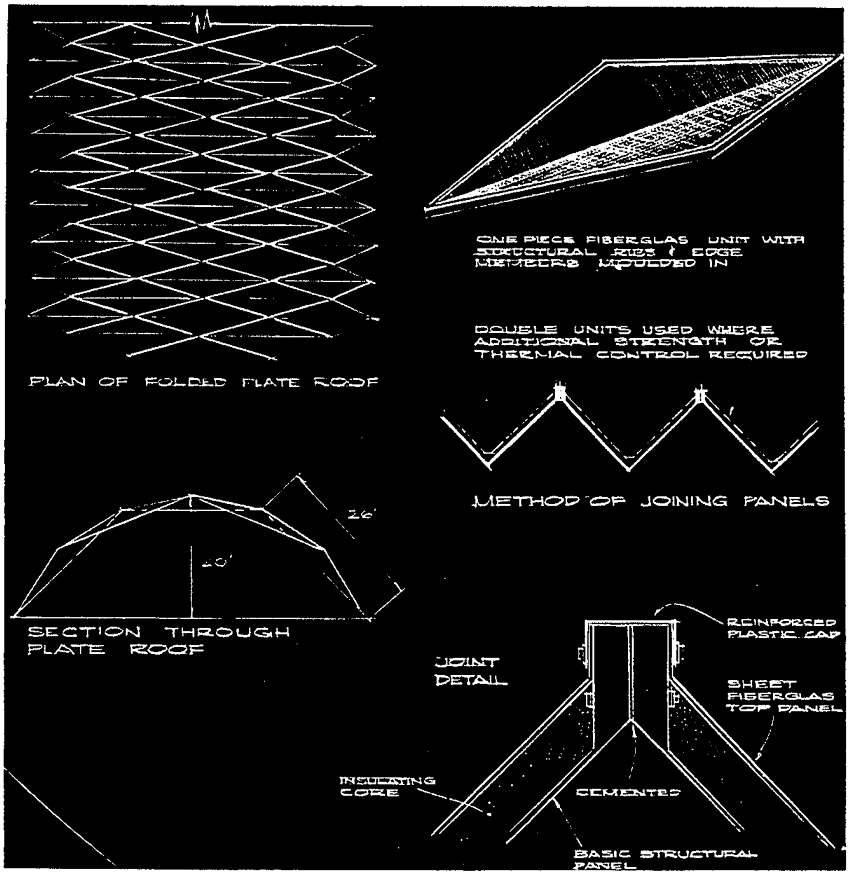


FIGURE 51.—Folded-plate design. (*Courtesy of Owens-Corning Fiberglas Corp.*)

roving and resin 0.3 in. thick. Another unitized design, shown in figure 54, is for portable shell units for motels.

"Buildings of the Future" include a home design with the basic roof and floor structures consisting of quarter moldings of fiber-reinforced plastic bonded together to form modules of shell construction; flat fiber-reinforced plastic panels and glass provide the other surfaces. In addition, a swimming pool, diving board, sun lounge, garden furniture, and woven fencing are molded in fiber-reinforced plastic.

A dynamic new design for tall buildings made of lightweight steel (or aluminum) girders supporting prefabricated, fiber-reinforced plastic modules molded on the shell principle has also been published (ref. 50). Another imaginative design of the future is a school building

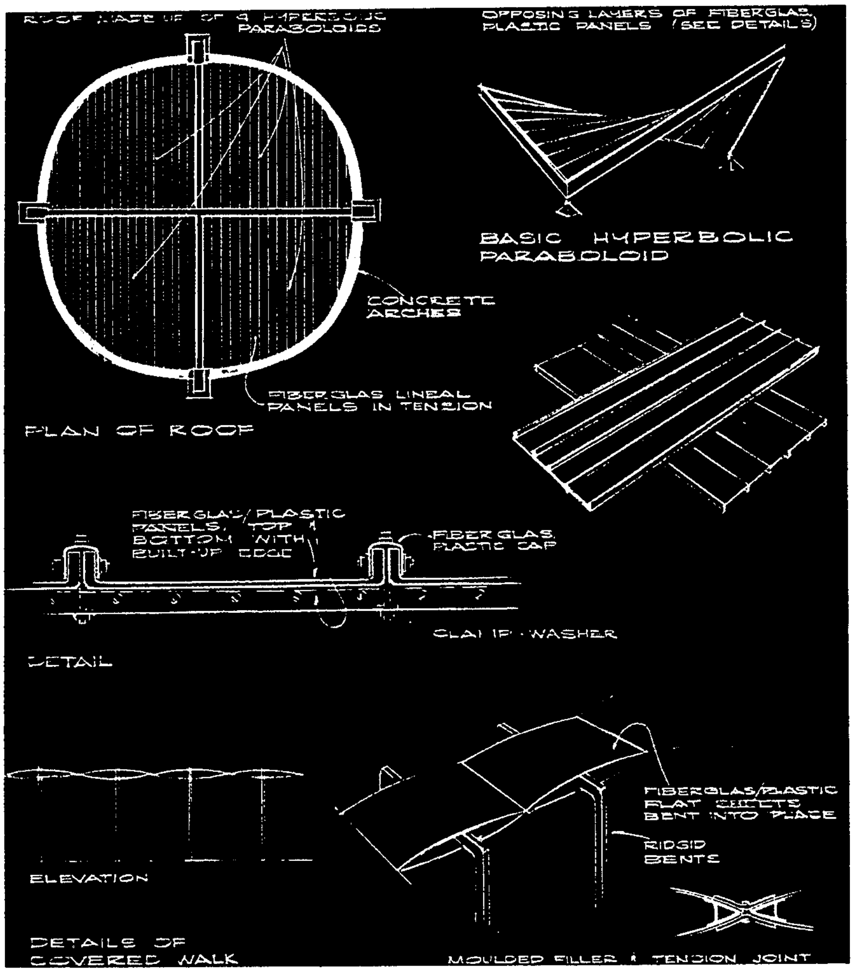


FIGURE 52.—Sheet-panel structure. (*Courtesy of Owens-Corning Fiberglas Corp.*)

with roof sections, supported on a stressed framework of fiber-reinforced plastic, forming a light, immensely weatherproof structure. Panels can vary from translucent to opaque, whereas the walls are clad in a sandwich construction of pigmented fiber-reinforced plastic with a core of insulating material.

Some other treatments of futuristic houses are filament-wound glass houses (shown in fig. 55 from ref. 51), which could be produced in round, conical, doughnut, or mushroom shapes by using filament-winding techniques. Double-wall construction, decorated with permanent, nonfading color, would provide insulation, and walls would

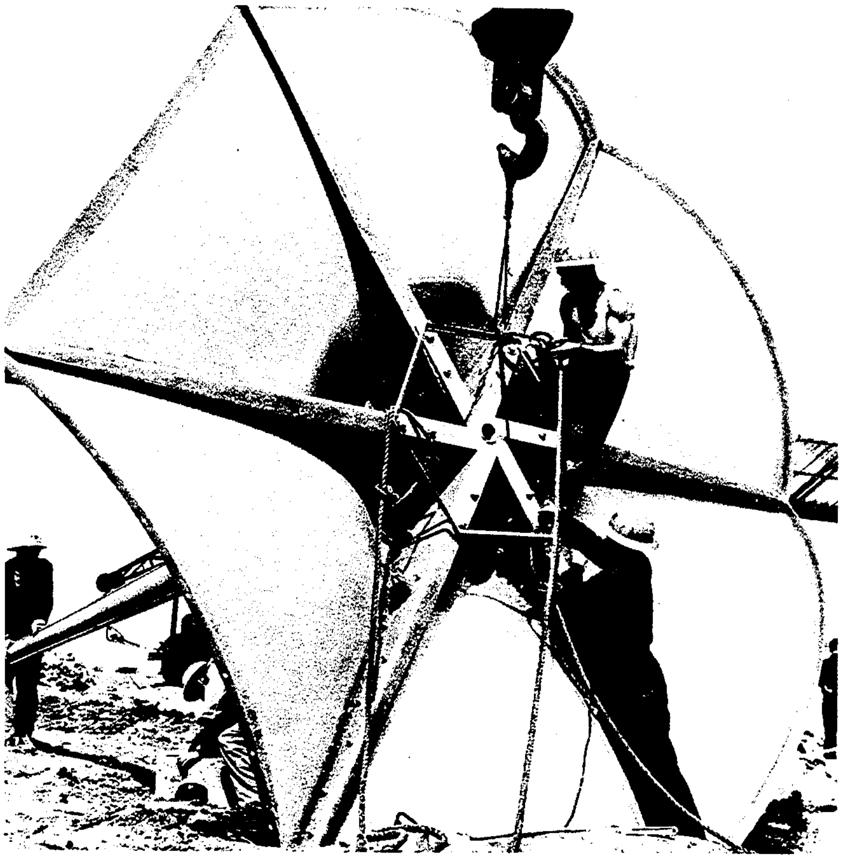


FIGURE 53.—Formed-skin structure.

be translucent to eliminate dark corners during the day. The entire structure could also be fabricated on the building site, once winding machines capable of such large tasks are developed.

An innovation not yet seen in this country is the isostatic-ribbed plate. As the reader knows, a plate is capable of developing grid action in any direction. Furthermore, any point in the plate may be considered as the intersection of two beams of a rectangular grid system, and any number of rectangular grid systems may be considered to be passing through a plate point. At each point there are two directions for which the bending stresses are, respectively, maximum and minimum, and for which the shear stresses are zero. If one indicates the principal stress directions at various points by crosses, principal stress-line patterns, or isostatics, may be plotted; these

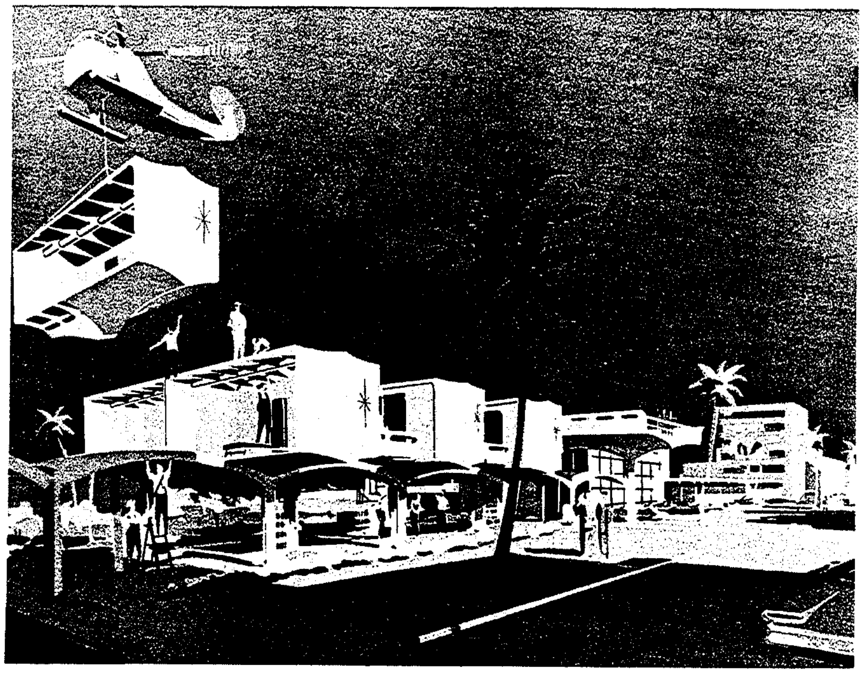


FIGURE 54.—Fiber-composite structures. (*Courtesy of Owens-Corning Fiberglas Corp.*)

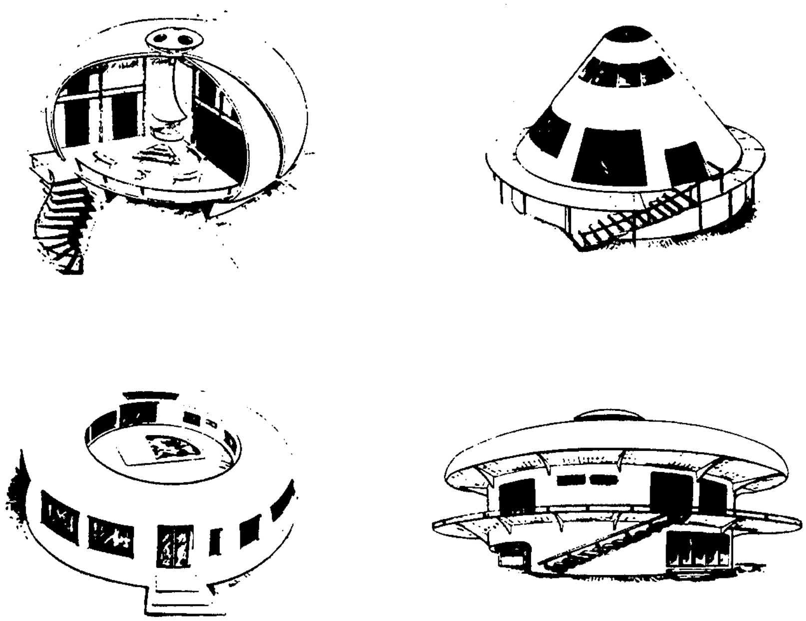


FIGURE 55.—Futuristic houses spun by large filament-winding machine.

represent the flow of stress in the plate. The pattern, of course, depends on the support conditions at the plate boundary and the loading. Since the shear stress is zero along isostatics, the plate may be visualized as a grid of curved beams that intersect at right angles and do not transmit loads to adjoining beams by shear action.

Isostatic lines form very interesting patterns and have been used by Nervi in floor designs (see frontispiece). The appearance of a ribbed plate can be made very attractive by constructing the ribs along isostatics.

These are examples of building ideas that aerospace research may help bring into wider use.

CHAPTER 5

Design Synthesis and Optimization

Design is the process of evolving a configuration to perform specific functions. It proceeds from the abstract to the concrete, and the initial concept is a relationship among ideas or geometrical forms. A final engineering design results from a series of problem-solving steps by which the configuration is evolved. This sequence of operations, called the "design process," carries a problem through analysis, synthesis, and evaluation and decision into optimization, revision, and implementation (ref. 52). In most engineering situations, the design process involves a number of successive iterations rather than a single direct solution of a closed-form equation that expresses the primary design problem (ref. 53). In general, the designer selects a configuration, analyzes it, selects another, analyzes it, selects a third, and so on; he also sets cost schedules and evaluates performance implications of alternate structures. In other words, the designer inputs the design requirements and makes systematic variations in concept, material, and detail to arrive at a set of designs that satisfy the design criteria and performance requirements.

To see the difference between the analysis and the design of a structure, suppose that the total geometry and load conditions are known and the material is given in the analysis. Then it is necessary only to analyze for structural behavior under the given load conditions. In the design process, however, only performance requirements are given; the geometry is not generally known. Thus the designer must generate a structure to meet specified requirements.

Synthesis is the fitting together of parts or separate concepts to produce an integrated whole. Because of the complexity of the process, constant revisions and reevaluations of the revised results often must be made and newly developed information added to the design considerations. Although the synthesis step formally begins after the design problem is well understood, some hypothesis for possible solutions will probably be suggested during earlier steps.

When the major design parameters must be set at specific values, optimization is used to find the best combination of parameter values to satisfy design requirements. This may be done mathematically, in which case all considerations associated with selection of the prime solutions are set forth in an equation called the "criterion (merit) function."

The mathematical description of a design problem involves input (independent) and output (dependent) variables associated through transforming mechanisms analytically expressed. If all the variables and design parameters are known, one can calculate the criterion value that will give a measure of the excellence of a particular design. In general, we must compare a particular design choice with other possible choices to determine its excellence.

While a complete exploration of all physically realizable design parameters could be made, and the best set determined by elimination, the usual design situations have two constraints: functional and regional. Functional constraints essentially constitute the mathematical description of the archetype of the proposed design. Regional constraints set the allowable limits on design parameters, or on derived groups of parameters, representing more complex attributes of the proposed object. Thus, in the general optimization problem, three factors must be considered: (1) the criterion function, which is brought to a maximum or minimum, depending on which corresponds to an optimum through proper choices of design parameters; (2) the functional constraints; and (3) the regional constraints.

To state the problem mathematically (ref. 52), let us lump all of the variables together, so that the set $(X_1, \ldots, X_n)$ contains the design parameters, input variables, and output variables. The criterion, represented by U, takes on the values of the criterion function $U(X_1, \ldots, X_n)$; the set of functional constraints is represented by $(\psi_1, \ldots \psi_m)$; and the regional constraints by $(\phi_1, \ldots, \phi_p)$.

For the regional constraints, the ith one will be constrained between the lower limit l_i and the upper limit L_i.

Thus, the optimization problem is described by the following set of equations:

$$U = U(X_1, \cdots, X_n) \rightarrow \text{optimum}$$

$$\psi_1 = \psi_1(X_1, \cdots, X_n) = 0$$

$$\vdots$$

$$\psi_m = \psi_m(X_1, \cdots, X_n) = 0$$

$$l_1 \leq \phi_1(X_1, \cdots, X_n) \leq L_1$$

$$\vdots$$

$$l_p \leq \phi_p(X_1, \cdots, X_n) \leq L_p$$

To give a geometrical interpretation of the analytical description, let us consider a three-dimensional space in which we can plot two variables (X_1,X_2), since higher order spaces, or hyperspaces, are impossible to visualize. Let X_1 correspond to the x-axis and X_2 to the y-axis; the criterion value, U, corresponds to the z-axis. In order to plot n variables, X_n, an $(n+1)$-hyperspace is required. The X_1- and X_2-axes form a basis plane over which the criterion (merit) functions, $U(X_1,X_2)$, are constructed.

The functional constraint $\psi_1(X_1,X_2)=0$ describes a curve on the basis plane. Projection of this curve onto the criterion surface gives a curve in space corresponding to $\psi_1(X_1,X_2)=0$; while at the same time, the curve conforms to the surface $U(X_1,X_2)$. This space curve "rises" or "falls" depending on the shape of U. If the optimum corresponds to a maximum, the highest point on the curve is sought; if it corresponds to a minimum, the lowest point is sought. In either case the X_1,X_2 coordinates of the optimum point will satisfy the constraint ψ_1, since the projection of its curve on the surface, U, was followed. If there is a second functional constraint, $\psi_2(X_1,X_2)=0$, the two curves corresponding to the two constraints will intersect at a particular point, or possibly at several points on the basis plane. In an n-dimensional problem, ψ_1 and ψ_2 will intersect to form a new hypercurve that will still be on the n-dimensional basis plane.

Now if we project the curve and the n-dimensional basis plane to the criterion function surface, a curve is traced on the latter surface. If we move along the projected curve, we shall rise or fall according to the shape of the criterion function surface. As before, the highest point on the curve is sought for a maximum; the lowest point, for a minimum.

Consider now the applications of regional constraints on a two-dimensional basis plane. The relation $\phi_1(X_1,X_2)$ represents a family of curves on the X_1, X_2 basis plane. Of this family, one extreme is $\phi_1(X_1,X_2)=l_i$, and the other is $\phi_1(X_1,X_2)=L_1$. Projection of the region between the two extreme curves to the criterion function surface defines a region on the surface that may be explored for an optimum. The introduction of a second regional constraint $l_2\leq\phi_2(X_1,X_2)\leq L_2$ leads (upon projection) to another region on the criterion function surface that may be explored for an optimum. The two curved strips of area will intersect on the basis plane to form a four-sided area, each of the sides being a segment of one of the limiting curves. Projection of this area onto the criterion function surface maps a specific region in which the optimum can be found. If other regional constraints are added, a region of many sides (equal to the number of constraints) must be projected onto the criterion function surface.

273-140 O - 67 - 6

STRUCTURAL DESIGN-SCIENCES APPROACH

Although stringent demands may be placed on materials to achieve overall design efficiency, it is important to utilize materials effectively. Although many aspects are involved in the effective utilization of materials, particularly under severe environmental conditions, three basic factors are summarized in figure 56 (from ref. 54): design, structure, and materials. The optimum design of a system requires consideration of all three factors simultaneously.

Aircraft, spacecraft, surface ships, submarines, and other vehicles have configurations, overall loads, and leading dimensions specified within rather narrow limits by performance requirements (ref. 55). The structural designer has some freedom within the confines of the leading dimensions to subdivide the structure with suitable stiffening systems to achieve a minimum-weight design (see fig. 57), but he must select materials that meet the particular structure and design conditions.

The design-sciences approach synthesizes the statement of the

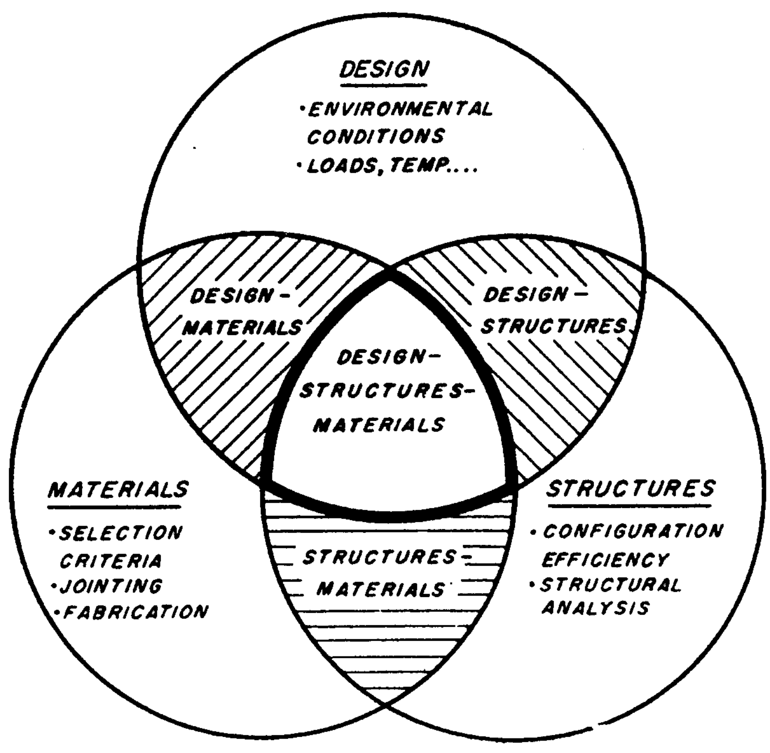


FIGURE 56.—Nature of the interplay.

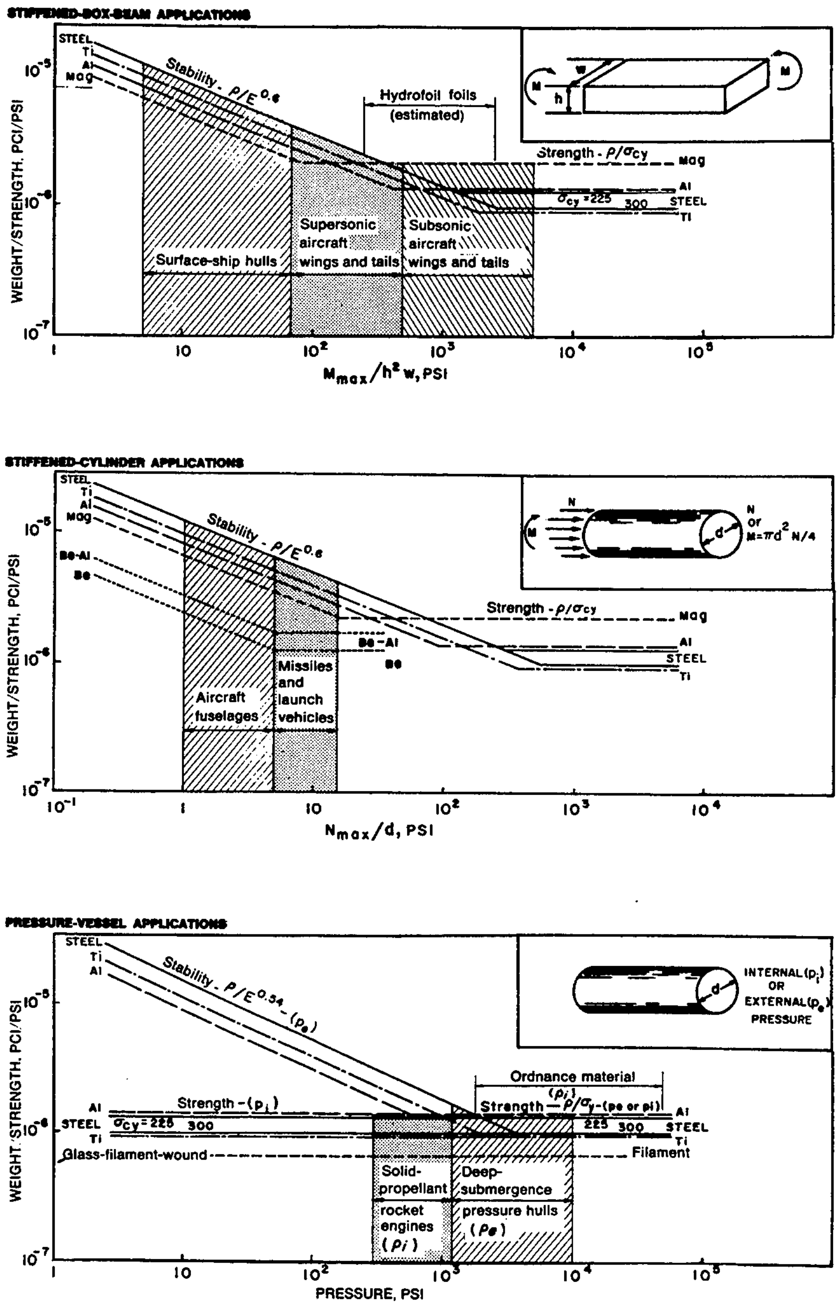


FIGURE 57.—Examples of design-sciences approach. (*Reprinted from* Astronautics and Aeronautics.) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

design problem by using certain design indices to combine the external loads and leading dimensions. It then uses idealized structural configurations, such as stiffened box beams and stiffened cylinders, to establish optimum designs from which to evaluate the comparative efficiencies of various materials.

This approach reached maturity within the last two decades. It permits the engineer to establish significant design parameters to evaluate the efficiency of various structural configurations and materials, using minimum weight as the criterion of optimum design.

As a broad generalization, the results of various types of minimum-weight analyses of representative structures can be expressed in the form:

$$W=SMD^m$$

where

W=weight efficiency factor
S=structural efficiency factor
M=material efficiency factor
D=structural design index
m=exponent $(0<m\leq 1)$

Although the weight efficiency factor, W, can be interpreted in several different ways, it is expressed in the form of a weight/strength ratio in the aerospace field. The structural efficiency factor, S, is generally a nondimensional quantity, whereas M is generally a density/modulus ratio representative of the material efficiency. The design conditions involving the external loads and leading dimensions are characterized by the structural design index, D. Thus, this equation, shown graphically in figure 57, represents the interrelationship among structures, materials, and design. Examples of the design-sciences approach follow:

Box-Beam Structures

Surface-ship hulls, aircraft wings and tails, hydrofoils, Army combat vehicles, and military bridges can be characterized in idealized form as stiffened box beams under bending loads. (See fig. 57 from ref. 55.)

Longitudinal stiffeners are I, Z, or hat sections, supported by transverse stiffeners that are transverse ribs at optimum spacing.

$$W=\frac{\text{weight}}{\text{strength}}=2.38\left(\frac{\rho}{E^{0.6}}\right)\left(\frac{M}{h^2w}\right)^{-0.4}$$

stability limitation due to buckling

with

$S=2.38$
$M=\rho/E^{0.6}$
$D=M/h^2w$
$m=-0.4$

When the buckling strength equals or exceeds the compressive yield strength of the material,

$$\frac{\text{weight}}{\text{strength}}=\frac{\rho}{\sigma_{cy}}$$

Therefore,

$$M=\frac{\rho}{\sigma_{cy}}$$

when strength limitations govern, or

$$M=\frac{\rho}{E^{0.6}}$$

when stability limitations govern.

Stiffened-Cylinder Structures

Aircraft fuselages, missiles, and launch vehicles can be idealized as stiffened cylinders under bending and axial compression, respectively. (See fig. 57.)

The longitudinal stiffeners are I, Z, or hat sections, supported by transverse frames which are I, Z, or hat sections.

$$W=\frac{\text{weight}}{\text{strength}}=1.25\left(\frac{\rho}{E^{0.6}}\right)\left(\frac{N}{d}\right)^{-0.4}$$

with

$S=1.25$
$M=\rho/E^{0.6}$
$D=N/d$
$m=-0.4$

Also,

$$\frac{W}{S}=\frac{\rho}{\sigma_{cy}}$$

Pressure Vessels

Submarine pressure hulls, solid-propellant rocket motors, and various ordnance can be treated as pressure vessels under external or internal pressure.

For an I-, Z-, or hat-frame-stiffened cylinder ($L/D=1$),

$$W=\frac{\text{weight}}{\text{strength}}=1.5\left(\frac{\rho}{E^{0.4}}\right)(p)^{-0.46} \qquad \text{external pressure}$$

Also, $W/S=\rho/\sigma_{cy}$, the strength-limitation region, is valid for both internal and external pressure.

Further Development of the Approach

The foregoing examples demonstrate the application of the design-sciences approach to the evaluation and improvement of material properties in terms of their potential applications. This approach was also used to identify structural design limitations in the case of deep submersibles.

If we accept improved weight/strength efficiency as a desirable goal, we can summarize potential improvements, as shown in figure 58. Here, if a given design application has a design-index range corresponding to D_1, then design, structures, and materials (density and elastic modulus) improvements can lead to greater weight/strength efficiency. On the other hand, if the design-index range corresponds to D_2, only material improvements (density; yield or ultimate, strength; ductility) can contribute to weight/strength efficiency.

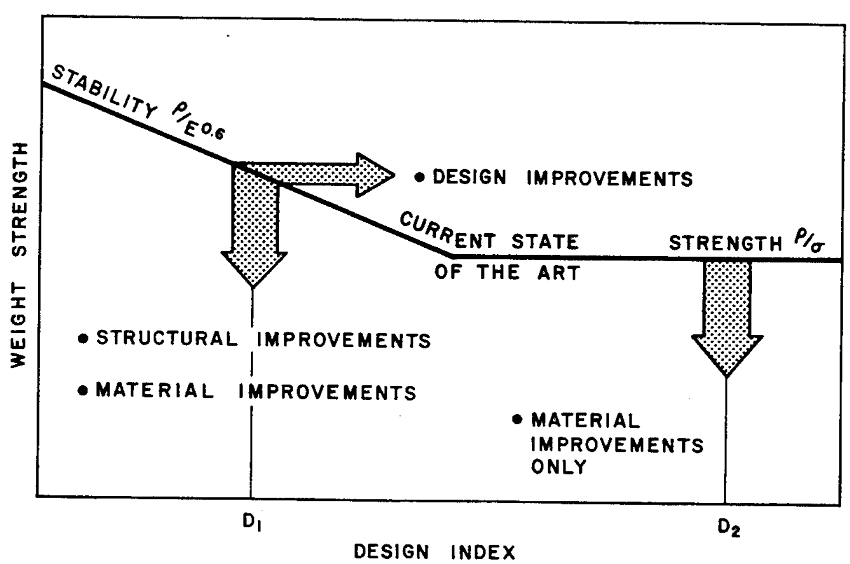


FIGURE 58.—Potential improvements in current state of the art. (*Reprinted from* Astronautics and Aeronautics.) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

Although conclusions concerning improved materials depend on the design-index range corresponding to the application, the surprising fact is that the various applications indicated in figure 57 are characterized by rather narrow design-index ranges. This permits valid conclusions regarding current designs to be drawn from this approach and also permits rather safe conclusions for future designs. As a result, this approach can help in technical decisions for long-range material-development cycles.

The design-sciences approach is reasonably well developed in certain aspects and can be effectively employed in the following areas:

(1) To evaluate current and experimental materials over a broad temperature range extending from cryogenic to elevated temperatures

(2) To provide guidelines for identifying and developing improved material properties for projected applications

Further investigations in the following areas could greatly advance this approach (ref. 55):

(1) Engineering studies of structures to provide design-index data for current and projected applications

(2) Determinations of why applications fall within a narrow range of design-index values

(3) Study of various design configurations to alleviate or remove the design limitations

(4) A project relating minimum-weight results to cost for optimum structures

FULLY STRESSED DESIGN

For a structure under multiple-loading conditions, the method of fully stressed design proportions the structural members by equalizing the allowable stress in any member in at least one loading condition (ref. 56). If analysis shows that a certain member is overstressed in a critical load condition, the method of fully stressed design increases the area of that member enough to remove the overstress. Conversely, this method does the opposite if the member is understressed.

For structures with so-called hybrid action, each member must be designed with consideration of its effect on other members. For this type of structure, the convergence is generally slow; and the resulting repetition of analysis and fully stressed redesign often tends to simplify the structure by eliminating some of its members.

The minimum-weight design of a structure is an arrangement of the structural element in which all the design requirements (such as stresses, deflections, and geometric constraints) are satisfied, while the total weight of the entire structure is minimized. This minimum-weight design can generally be set up as a mathematical programing problem. Efficiency of the fully stressed design and its relationship to a minimum-weight design has been discussed by a few investigators. Although Schmidt (ref. 57) has argued that a minimum-weight design may be selected from among fully stressed designs, Schmit (ref. 58) has shown that a fully stressed design is not necessarily a minimum-weight design. Under some loading conditions, in fact, the fully stressed method may lead to an inefficient design. Razani (ref. 56) has sought to determine when a fully stressed design has minimum weight and when it has not; when it has not, he suggests a method of determining optimum structure.

The iterative, fully stressed design usually changes the configuration of the structure considerably in the initial cycles, but successive changes generally result in progressively fewer modifications.

In the method of fully stressed design, the problem of convergence is studied within the range where changes in area or stiffness of structural members are small. It is assumed, in addition, that the critical loading condition for each member does not change abruptly because of a small change in design configuration; thus, the critical forces in the members can be treated as continuous functions (ref. 56).

Relationship of Fully Stressed and Minimum-Weight Designs

For determinate structures (see fig. 59) then, the fully stressed design is the minimum-weight design; whereas for indeterminate structures, the critical force in each member is not only a function of the applied loading but also a highly nonlinear function of the areas of all the members of the structure. Consequently, the fully stressed design is not always an optimum design.

Condition of optimality

$$\lambda=(I-B^T)^{-1}\rho L>0 \qquad \text{Kuhn-Tucker optimality conditions}$$

where

λ's=optimality coefficients

$B=m\times m$ design variation matrix, $B=(b_{ij})$

$b_{ij}=\left(\frac{1}{\sigma}\right)\left(\frac{\partial \tilde{F}_i}{\partial A_j}\right)=\frac{\partial A_i}{\partial A_j}$

B^T=transpose of matrix B

$I=m\times m$ unit matrix

ρ=material density or unit weight of material

L=length of section

$\tilde{F}_i$=critical load of ith member

σ=corresponding stress for the critical load

A_i=area of ith member

m=number of members

When the fully stressed design is not optimum, the productivity test can be used to determine and separate the free variables from the fully stressed ones (refs. 56 and 59),

$$P_i=\frac{\partial V}{\partial A_i}\cong L_i+\sum_{\substack{j=1\\ j\neq i}}^{j=m}(A_j^n-A_j^o),\quad i=1,2,\ldots,m$$

where

P=productivity coefficients
A_j^n=final area of jth member obtained by an iterative, fully stressed design while keeping the area of the ith member constant and equal to $A_i^o+\Delta A_i$
A_j^o=initial area of the jth member before change in the ith member
ΔV=total change in the volume of the truss due to a change ΔA_i—the ith member

In this case, dimensionality of the problem is reduced and optimization is decentralized to an optimal search for free variables and to the fully stressed design of the remaining variables. In general, the faster

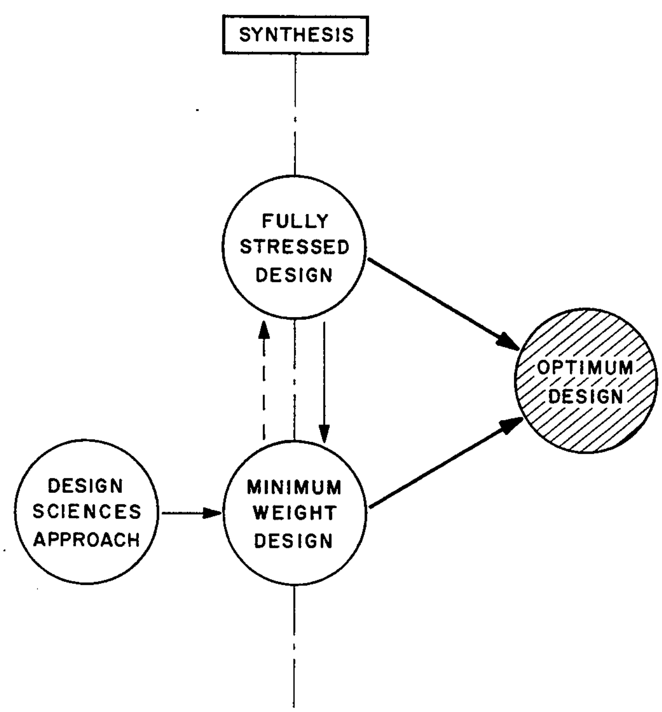


FIGURE 59.—Relationship of design steps.

the convergence rate of the iterative, fully stressed design, the more likely the optimality of the design. Consequently the fully stressed design of structures with normal action is more likely to be optimum. We, therefore, have another approach for structural-design optimization.

STRUCTURAL OPTIMIZATION METHODS

We may divide the classical numerical optimization methods into three general groups: (1) "Perturbation Methods," which include the indirect methods of Adjoint Functions and Perturbation Functions; (2) the "Quasilinearization Methods," which also include the indirect methods of the "Generalized Newton-Raphson," a "Modified Generalized Newton-Raphson," and the "Modified Quasilinearization"; and (3) "Gradient Methods," which are direct methods including the "Method of Steepest Descent" and the "Modified Method of Steepest Descent." An excellent paper which analyzes and compares these conventional methods was recently given by Lewallen and Tapley (ref. 60). We shall briefly discuss one of the more recent methods devised for structural optimization, called the "Random-Sampling (RS) Method."

Random-Sampling Method

The general problem of optimization with arbitrary constraints is treated by means of random numbers and Monte Carlo sampling techniques. Kiciman (ref. 61) demonstrated the validity of the technique by comparing his results on structural synthesis problems with published results using the gradient method. Although the design configurations produced by this approximating technique are not better than those given by the gradient method, they are acceptably close. A specific application to the minimum-weight design of a Z-stiffened compression panel is also given and the results checked against values computed by the designer using conventional methods. Indications are that this generalized and readily applicable synthesis approach will enable the designer to investigate several different design concepts for their relative design values without waste of time and effort. Two main advantages of this technique are: (1) no restrictions are placed on any of the constraint and merit functions, and (2) any number of variables and constraint conditions can be used.

Application of Random-Sampling Method

Structural synthesis (ref. 61) can be defined as rational selection and improvement of a structural design configuration, in terms of weight or cost, without violating given failure conditions, manufacturing, or design limitations. The conventional way of designing an efficient structure is based on the designer's past experience, good judgment, and trial and error until a satisfactory solution is found.

The basic rationale for applying random-sampling (RS) methods to structural synthesis problems is the similarity in method between an RS-type solution and the conventional solution, previously described. Another point in favor of an RS method is that it can be applied to almost any type of structural synthesis problem with little statistical theory. Finally, since the method is a completely random procedure, those using it cannot be handicapped by prejudices or oversights unless purposely biased by the programer.

For the sake of illustration, let us assume that the structural part to be designed has three variables of thickness, spacing, and height, each with given limitations. This design can be expressed as $X = X(\xi_1, \xi_2, \xi_3)$, where ξ_1, ξ_2, ξ_3 are thickness, spacing, and height, respectively. The requirements are given as the local and general stability for a given load; the merit function is the weight.

This problem can be solved by checking all possible design configurations, that is, all the distinct X's for local and general instability, and selecting the one with the minimum weight among the stable configurations. However, ξ_1 is a continuous variable that can take any value between $\xi_{1_{min}}$ and $\xi_{1_{max}}$ making the number of distinct X's infinite. In practice, however, the interval $(\xi_{1_{min}} - \xi_{1_{max}})$ can be divided into a finite number of slices, assuming that ξ_1 is a variable that can take only a given number of values between minimum and maximum.

Assuming that thickness, spacing, and height can be divided into 100 slices each, the number of distinct design configurations is 1 million. Since there are two stability conditions in addition to weight, 3 million computations are the number of points theoretically to be checked. The function of the RS procedure is to cut the number of computations to an economical few hundred. Some of the sampling steps used are described below.

Importance sampling gives a way of biasing the random sampling so that some samples are drawn from zones where the probability of success is high, and less from zones where the probability of success is small. In other words, biasing is done to increase the probability that the sample will be drawn from an interesting region (ref. 62).

The combination of analytical and probabilistic solutions sometimes reduces the variance of the outcome; therefore, the optimum sample size is computed for some of the steps whenever that can be done without excessive effort.

If the density function of the random sample is approximated from the initial trial with respect to certain sections of the sample space, this function is used to assign a certain size of sample to each section for consecutive trials.

Basic Screening Steps of the Program

The problem consists of locating a design point in the space of all possible design points, so that all the design requirements are satisfied and the evaluated merit function is as close to its optimum value as desired. Such a design point is denoted as X^*. Initially, then, we have

$$P(X=X^*)=P(X\epsilon S^*) \qquad \text{if} \quad X\epsilon S$$

S is the n-dimensional space of design variables where n is the number of variables for the particular problem. The position of each design point X in this space is specified by the value of its coordinates ξ_k:

$$X=X(\xi_1, \xi_2, \ldots, \xi_n)$$

The boundaries of the design space are specified by the minimum and maximum allowable values of the design variables. In the literature these boundaries are commonly referred to as side constraints.

In design problems the number of significant digits is limited for practical reasons; therefore, the random variable ξ_i can take only discrete values.

A design point is considered unacceptable if it violates any of the constraint conditions. Thus the only requirement for the g_i is that it must have a computable value for every X in S.

The concepts mentioned so far have been illustrated in a problem having two variables. (See fig. 60 from ref. 61.) The boundary between U and A is designated by G, which represents a hypersurface having concave and convex portions. By this program, random points are chosen and checked against the given constraints to determine whether they are in A or U; this checking continues until no point in A can be found with a merit function lower than the previous one. The merit function $F(X)$ is the function to be optimized. It has a unique value for every X, which is computable. To improve the probability of success with a minimum number of computations, a system of sampling techniques is utilized (described in ref. 56). This operation is based largely on the assumption that the evaluation of main constraints for a given X demands an effort much greater than the computation of merit function for that design point; therefore, X must be avoided as much as possible, and the information obtained from the merit function values must be fully used.

COMPUTER-AIDED DESIGN OF STRUCTURES

Let us consider the possible utilization of computer capabilities to make design decisions more rapid and effective (ref. 53). In the past, engineers have carried out much of their design and practice

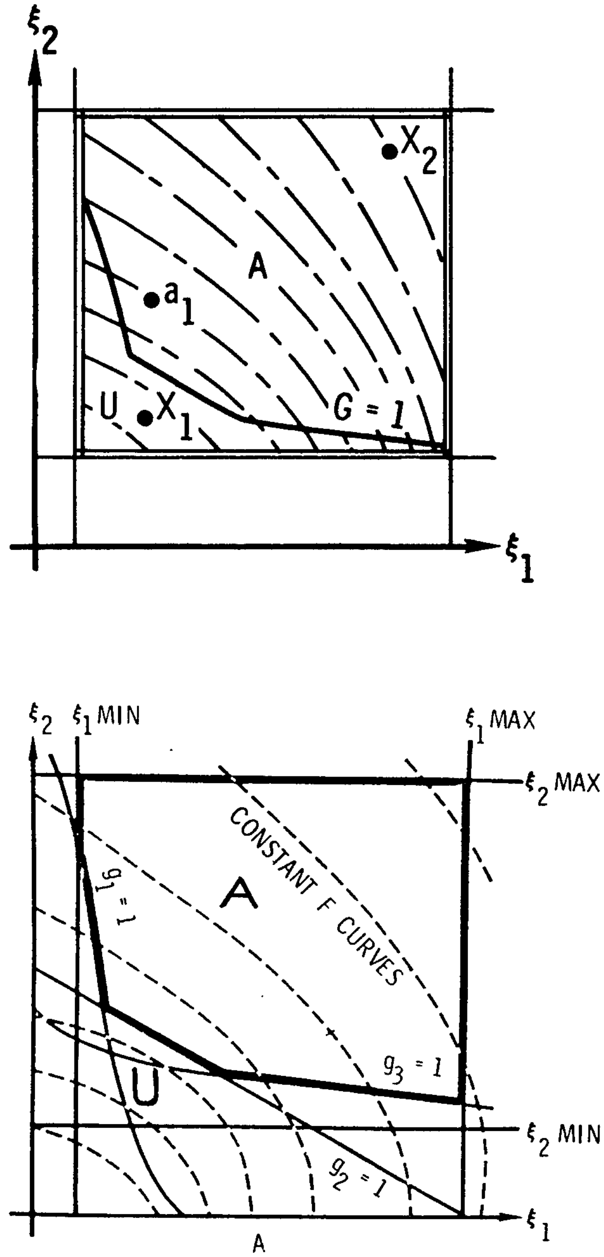


FIGURE 60.—Search for zone of optimum design. (*Reprinted from Proceedings 4th Aerospace Science Meeting.*) (*Courtesy of American Institute of Aeronautics and Astronautics.*)

efforts by analytical investigations using computers. The engineer, for example, determines the response of a given structure under applied loads and compares the behavior to allowable criteria. Generally, this design process involves a number of successive iterations. Although it is conceivable that a design can be made by direct solution of a close-form equation that expresses the primary design problem, the difficulties associated with the expression of design-problem parameters make use of this process very unlikely in the near future. Rather, the computer can be used as an effective design tool for analytical techniques, since it allows rapid synthesis by iteration.

In the past, designers often had the solutions to a limited number of discrete problems compiled from experience. With increased knowledge, experience, and the aid of high-speed, electronic digital computers, today's designer can execute the design process with greater effectiveness. Needless to say, many problems formerly beyond the designer's capabilities can now be solved.

In the preliminary design process, furthermore, several simplified techniques enable the computer to generate considerably more data

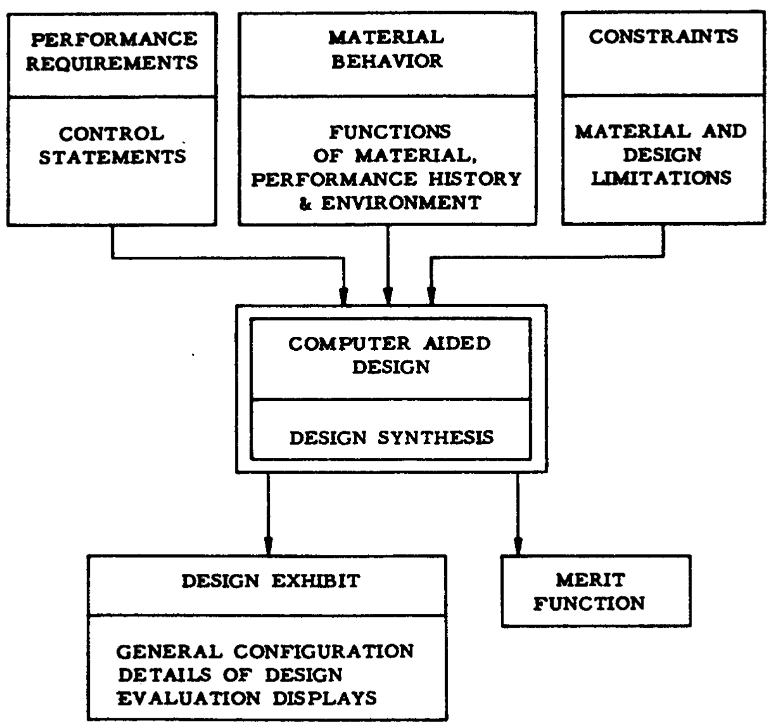


FIGURE 61.—General logic flow of computer-aided solution.

than the engineer. A computer can also exhibit these data in the form of drawings and specifications. Figure 61 shows the logic flow of computer-aided solutions. In table 2 (from ref. 53) a summary-comparison of computer-aided design procedures is given with applications to multistage launch vehicles, bridges, and domelike structures.

TABLE 2.—*Comparison of Computer-Aided Design Procedures*

Launch vehicle	Bridges	Domelike structures
Performance requirements:		
Velocity requirements.	Number of traffic lanes.	Specified enclosed volume of surface area.
Specified payload weight in specific (orbital) mode.	Magnitude and distribution of loads.	Specified maintained environment in the enclosed volume.
Type of mission (scientific vs. military; manned vs. unmanned).	Clearance height for vehicles.	Specified esthetic requirements.
	Topographical conditions.	Structural provisions to resist specified type, magnitude, and distribution of loads.
	Subsurface conditions.	
	Arterial approach requirements.	
Material behavior properties:		
Strength/density relationships.	Strength/density relationships.	Strength/density relationships.
Elastic/density relationships.	Elastic/density relationships.	Elastic/density relationships.
Chemical compatibility factors.	Corrosion-resistant properties.	Absorptivity/emissivity properties.
		Thermal insulation/weight relationships.
		Acoustic insulation/weight relationships.
Constraint functions:		
Acceleration limits.	Feasible geometric proportions.	Minimum height as a function of distance from dome perimeter.
Feasible stage diameters and fineness ratios.	Navigation clearance.	Upper and lower bounds on temperature and humidity.
Time element for design (i.e., 1965 vs. 1970 type structure systems).	Minimum traffic lane width.	Upper and lower bounds on acoustic characteristics.
Manufacturing-procurement feasibility of components.	Applicable specifications of the American Association of State Highway Officials.	Fabrication limitations.
	Gradient limitations.	
	Dynamic response.	

TABLE 2.—*Comparison of Computer-Aided Design Procedures*—Con.

Launch vehicle	Bridges	Domelike structures
Least cost components. Manufacturing, R&D time scheduling. Launch environment. Recovery problems. Minimum gage restrictions.	Aerodynamic response. Fabrication limitations. Construction period limitations (time and environment). Construction time period. Depreciation method.	Maximum construction time. Construction time period. Amortization method.
Scope of design investigation:		
Vehicle geometric models (configurations). Construction concepts (components). Variations in strength/density of materials. Effects of loadings induced by various trajectories. Pressure and temperature variations due to flight loadings, trajectories, and system changes. Performance weight partials. Cost analysis (R&D plus Operational).	Variations in deck width as function of number of traffic lanes. Deck stacking concepts. Variations in span lengths. Variations in support concepts. Variations in profile gradients. Variations in anchorage configurations. Variations in strength/density ratios of principal structural materials. Variations in ramp concepts. Cost effects upon adjacent land areas.	Variations in aspect ratios (height/radius at base). Variations in meridian curvature properties. Variations in framing geometry. Variations in framing material. Variations in support concepts.
Design exhibit:		
Profile drawings. Design sketches. Component detail sketches. Master dimensions. Parts lists. Cost analysis weight statements. Detailed weight statements. Performance weight statements. Detailed geometry of components.	Drawings of bridge profile, cross sections, and elevation. Detailed geometry of components. Horizontal, vertical and torsional rigidities of bridge sections. Dynamic response properties. Aerodynamic response properties. Parts list. Excavation and	Drawings of dome cross section. Three-dimensional coordinate values of all space frame joints and other significant positions. Internal loads and stresses in all members. Load deflection of all significant joints. Aerodynamic response properties.

TABLE 2.—*Comparison of Computer-Aided Design Procedures*—Con.

Launch vehicle	Bridges	Domelike structures
Design conditions (loads, pressures, temperatures). Evaluation logic in selection of candidate vehicle evaluation curves).	foundation costs. Material costs. Fabrication costs. Erection costs. Maintenance costs. Operational costs. Total construction time.	Excavation and foundation costs. Material costs. Fabrication costs. Erection costs. Maintenance costs. Operational costs. Total construction time.
Merit function:		
Cost per pound payload in specified earth orbit or space trajectory as a function of number of launches within particular time period.	Cost per year of operation per vehicle ton capacity.	Cost per year of operation per usable unit enclosed, operating in the specified environmental condition.

Finally, let us emphasize one aspect of the design process that has not yet received particular attention: proposed, or baseline requirement, changes. Often changes are made in design criteria, design philosophy, geometrical constraints, and/or environmental conditions for one or more reasons. These changes may result in revised engineering drawings and specifications and, perhaps, in additional tooling and testing. Whatever the effects of a given change are, however, the objective of that change must be met. Since detailed analyses cannot be made to check the results of a proposed change, a tool is needed to assess it rapidly and efficiently. This can be a digital computer programed to synthesize a structure for loads imposed on the body for a specific function and to calculate a specific parameter, or its changes, in terms of other suitable parameters.

273-140 O - 67 - 7

Appendix

SELECTING MATERIALS FOR MINIMUM WEIGHT

Particularly in aerospace work (ref. 63), reduced weight means improved performance. Weight savings can also be important for autos, trucks, and railway cars, because a pound saved in the structure may permit a greater payload or increase general performance.

Since an aerospace vehicle cannot be designed for minimum weight alone, however, the designer must consider the environmental effects to which the structure may be subjected. No single material or construction can retain superior strength over the entire range of potential loads and temperatures. The optimum structure must consist of many materials, each suited for a particular job. In addition to strength and stiffness, the materials must be evaluated for cost, ease of fabrication, toughness, durability, and other properties.

Weight Index for Stiffness

Geometry is particularly critical in structures designed for stability and stiffness. Although stiffness will change with the geometry of the structure, the efficiency of a given geometry is proportional to the merit-weight index. This index can be calculated using the ratio of a material's modulus of elasticity to density (E/ρ).

In an aerospace vehicle, increases in stiffness are accompanied by increases in structural stability and natural frequencies. Thus, a high merit-weight index for stiffness (E/ρ) would indicate that the structure can efficiently handle static and dynamic problems of elasticity (e.g., aerodynamic response and flutter), acoustics (e.g., vibration fatigue), as well as load-carrying capacity.

Weight Index for Tensile Loads

Selection of the optimum material and construction is often made easier by using merit-weight indices. In a structure subject to short-time tensile loading, for example, the weight is inversely proportional to the (σ_{allow}/ρ) index, which is the ratio of allowable tensile stress to the density of the structure. This index is commonly known as the merit-weight index for tensile criteria.

An evaluation of the merit-weight index for tensile conditions as a function of temperature and time is shown in figure 62 (from ref. 63) for a few typical materials. This illustrative guide shows that an increase in time at temperature is equivalent to shifting the abscissa (time-temperature parameter) to the right, thus resulting in a decrease in strength.

The merit-weight index of a particular material is determined by the intersection of the curve with a vertical line, the lower end of which passes through the intersection of the appropriate time and temperature lines. In figure 62, which gives a typical example of how this index is obtained, titanium is shown to be the lightest of the materials considered for withstanding a tensile load after exposure to 800°/F for 100 hrs.

Plots similar to those shown in figure 62 can also be made for other merit indices, such as allowable rupture stress/density (σ_r/ρ), or allowable creep stress/density (σ_c/ρ) in areas of constant stress, such as the powerplant.

Except for orthotropic constructions such as filament-wound pressure vessels, the geometry of tensile-loaded structures is not too critical. This is generally true provided that good design practices, such as avoiding stress concentrations, are observed.

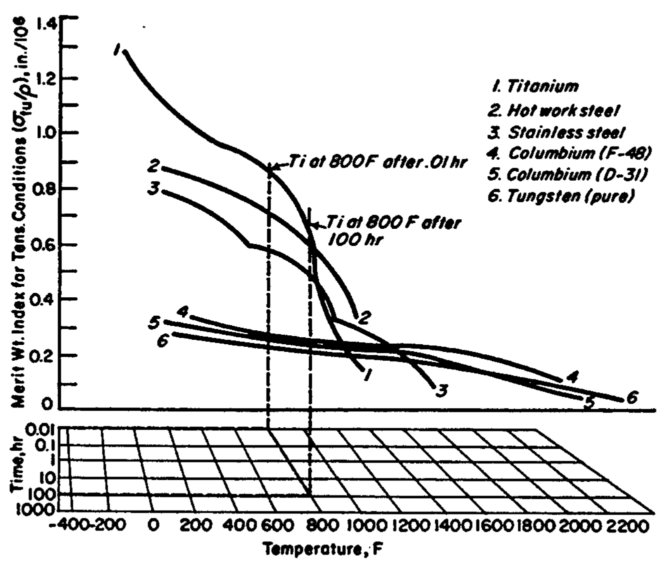


FIGURE 62.—Merit-weight indices of materials under tensile loading as a function of temperature and time. (*Courtesy of* Materials in Design Engineering.)

Weight Index for Compressive Loads

The weight efficiency of materials carrying compressive loads can be expressed by a merit-weight index for stability (σ_{cr}/ρ). Here σ_{cr} is the instability stress depending on the geometry of the structure , the load, and stress-strain relationship of the material.

Evaluations can be made independent of structural size by using weight (W/b^2) and load (P/b^2) indices, where W represents the weight per inch of the structural cross section, P represents the load, and b the characteristic dimension of the structure (e.g., width of plate or length of column). Thus, as shown in figures 63 and 64 (from ref. 63), typical plots of these indices can be derived for various materials and structural configurations. Such plots are an effective tool for selecting the right material and/or structural configuration.

Unstiffened Plate

A comparison of the weight and load indices (in the form of a log-log plot) of typical engineering materials is shown in figure 63. By mathematical analysis, the weight index of a material is proportional to the cube root of the load index at low load indices. The relationships can be represented by straight lines with a slope of 1/3 as a result of comparing materials. Thus, at low load indices, the merit index or efficiency

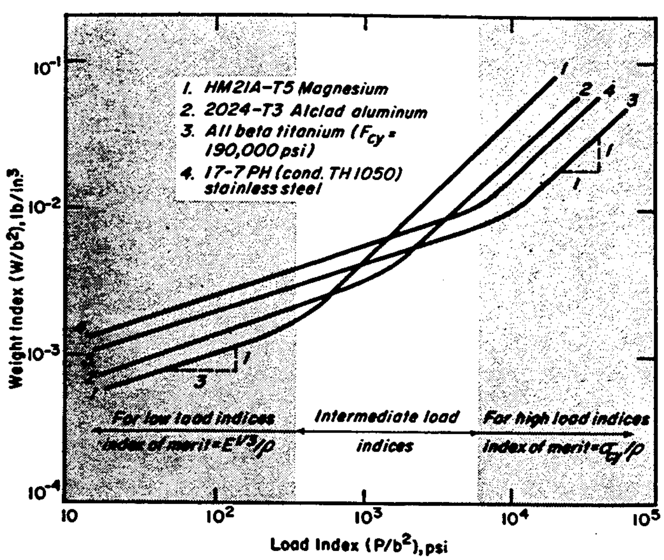


FIGURE 63.—Weight-load comparison of flat unstiffened compression panels at room temperature. (*Courtesy of* Materials in Design Engineering.)

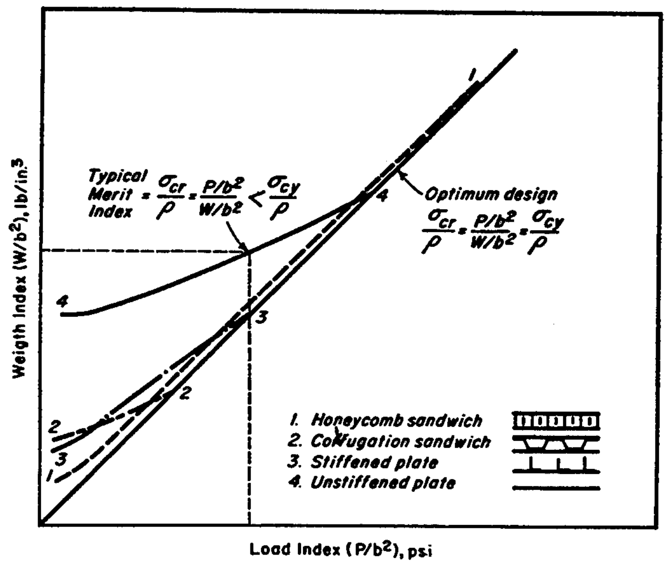


FIGURE 64.—Weight efficiencies of four configurations. Comparison is generalized for any material. (*Courtesy of* Materials in Design Engineering.)

of a material can be calculated by ($E^{1/3}/\rho$). (For double-faced corrugations, instead of unstiffened plate, the merit index or efficiency of a material would be ($E^{1/3}/\rho$).)

For high load indices, the straight lines assume a slope of 1/1, and the weight index is directly proportional to the load index. In this area the efficiency of the material can be measured by the allowable compressive yield stress/density (σ_{cy}/ρ). This index can be used for all types of construction.

Types of Construction

A guide to the weight efficiency of four typical structural configurations is given in figure 64. This comparison can be generalized for any material used in stiffened or unstiffened flat plate, honeycomb, or corrugated sandwich form.

For a given load in figure 64 the construction having the best weight efficiency is indicated by the lowest vertical ordinate. For any of the four constructions, the efficiency can be calculated by the merit index (σ_{cr}/ρ) (allowable stability stress/density), which is equal to the abscissa divided by the ordinate, $P/b^2{:}W/b^2$. This figure shows that the optimum stress efficiency is (σ_{cy}/ρ) (allowable compressive yield stress/density). Each of the four constructions approaches the optimum value asymptotically with increasing load indices.

Figure 64 also shows that at low load indices, a honeycomb sandwich

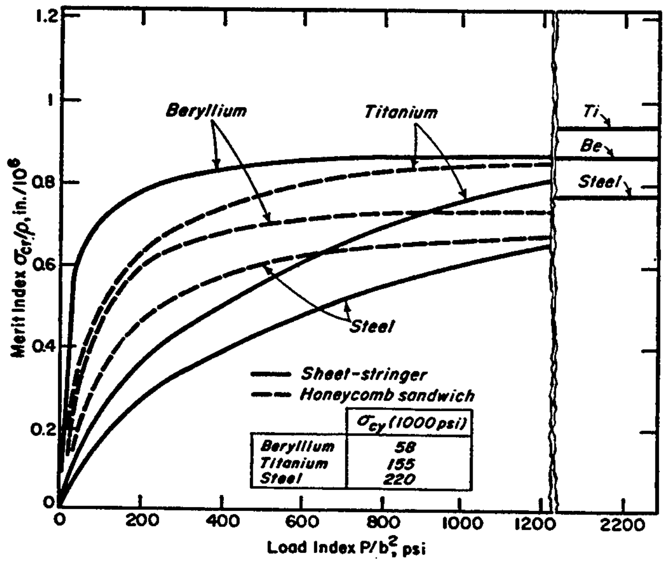


FIGURE 65.—Efficiency of beryllium, steel and titanium in two types of constructions at room temperature. (*Courtesy of* Materials in Design Engineering.)

construction is most efficient, since it is capable of stabilizing the faces to yield, although it suffers a slight weight penalty due to core and bonding materials. Keep in mind, however, that variations in efficiency can occur at very low loads because of minimum gage and fabrication requirements.

Next in weight efficiency is the corrugated sandwich, in which the stabilizing stress rapidly approaches the asymptotic yield stress. The stiffened steel-plate construction, in which efficiency increases with the number of stiffeners used, is next in efficiency; whereas the unstiffened plate is the least efficient structure, since it requires the largest load index to attain the yield stress. The relative efficiency of honeycomb sandwich can be offset by an extremely high stiffness merit index (E/ρ), such as is exhibited by beryllium.

Additional weight indices, given in figure 65 (from ref. 63), show the efficiency of steel, titanium, and beryllium when used in a honeycomb-sandwich or sheet-stringer construction. This study is far from complete; additional materials, configurations, and temperatures should be considered before making a final selection.

Other Important Factors

In choosing material and configuration for minimum weight, allowance must also be made for factors such as tolerances, available gages, and required design details (e.g., joints and reinforcements). In addition, the physical properties of the material have to be considered, and compromises often must be made in determining the best combination of properties.

For example, a low thermal expansion coefficient, α, is desirable for reducing thermal distortion. Similarly, a low product of elastic modulus and thermal expansion coefficient, $E\alpha$, will help reduce thermal stresses. Reductions in thermal gradients that result in thermal stresses require materials with a high thermal conductivity (k), specific heat (c), emissivity (e), or diffusivity (k/c_p).

Although minimum weight analyses are relatively simple when a monolithic structure is used, these procedures become more complex with composite structures. An aerospace vehicle, for example, may consist of a hot structure that supports the applied loads near the equilibrium temperature, or a protected structure which consists of a thermal protection system to resist local air loads, while maintaining an efficiently lower temperature on the load-carrying structure. When designing such a composite structure, consequently, tradeoff studies must be made to determine what combination of thermal protection and load-carrying structure will provide the minimum weight.

CALCULATING WEIGHT EFFICIENCIES OF PRESSURE VESSELS

Although unfired pressure vessels are often selected without regard to weight, many applications require them to be as light as possible. When low vessel weight is needed, it is important to know how the relative weight efficiencies of different materials compare when they are used in such common shapes as cylinders, spheres, and oblate spheroids.

Material and Shape Comparison

A basic formula that can be used in evaluating the weight efficiency of pressure vessel materials is

$$\frac{PV}{W}=\frac{1}{K}\left(\frac{\sigma}{\rho}\right)$$

In essence, this formula tells us that the product of pressure P and volume V divided by weight W is theoretically equal to a shape factor $1/K$ times the strength-to-density ratio (σ/ρ) of the pressure vessel material.

So as to make a direct weight comparison for vessels of the same pressure-volume (PV) capability, this formula can be rearranged as follows:

$$W=K\left(\frac{\rho}{\sigma}\right)PV$$

Since we want to keep the PV product constant when comparing materials, the formula simplifies to:

$$W \propto K\left(\frac{\rho}{\sigma}\right)$$

This formula tells us that for a given pressure-volume value, the vessel weight for a material is equal to its density/strength times a shape factor. The shape factor (K) is 2 for cylinders, 1.5 for spheres, and 3 for oblate spheroids.

This last formula can be used to make a side-by-side comparison, such as that shown in figure 66 (from ref. 63), which lists the relative weights, at constant pressure and volume, of three basic shapes fabricated from fiber glass, titanium, aluminum, and high-strength steel. In this chart, the relative weight values are calculated simply by taking the weight of the lightest vessel (S fiber glass) and dividing all other weight values by it. Thus, the following formula can be used to find the relative weight for a cylinder of aluminum (A) compared with S fiber glass (F):

$$\frac{W_A}{W_F} = \frac{2(\rho_A/\sigma_A)PV}{2(\rho_F/\sigma_F)PV} = \left(\frac{\rho_A}{\sigma_A}\right)\left(\frac{\sigma_F}{\rho_F}\right) = 2.9$$

Naturally, in a relative weight evaluation, several assumptions have to be made in establishing the appropriate strength values.

Establishing Strength Values

The strength values for fiber glass in figure 66 are based on total wall composite strength (adjusted to vessel axis) computed from room-temperature burst tests on 4-in. balanced cylinders, 8-in. spheroids, and 17-in. spheres. Strength values will be different for other vessels, depending on their sizes and proportions; however, scaling factors can be calculated.

The strength of virgin glass filaments is much higher than the strengths actually achieved when these filaments are used in pressure vessels. Nevertheless, all other considerations aside, fiber-glass vessels have a higher strength-to-density ratio than metals. Ultimate strength, rather than yield strength, is used here because some fabricators have the capability to provide the higher strength levels. However, care must be exercised in selecting the spread between yield and ultimate strengths of the pressure-vessel materials.

In the calculations for relative weight, allowance has not been made for the beneficial effect of biaxial reinforcement under load for homogeneous metals. Allowance for the biaxial reinforcement effect may

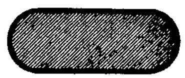
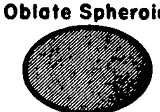
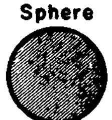
Vessel Shape \ Material	S Fiber glass (20 end-HTS finish epoxy-anhydride resin)	E Fiber glass (20 end-HTS finish epoxy-anhydride resin)	Titanium (6 Al-4V, forged & heat treated)	Aluminum (7075-T6)	High Strength Steel (Ladish $D6_AC$, heat treated)
Cylinder  $W = 2\left(\frac{\rho}{\sigma_{hoop}}\right) PV$	1.0 $\frac{\sigma_{hoop}}{\rho} = \frac{170{,}000}{0.073} = 2.33 \times 10^6$	1.4 $\frac{\sigma_{hoop}}{\rho} = \frac{125{,}000}{0.075} = 1.67 \times 10^6$	2.3 $\frac{\sigma}{\rho} = \frac{165{,}000}{0.16} = 1.03 \times 10^6$	2.9 $\frac{\sigma}{\rho} = \frac{78{,}000}{0.097} = 0.8 \times 10^6$	3.0 $\frac{\sigma}{\rho} = \frac{220{,}000}{0.283} = 0.78 \times 10^6$
Oblate Spheroid  $W = 3\left(\frac{\rho}{\sigma_{long.}}\right) PV$	1.0 $\frac{\sigma_{long.}}{\rho} = \frac{245{,}000}{0.073} = 3.36 \times 10^6$	1.4 $\frac{\sigma_{long.}}{\rho} = \frac{185{,}000}{0.075} = 2.47 \times 10^6$	—	—	—
Sphere  $W = 1.5\left(\frac{\rho}{\sigma}\right) PV$	1.5 $\frac{\sigma}{\rho} = \frac{85{,}000}{0.073} = 1.16 \times 10^6$	2.1 $\frac{\sigma}{\rho} = \frac{65{,}000}{0.075} = 0.87 \times 10^6$	1.7 $\frac{\sigma}{\rho} = \frac{165{,}000}{0.16} = 1.03 \times 10^6$	2.2 $\frac{\sigma}{\rho} = \frac{78{,}000}{0.097} = 0.8 \times 10^6$	2.2 $\frac{\sigma}{\rho} = \frac{220{,}000}{0.283} = 0.78 \times 10^6$

FIGURE 66.—Relative weights for unlined pressure vessels. (*Courtesy of* Materials in Design Engineering.)

provide as much as a 15-percent reduction in the relative weights that are indicated for homogeneous metal vessels.

The differences in weight of thermal insulations required for glass or metal vessels are not especially significant for some applications, such as rocket motor cases. Unlike a metal vessel, however, a glass vessel for gas storage requires a sealant liner that may vary in weight from 5 percent (for thick-walled vessels) to as much as 30 percent (for extremely thin-walled vessels) of the structural membrane weight.

Correction factors should be made in allowable design stresses of the various materials to compensate for the effect of extended internal pressure cycling or holding time. The degree to which these factors become significant depends on the specific application.

Allowance should also be made for all structural loads other than those imposed by internal pressure. In such cases, factors such as compressive strength, modulus of elasticity, and other material constants (as applied to buckling relationships) become important. These additional factors can change the comparisons which have been made here for conditions of simple internal pressure loading.

EXAMPLES OF COMPUTER-AIDED DESIGN OF STRUCTURES

Space-Frame Dome

The design requirements for a space-frame dome (ref. 53) include: a certain volume, specified (or range) height-to-radius ratio(s), load conditions, and environmental conditions. Common variations in dome surfaces are shown in figure 67 (from ref. 53).

By using computer-aided design techniques, a study of the effects of perimeter discontinuities along with wind-load distributions, irregular loads, thermal stresses, and support conditions can be performed simultaneously. These computer technques have been developed to analyze space frames by using the principle of minimum-strain energy. In addition to the geometry and wind-load inputs, structural parameters, such as cross-sectional areas of members, moments of inertia, and connection rigidities, are coded into the computer program so that they may be easily revised for design optimization. These programs permit summing of the total inter-strain energy of the structures, case by case, together with all the interaction products, so that required influence coefficients and structural deflections may be computed.

The design display may include such items as: a graphic profile of each configuration investigated, three-dimensional coordinate values of all space-frame joints and other significant positions, internal loads and stresses in all members, deflection of all significant joints, cost data, and construction time, to mention a few.

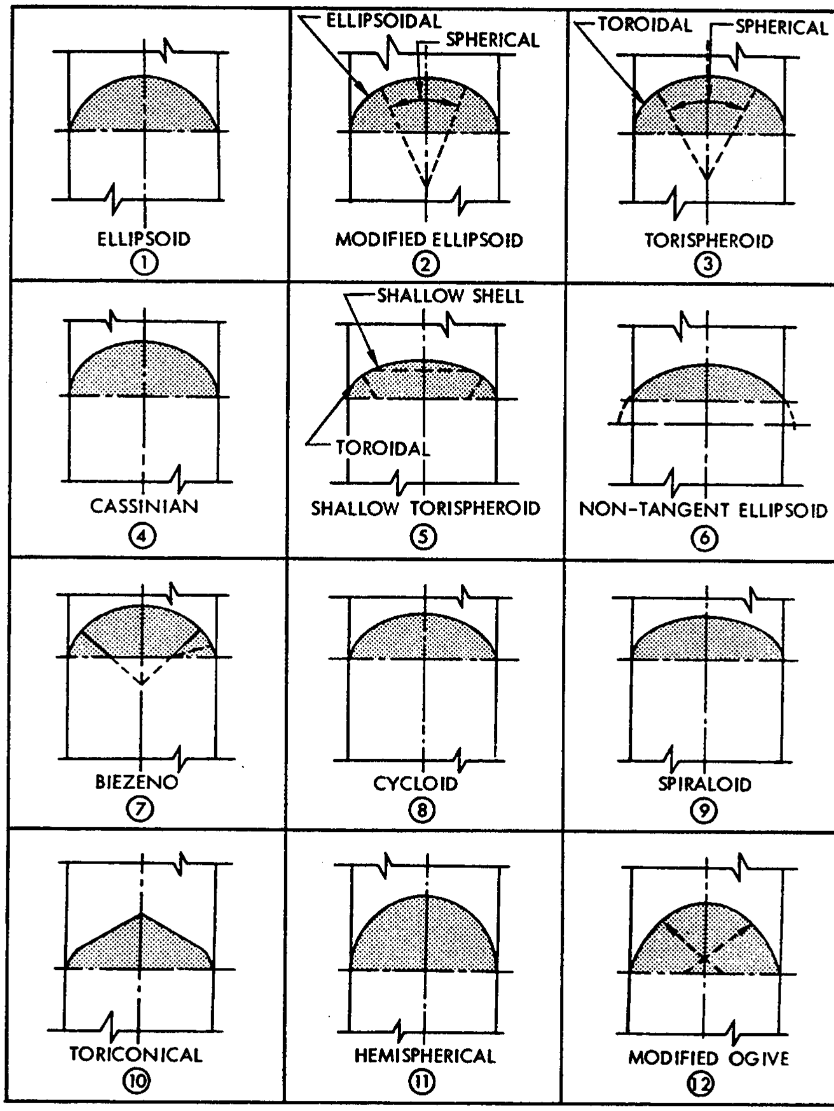


FIGURE 67.—General categories of surfaces of revolution analyzed by computer subroutines.

A typical geometrical configuration resulting from a computer-aided design study is shown in figure 68 (from ref. 53). In this study, the base radius and dome height were input data; load magnitudes and distributions were also specified. The merit function used in this study was defined as "cost-in-place," expressed in terms of enclosed volume. The configuration, called "Geolatic Framing," resulted

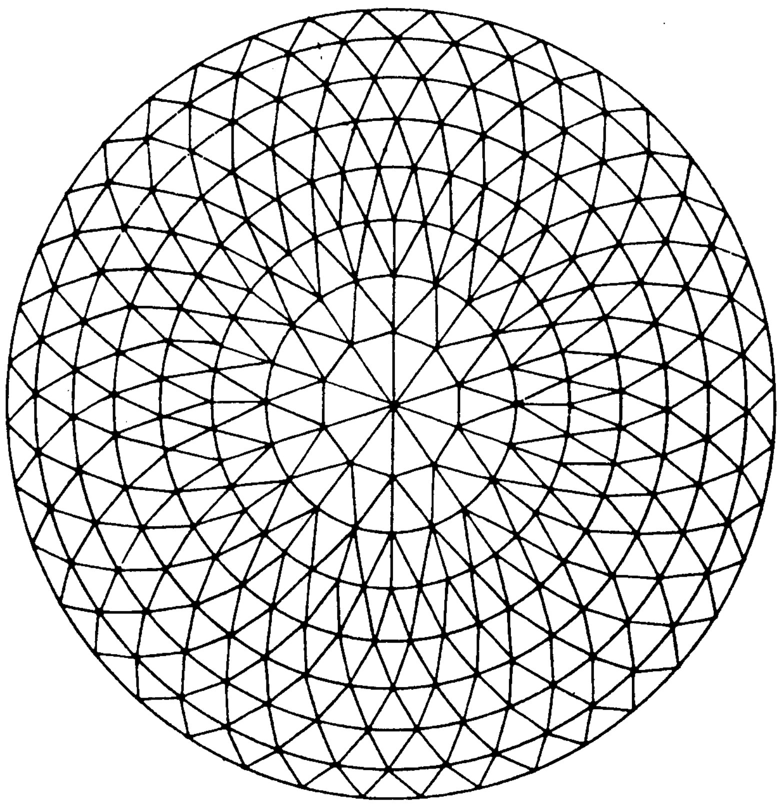


FIGURE 68.—Dome-framing plan developed by computer-aided design techniques.

from a study at North American Aviation, Inc., relating to the application of computer techniques to civil structures. "Geolatic" refers to framing systems in which a considerable number of framing members are in parallels of latitude. In the configuration shown, approximately 60 percent of the framing members were identical in fabrication.

Bridge Design

Another example of a computer-aided structural design is a bridge which spans a given distance and provides a specified traffic-flow capability. The performance requirements may include the number and width of traffic lanes; magnitude and distribution of imposed loads (static, dynamic, aerodynamic, seismic); and arterial-approach geometry. Constraint functions may include minimum clearance below the bridge, maximum allowable profile products, and applicable

Government specifications. Outlines of general bridge concepts are shown in figure 69 (from ref. 53).

The investigation of each proposed configuration should include principal variations in truss-framing geometry, pier locations, deck-framing concepts, approach ramp effects, material usage, maintenance

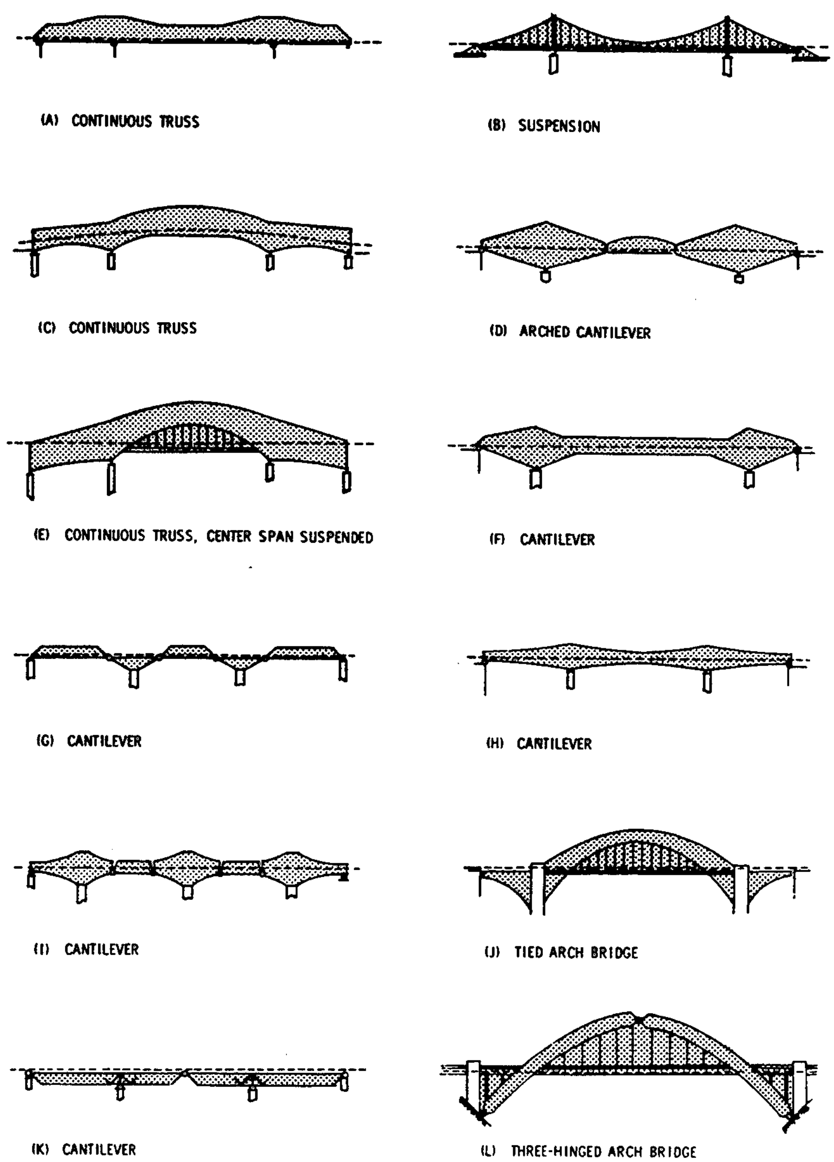


FIGURE 69.—Representative bridge configuration available for evaluation.

concepts, and related problems. An overall merit function can be expressed for each proposed design to indicate the total cost of ownership and operation per year in terms of vehicle-ton capacity. Output data may include: graphic profile of each configuration investigated, including cross sections and elevations; detailed geometry of components; rigidities of bridge sections; aerodynamic and dynamic response properties; cost data; and construction time, among others.

EXAMPLE OF STRUCTURAL OPTIMIZATION

The purpose of this analysis (ref. 64) is to establish a procedure for optimizing an integral stringer and ring-stiffened shell subjected to axial load. (See fig. 70 from ref. 64.) Two modes of failure are considered: strength based on the Von Mises yield criteria and elastic instability. The elastic instability consists of general instability (overall collapse of the cylinder), buckling of the unsupported panel lengths between rings, buckling of the skin bounded by the ring and stringers, and crippling of the outstanding stringer rib.

The optimization procedure is based upon elastic buckling with the following parameters being optimized: depth of rib, skin thickness, rib thickness, rib spacing, and ring spacing. The following assumptions are made:

(1) Internal pressure has no effect on the overall general instability.

(2) Ring spacing is sufficiently close so that the rings and skin are equally stressed.

(3) Curved panels are treated as flat plates, since the ribs are closely spaced.

(4) Critical buckling stresses are within the elastic limit.

To minimize the number of design parameters, two relationships are established:

(1) The depth of the ring is 2½ times that of the longitudinal stringer. This depth is arrived at by equating the local crippling stress of the outstanding leg of the longitudinal stringer with that of the ring web:

$$k_s \frac{E}{1-\mu^2}\left(\frac{t_w}{b_s}\right)^2 = k_r \frac{E}{1-\mu^2}\left(\frac{t_w}{K_1 b_w}\right)^2$$

where

$k_s=0.385$ (one edge free)

$k_r=3.29$ (both edges simply supported)

and K_1 equals 2.92; however, since one of the edge conditions of the web is actually elastically supported, use $K_1=2.5$. Therefore, depth of ring equals $2.5b_w$.

(2) By equating the local crippling stresses of the outstanding leg

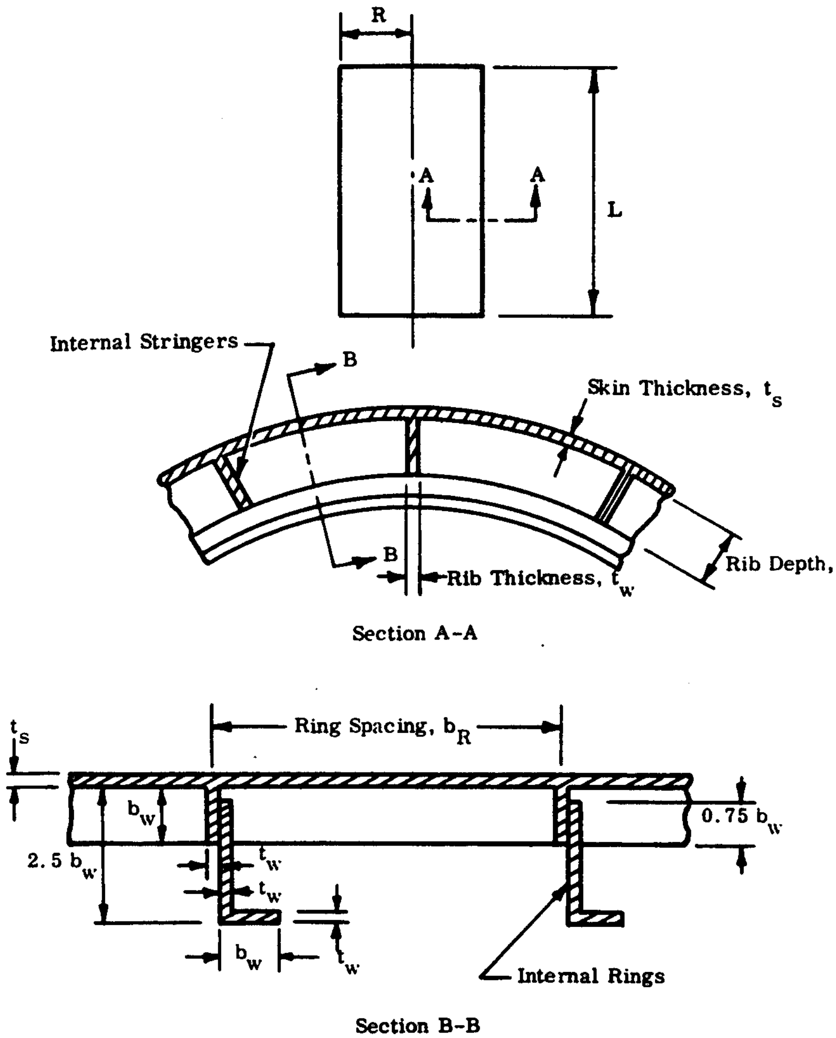


FIGURE 70.—Integral stringer and ring-stiffened cylinder geometry.

of the longitudinal stringer with that of the ring flange, we obtain a flange width equal to that of the stringer depth.

Failure Modes

Buckling Criteria

In predicting general instability, the equations developed by Block, Card, and Mikulas (ref. 65) are used. These equations represent the latest state of the art in buckling of orthotropic cylinders and take into consideration the effects of asymmetry; i.e., the effect of whether the rings and stringers are located on the inside or outside of the skin. The equations are:

$$N_x \frac{L^2}{\pi^2 D}=m^2(1+\beta^2)^2+m^2\frac{EI_s}{dD}+m^2\beta^4\frac{EI_r}{lD}+\left(\frac{G_sJ_s}{dD}+\frac{G_rJ_r}{lD}\right)m^2\beta^2$$

$$+\frac{12Z^2}{m^2\pi^4}\left(\frac{1+\overline{S}\Lambda_s+\overline{R}\Lambda_r+\overline{SR}\Lambda_{rs}}{\Lambda}\right)$$

where

$$\Lambda_r=1+2\alpha^2\beta^2(1-\beta^2\mu)\frac{\overline{Z}_r}{R}+\alpha^4\beta^4(1+\beta^2)^2\left(\frac{\overline{Z}_r}{R}\right)^2$$

$$\Lambda_s=1+2\alpha^2(\beta^2-\mu)\frac{\overline{Z}_s}{R}+\alpha^4(1+\beta^2)^2\left(\frac{\overline{Z}_s}{R}\right)^2$$

$$\Lambda_{rs}=1-\mu^2+2\alpha^2\beta^2(1-\mu^2)\left(\frac{\overline{Z}_r}{R}+\frac{\overline{Z}_s}{R}\right)+\alpha^4\beta^4[1-\mu^2+2\beta^2(1+\mu)]\left(\frac{\overline{Z}_r}{R}\right)^2$$

$$+2\alpha^4\beta^4(1+\mu)^2\frac{\overline{Z}_r\overline{Z}_s}{R^2}+\alpha^4\beta^2[2(1+\mu)+\beta^2(1-\mu^2)]\left(\frac{\overline{Z}_s}{R}\right)^2$$

$$\Lambda=(1+\beta^2)^2+2\beta^2(1+\mu)(\overline{R}+\overline{S})+(1-\mu^2)\cdot[\overline{S}+2\beta^2\overline{RS}(1+\mu)+\beta^4\overline{R}]$$

with

$$Z^2=\frac{L^4(1-\mu^2)}{R^2t^2}$$

$$\overline{S}=\frac{A_s}{td}$$

$$\alpha=\frac{m\pi R}{L}$$

$$D=\frac{Et^3}{12(1-\mu^2)}$$

$$\overline{R}=\frac{A_r}{tl}$$

$$\beta=\frac{nL}{m\pi R}$$

In order to use the previously defined equations, the basic stability equation must be minimized with respect to m and n to obtain the minimum allowable loading. Because of the complexity and time limitation involved, however, it is assumed that the ring and stringer eccentricities do not affect the buckling-mode shape. Based on this

273-140 O - 67 - 8

assumption, which has been subsequently proved to be valid, the equations for determining the buckling-mode shape for the Becker equation (ref. 64) are used. By utilizing the Becker equation to determine the buckling-mode shape and to nondimensionalize the design parameters, the following equations are obtained, letting

$$t_s = C_1 b_w$$

$$t_w = C_2 b_w$$

$$b_s = C_3 b_w$$

$$b_r = C_4 b_w$$

$$N_x = f(C_1, C_2, C_3, C_4)\,\frac{2ECb_w^2}{R}$$

where

$$f(C_1,C_2,C_3,C_4) = \frac{1}{2\phi^{1/2}}\left[\frac{(1+\beta^2)^2 C_1^3}{12(1-\mu^2)} + \frac{C_2}{12C_3} + \beta^4\,\frac{2.92C_2}{C_4} + 0.375\beta^2\left(\frac{C_2^3}{3C_3} + \frac{1.166C_2^3}{3C_4}\right)\right] + \frac{\phi^{1/2}}{2}\left(\frac{1+\bar{S}\Lambda_s+\bar{R}\Lambda_r+\overline{SR}\Lambda_{rs}}{\Lambda}\right)$$

$$\phi = (d_{11}+0.375\beta^2 d_{33}+\beta^4 d_{22})\left(\frac{1}{a_{22}}+\frac{0.375\beta^2}{a_{33}}+\frac{\beta^4}{a_{11}}\right)$$

$$\beta^2 = P + (P^2+Q)^{1/2}$$

$$P = \frac{a_{33}}{a_{22}}\left(\frac{a_{22}d_{11}-a_{11}d_{22}}{a_{11}d_{22}-2a_{33}d_{33}}\right)$$

$$Q = \frac{a_{11}}{a_{22}}\left(\frac{a_{22}d_{11}-2a_{33}d_{33}}{a_{11}d_{22}-2a_{33}d_{33}}\right)$$

If β^2 is negative, set $\beta^2=0$
If (P^2+Q) are negative, set $\beta^2=0$

$$\bar{S} = \frac{C_2}{C_3 C_1}$$

$$\bar{R} = \frac{4.25C_2}{C_4 C_1}$$

$$\Lambda_s = 1 + \frac{2(\beta^2-\mu)\bar{\zeta}_s\psi_s}{\phi^{1/2}} + \frac{(1+\beta^2)^2\bar{\zeta}_s^2\psi_s^2}{\phi}$$

$$\Lambda_r = 1 + \frac{2\beta^2(1-\beta^2\mu)\psi_r\bar{\zeta}_r}{\phi^{1/2}} + \frac{\beta^4(1+\beta^2)^2\bar{\zeta}_r^2}{\phi}$$

$$\Lambda_{rs}=1-\mu^2+\frac{2\beta^2(1-\mu^2)}{\phi^{1/2}}(\psi_r\bar{\zeta}_r+\psi_s\bar{\zeta}_s)+\frac{\beta^4[1-\mu^2+2\beta^2(1+\mu)]\bar{\zeta}_r^2\psi_r^2}{\phi}$$

$$+\frac{2\beta^4(1+\mu)^2\psi_r\bar{\zeta}_r\psi_s\bar{\zeta}_s}{\phi}+\frac{\beta^2[2(1+\mu)+\beta^2(1-\mu^2)]\bar{\zeta}_s^2\psi_s^2}{\phi}$$

$$\Lambda=(1+\beta^2)^2+2\beta^2(1+\mu)(\overline{R}+\overline{S})+(1-\mu^2)[\overline{S}+2\beta^2\overline{RS}(1+\mu)+\beta^4\overline{R}]$$

$$d_{11}=\frac{1}{C_3}\left[\frac{C_2}{3}+\frac{C_1C_2}{2}+\frac{C_1^2C_2}{4}-\frac{C_2^2(1+C_1)^2}{4(C_2+C_3C_1)}\right]$$

$$d_{22}=\frac{i_r+a_r(\bar{\zeta}_r-\overline{Y})^2+C_1C_4(\overline{Y})^2}{C_4}$$

$$a_r=4.25C_2$$

$$i_r=2.92C_2$$

$$\bar{\zeta}_r=1.44$$

$$\bar{\zeta}_s=\frac{1+C_1}{2}$$

$$\overline{Y}=\frac{6.11C_2}{4.25C_2+C_1C_4}$$

$$d_{33}=\frac{C_2^3}{16C_3}+\frac{0.219C_2^3}{C_4}+\frac{C_1^3}{16}$$

$$a_{11}=C_1+\frac{C_2}{C_3}$$

$$a_{22}=\frac{4.25C_2+C_1C_4}{C_4}$$

$$a_{33}=\frac{3C_1}{8}$$

$$D_{11}=d_{11}Eb_w^3$$

$$D_{22}=d_{22}Eb_w^3$$

$$D_{33}=d_{33}Eb_w^3$$

$$A_{11}=a_{11}Eb_w$$

$$A_{22}=a_{22}Eb_w$$

$$A_{33}=a_{33}Eb_w$$

C, the buckling correction factor, equals 0.58.

Because no test data are available for this type of construction, the same buckling correction factor will be used as for the single-face corrugation (ref. 64).

In predicting the buckling of the unsupported panel lengths between rings, the same equation used for general instability will be used, with, of course, the stiffness of the circumferential rings being taken as zero.

The equation is

$$\frac{N_x^p l^2}{\pi^2 D} = m^2(1+\beta^2)^2 + m^2\frac{EI_s}{dD} + m^2\beta^2\frac{GJ_s}{dD} + \frac{12Z^2}{m^2\pi^4}\left[\frac{1+\bar{S}\Lambda_3}{(1+\beta^2)^2+2\bar{S}\beta^2(1+\mu)+\bar{S}(1-\mu^2)}\right]$$

In predicting the theoretical panel buckling load, the above equation must be minimized with respect to m and n. In order to simplify the minimization, a value of 1 is used for m, the number of buckling half wavelengths between rings. This is analogous to the buckling-wave pattern of a simply supported Euler column between rings. In order to minimize with respect to n, a numerical iteration scheme is used to obtain the minimum value of N_x^p. (See fig. 71 from ref. 64.)

To do this, let

$$m=1$$
$$l=C_4 b_w$$
$$d=C_3 b_w$$

so that

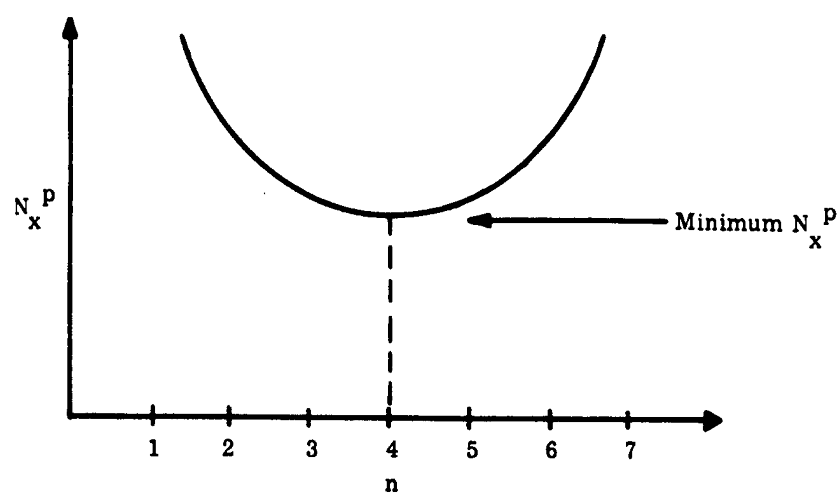


FIGURE 71.—Minimum value of $N^p{}_X$.

$$N_x^p=\left\{\frac{\pi^2D(1+\beta^2)^2}{C_4^2b_w^2}+\frac{\pi^2C_2Eb_w}{12C_4^2}+\beta_2\frac{\pi^2C_2^3Eb_w}{8C_4^2C_3}+\frac{12DZ^2}{C_4^2\pi^2b_w^2}\left[\frac{1+\overline{S}\Lambda_s}{(1+\beta^2)^2+2\overline{S}\beta^2(1+\mu)+\overline{S}(1-\mu^2)}\right]\right\}C_p$$

$$\beta=\frac{nC_4b_w}{\pi R}$$

$$\alpha=\frac{\pi R}{C_4b_w}$$

$$\overline{S}=\frac{C_2}{C_1C_3}$$

$$\overline{R}=\frac{4.25C_2}{C_4C_1}$$

$$D=\frac{C_1^3Eb_w^3}{12(1-\mu^2)}$$

$$Z^2=\frac{C_4^4(1-\mu^2)b_w^2}{C_1^2R^2}$$

$$\Lambda_s=1+\frac{\pi^2R}{C_4^2b_w}(\beta^2-\mu)(1+C_1)\psi_s+\frac{\pi^4R^2}{C_4^4b_w^2}(1+\beta^2)^2\frac{(1+C_1)^2}{4}\psi_s^2$$

C_p, the buckling correction factor, equals 0.58, which is the same factor used for general instability.

Assuming simply supported edge conditions and an aspect ratio of infinity, the critical rib-crippling stress (ref. 66) is

$$\sigma_{cr}=0.385\frac{E}{1-\mu^2}C_2^2$$

Assuming simply supported edge conditions and an aspect ratio of infinity, the critical skin buckling (ref. 66) is

$$\sigma_{cr}=3.29\frac{E}{1-\mu^2}\left(\frac{C_1}{C_3}\right)^2$$

Strength Criteria

In order to determine the maximum stress level in the skin, a modified form of the Von Mises yield equation is used. The skin is investigated, since its resultant stress will always be greater than, or equal to, that of the stiffening elements

$$\sigma=\sqrt{\left(\frac{N_x}{a_{11}b_w}\right)^2-\left(\frac{N_xN_y}{a_{11}a_{22}b_w^2}\right)+\left(\frac{N_y}{a_{22}b_w}\right)^2}$$

Optimization Procedure

Optimum design parameters C_1, C_2, C_3, C_4, and b_w must be determined to obtain a minimum-weight configuration. The approach to be taken is the concept of maximum strength-to-weight ratio. A logical range of C_1, C_2, C_3, and C_4 will be investigated and the corresponding strength-to-weight ratios calculated. The configuration with the maximum ratio will be investigated for panel buckling and for local forms of instability (skin buckling and rib crippling). If any of these forms of instability are violated, the values of C_1, C_2, C_3, and C_4 with the next highest strength-to-weight ratio are investigated. This process is continued until all forms of instability have been satisfied. Having determined the optimum values of C_1, C_2, C_3, and C_4, the value of the rib depth can be calculated to satisfy general instability using:

$$b_w=\sqrt{\frac{N_xR}{2CE[f(C_1, C_2, C_3, C_4)]}}$$

In determining the strength-to-weight ratios, the following equations are required:

$$\text{Average thickness, } t_{ave}=g(C_1, C_2, C_3, C_4)b_w$$

where

$$g(C_1, C_2, C_3, C_4)=C_1+\frac{C_2}{C_3}+4.25\,\frac{C_2}{C_4}$$

Substituting the value of b_w into the average thickness equation results in

$$t_{ave}=\frac{g(C_1, C_2, C_3, C_4)}{[f(C_1, C_2, C_3, C_4)]^{1/2}}\left(\frac{N_xR}{2CE}\right)^{1/2}$$

In order for the average thickness, and consequently the weight, to be a minimum, the following ratio must be maximum:

$$\frac{[f(C_1, C_2, C_3, C_4)]^{1/2}}{g(C_1, C_2, C_3, C_4)}\rightarrow\text{maximum}$$

The first step in determining a logical range of C_1, C_2, C_3, and C_4 is to investigate skin buckling, which is dependent on the ratio C_3/C_1. A plot of critical skin buckling versus C_3/C_1 was constructed and is shown in figure 72 (from ref. 64). Based on this plot, a range of C_3/C_1 from 20 to 120 is sufficient to cover a wide range of allowable stress levels. Using C_1 from 0.05 to 0.09 and C_3 from 2 to 6 will result in the desired range of C_3/C_1. Similarly, a plot of C_2-versus-critical rib-crippling stress is constructed to determine the range of C_2 to be

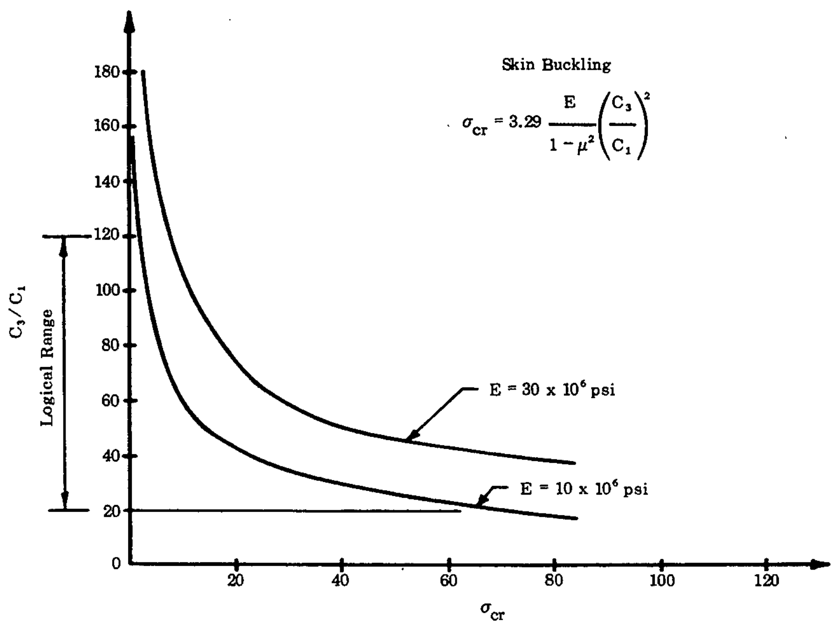


FIGURE 72.—Critical skin buckling versus C_3/C_1.

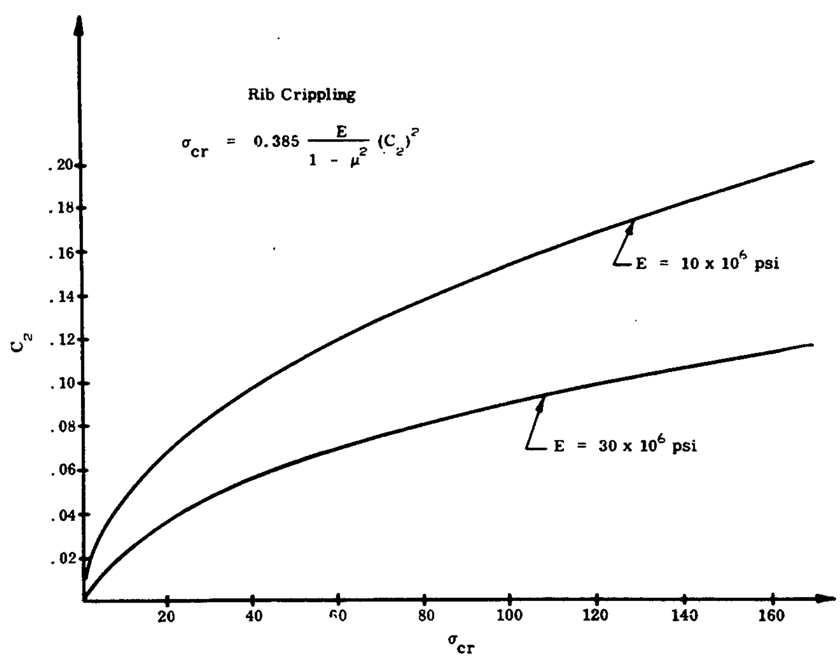


FIGURE 73.—C_2 versus critical rib crippling.

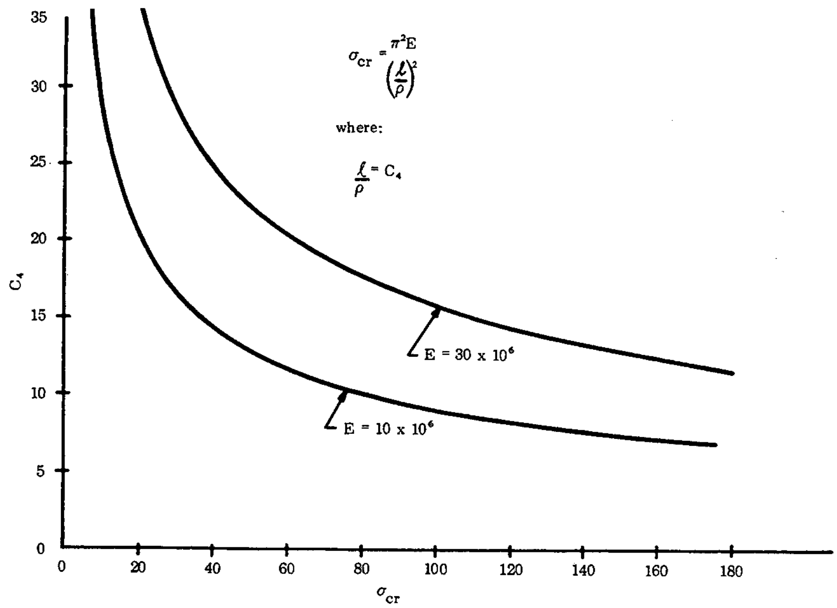
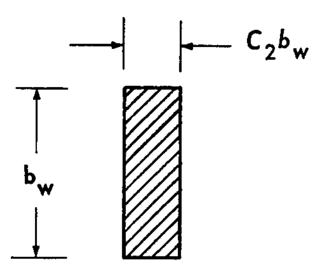


FIGURE 74.—Critical Euler stress levels versus C_4.

investigated (fig. 73 from ref. 64). Based on this plot, a reasonable range of C_2 is from 0.05 to 0.15.

Since C_4 is a measure of ring spacing, panel buckling must be investigated to determine the range of values. Due to the complexity of the panel-buckling equation, however, this form of instability will be simplified by considering the stringers as Euler columns simply supported between rings. Values of the critical Euler stress levels versus C_4 are plotted in figure 74 (from ref. 64). The value of $l/\rho = C_4$ was arrived at as follows:



Stringer cross section

$$\rho^2 = \frac{I}{A} = \frac{C_2 b_w^4}{12 C_2 b_w^2}$$

$$\rho = \frac{1}{\sqrt{12}} b_w$$

$$l = C_4 b_w \text{ (ring spacing)}$$

Therefore, $l/\rho = 3.42 C_4$. Upon investigating the curve, it was concluded that the logical range of C_4 was from 10 to 30.

Development of Weight Equation

In order to calculate the weight of the cylinder, the average smeared-out thickness, including the circumferential rings, is

$$t_{ave}=\left(C_1+\frac{C_2}{C_3}+4.25\,\frac{C_2}{C_4}\right) b_w$$

The weight per surface area equals $t_{ave}\rho F_b$, where F_b, the fabrication factor accounting for noncalculated items, equals 1.20.

SYMBOLS

- N_x axial load per inch, lb/in.
- N_y hoop load per inch, lb/in.
- R radius of cylinder, in.
- L length of cylinder, in.
- b_w depth of rectangular stringers, in.
- t_s skin thickness, in.
- t_{ws} thickness of rectangular stringers, in.
- b_s spacing of rectangular stringers, in.
- b_r spacing of circumferential rings, in.
- t thickness of cylinder shell wall, in.
- d stringer spacing, in.
- l ring spacing, in.
- J_r torsional constant for ring, $in.^4$
- J_s torsional constant for stringer, $in.^4$
- G shear modulus, psi
- E modulus of elasticity, psi
- μ Poisson's ratio
- A_s area of stringer, $in.^2$
- A_r area of ring, $in.^2$
- I_s moment of inertia of stringer, $in.^4$
- I_r moment of inertia of ring, $in.^4$
- $\bar{Z}_r$ distance from centroid of ring to middle surface of shell, positive if stiffener lies on external surface of shell, in.
- $\bar{Z}_s$ distance from centroid of stiffener to middle surface of shell, positive if ring lies on external surface of shell, in.
- ψ_s indicates whether stringers are external or internal to the skin surface; -1 if internal, $+1$ if external
- ψ_r indicates whether rings are external or internal to the skin surface; -1 if internal, $+1$ if external
- m number of half waves in cylinder buckle pattern in longitudinal direction
- n number of full waves in cylinder buckle pattern in circumferential direction

C	buckling correction factor
A_{11}	extensional stiffness in longitudinal direction, lb/in.
A_{22}	extensional stiffness in circumferential direction, lb/in.
A_{33}	shear stiffness, lb/in.
D_{11}	flexural stiffness in longitudinal direction, in.-lb
D_{22}	flexural stiffness in circumferential direction, in.-lb
D_{33}	torsional stiffness, in.-lb
σ	stress level, psi

References

1. Structural Systems and Program Decisions. Vol. 1, NASA SP-6008, 1966, pp. 1-10.
2. Childers, M. G.: Structures and Materials for Hypersonic Vehicles. Lockheed Horizons.
3. Hoff, N. J.: Thin Shells in Aerospace Structures (Fourth von Kármán Lecture). Presented at AIAA 3d Ann. Mtg. (Boston, Mass.), Nov. 29–Dec. 2, 1966. Astronaut. Aeron., Feb. 1967, pp. 26–43.
4. Korman, S.: Some New Metal and Metal-Ceramic Composites. NASA SP-5060, 1966.
5. Editors, Materials in Design Engineering: The Promise of Composites. M/DE Special Rept. no. 210, Sept. 1963.
6. Dietz, A. G. H.: Composite Materials. Presented at 68th ASTM Ann. Mtg. (West Lafayette, Ind.), June 16, 1965.
7. Sutton, W. H.; and Chorne, J.: Development of Heat-Resistant Alloys by Whisker Reinforcement. Metals Quart., vol. 3, no. 1, 1963, pp. 44–51.
8. Sutton, W. H.; et al.: Development of Composite Structural Materials for High Temperature Applications. Progress Reports 1–17, Space Science Lab., Missile and Space Vehicle Dept., General Electric Co., for U.S. Navy. Bureau of Weapons, Contract NOW-60-0465-d, 1960–64.
9. Sutton, W. H.: Whisker Composite Materials—A Prospectus for the Aerospace Designer. Astronaut. Aeron., Aug. 1966, pp. 46–51.
10. Garritano, R. A.: Techniques for Filament Winding Parts, Machine Design, Apr. 14, 1966, pp. 167-172.
11. Hauck, J. E.: Filament Winding. Mat. Design Eng., Apr. 1965, pp. 117-132.
12. Tsai, S. N.; Adams, D. F.; and Doner, D. R.: Analysis of Composite Structures. NASA CR-620, Nov. 1966.
13. Alexander, E. L.; Caputo, A. A.; Prado, M. E.; and Hilzinger, J. E.: Micromechanics of Fiber-Reinforced Composites. AFML-TR-65-283, Nov. 1965.
14. Milewski, J. V.: R and D Challenge of Whisker Composites. Research/Development, Mar. 1966, pp. 31-34.
15. Tsai, S. N.; and Assi, V. D.: The Strength of Laminated Anisotropic Composites. AIAA Preprint Paper no. 65–75, AIAA 2d Aerospace Sciences Mtg. (New York), Jan. 25–27, 1965.
16. Dong, S. B.; Matthiesen, R. B.; Pister, K. S.; and Taylor, R. T.: Analysis of Structural Laminates. ARL 76, Sept. 1961.
17. Plantema, F. J.: Sandwich Construction. John Wiley & Sons, Inc. (New York), 1966.
18. Anderson, R. A.: New Horizons in Structural Design. Proc. AIAA/ASME 7th Ann. Structures and Materials Conf. (Cocoa Beach, Fla.), Apr. 18–20, 1966, pp. 45–51.
19. Anderson, R. A.: Structures Technology—1964. Astronaut. Aeron., Dec. 1964, pp. 14–20.

20. Abraham, L. H.; and Lowy, M. J.: Shell Instability Problems as Related to Design. Collected Papers on Instability of Shell Structures, NASA TN D–1510, Dec. 1962, p. 8.
21. Bogdanoff, D.; et al.: Composite Structure for Orbiting Space Stations. SAE Preprint Paper 910D, National Aeronautic and Space Engineering and Manufacturing Mtg. (Los Angeles, Calif.), Oct. 1964.
22. Jackson, L. R.; et al.: Structural Concepts for Hydrogen-Fueled Hypersonic Airplanes. NASA TN D–3162, Feb. 1966, pp. 35–36.
23. Gerard, G.: Aerospace Pressure Vessel Design Synthesis. NASA CR–287, Aug. 1965.
24. Brewer, W. N.; and Jeppeson, N. L.: Methods of Evaluation of Inflatable Structures for Space Applications. Proc. AIAA/ASME 5th Ann. Structures and Materials Conf., Apr. 1964, pp. 344–360.
25. Hoffman, G. A.: Minimum-Weight Proportions of Pressure-Vessel Heads. J. Appl. Mech., vol. 29, no. 4, Dec. 1962, pp. 662–668.
26. Johns, R. H.; and Kaufman, A.: Filament-Overwrapped Metal Cylindrical Pressure Vessels. Proc. AIAA/ASME 7th Ann. Structures and Materials Conf. (Cocoa Beach, Fla.), Apr. 18–20, 1966, pp. 52–62.
27. Blumrich, J. F.: A Rising Tide of Structural Problems. Astronaut. Aeron., June 1965, pp. 54–57.
28. Blumrich, J. F.: Multicell Structures for Large Boosters. Space/Aeron., Jan. 1963, p. 86.
29. Blumrich, J. F.: The Semitoroidal Tank. Astronaut. Aeron., Feb. 1964, pp. 64–68.
30. Wuencher, H. F.; and Berge, L. H.: Multicell Tank (Development of Manufacturing Techniques for Large Space Vehicles of Multicell Configuration). In-House Prog. Rept., Marshall Space Flight Center.
31. Blum, R. E.: Analysis of Pressurized and Axially Loaded Orthotropic Multicell Tanks. NASA TN D–2799, May 1965.
32. Wilson, P. E.; Rogers, P. W.; Dharmarajan, S. N.; and Slick, E. M.: Pseudononlinear Bending Analysis of an Orthotropic Multicell Shell Structure. AIAA J., vol. 4, no. 12, Dec. 1966, pp. 2096–2104.
33. Levenetz, B.: Narmco Research and Development Rept. W.O. no. 090–031, Dec. 1961.
34. Lewis, D. J.; Modisette, J. L.; and Thibodaux, J. G.: High Mass Fraction Solid-Propellant Rocket Design Concept. Bull. 18th Mtg.—JANAF–ARPA–NASA Solid Propellant Group, vol. 1, Applied Physics Lab., Johns Hopkins Univ., June 1962, pp. 237–260.
35. Zeckel, J.: Isotensoid Pressure Vessels. ARS J., vol. 32, no. 6, June 1962, p. 950.
36. Anderson, M. S.; Robinson, J. C.; Buch, H. G.; and Fralich, R. W.: A Tension-Shell Structure for Application to Entry Vehicles. NASA TN D–2675, Mar. 1965.
37. Halberg, W.: Optimization of Tension Shell Voyager Base-Line Design. General Electric PIR–SM–8156–000–1659, Sept. 1, 1965.
38. Levy, S.; and Hess, T. E.: Tension Shell Design Configuration for Planetary Entry. Proc. AIAA/ASME 7th Ann. Structures and Materials Conf. (Cocoa Beach, Fla.), Apr. 18–20, 1966, pp. 439–446.
39. Alai, J. A.: The Tension String Structure. Proc. AIAA/ASME 7th Ann. Structures and Materials Conf. (Cocoa Beach, Fla.), Apr. 18–20, 1966, pp. 447–458.
40. Boler, L. J.; and Brown, C. R.: Lightweight Packageable Structural Materials for the Exploration of Space. Proc. AIAA/ASME 7th Ann.

Structures and Materials Conf. (Cocoa Beach, Fla.), Apr. 18–20, 1966, p. 324.
41. Forbes, F. W.: Expandable Structures. Space/Aeron., Dec. 1964, pp. 62–68.
42. Brink, N. O.; Anderson, B. C.; Thompson, C. E.; and Wolcott, C. E.: Development and Evaluation of the Elastic Recovery Concept for Expandable Space Structures. NASA CR-121, Dec. 1964.
43. Schuerch, H. U.; and Schindler, G. M.: Analysis of Foldability in Expandable Structures. AIAA J., vol. 1, no. 4, Apr. 1963, pp. 873–878.
44. Osgood, C. C.: Spacecraft Structures. Prentice-Hall, Inc., 1966, pp. 232–233.
45. Linton, R.: Thermal Design Evaluation of Pegasus. NASA TN D-3642, Dec. 1966.
46. Otto, F.: Stresses and Membranes. World Conference on Shell Structures (San Francisco, Calif.), 1962, NAS-NRC Publ. no. 1187, pp. 3-20.
47. Salvadori, M.; and Heller, R.: Structure in Architecture. Prentice-Hall, Inc., 1965, pp. 257–279.
48. Mark, H. F.; and Editors of Life: Giant Molecules, Life Science Library, Time, Inc. (New York), 1966, p. 175.
49. R. Buckminster Fuller—Dean of the Dome. Look Mag., Aug. 23, 1966, pp. 56–58.
50. Yarsley, V. E.; and Flavell, W.: Reinforced Plastics in the United Kingdom and Continental Europe in the Year 2000. Proc., 19th Ann. Mtg. SPI (Chicago, Ill.), 1964, Section 16C.
51. Baxter J., Jr.: Filament Winding Future? Good! Proc., 20th Ann. Mtg. SPI (Chicago, Ill.), 1965.
52. Asimow, M.: Introduction to Design. Prentice-Hall, Inc., 1962.
53. Harris, L. A.; Mitchell, J. C.; and Morgan, G. W.: Computer-Aided Design for Civil Engineering Structure. Symp. on Technology Status and Trends (Huntsville, Ala.), Apr. 21–23, 1965, NASA SP-5030, 1966, pp. 147–160.
54. Gerard, G.: Structural Interplay: Design and Materials. Aero/Space Eng., Aug. 1959, pp. 37–42.
55. Gerard, G.: Materials Evaluation and Design. Astronaut. and Aeron., Mar. 1966, pp. 58–62.
56. Razani, R.: Behavior of Fully Stressed Design of Structures and Its Relationship to Minimum-Weight Design. AIAA J., vol. 3, no. 12, Dec. 1965, pp. 2262–2268.
57. Schmidt, L. C.: Fully-Stressed Design of Elastic Redundant Trusses Under Alternative Load Systems. Australian J. Appl. Sci., vol. 9, Dec. 1958, pp. 337–348.
58. Schmit, L. A.: Structural Design by Systematic Synthesis. Proc. 2d Nat. Conf. on Electronic Computations, Am. Soc. Civil Engrs. (New York), 1960, pp. 105–132.
59. Kuhn, H. W.; and Tucker, A. W.: Non-linear Programming. Proc. 2d Berkeley Symp. on Mathemetical Statistics and Probability, Univ. of Calif. Press (Berkeley), 1951, pp. 481–492.
60. Lewallen, J. M.; and Tapley, B. D.: Analysis and Comparison of Several Numerical Optimization Methods. AIAA Preprint Paper no. 67–58, 5th Aerospace Sciences Mtg. (New York), Jan. 23–26, 1967.
61. Kiciman, M. O.: A Random Sampling Procedure With Application to Structural Synthesis Problems. AIAA Preprint Paper no. 66–531, 4th Aerospace Sciences Mtg. (Los Angeles, Calif.), June 1966.
62. Kahn, H.: Applications of Monte Carlo. Rand Corp. Memorandum RM-1237, AEC, 1954.

63. Editors, Materials in Design Engineering: Selecting Materials for Minimum Weight. Mat. Design Eng., Mar. 1964.
64. Anon.: Structural Systems and Program Decisions. Vol. 1, NASA SP-6008, 1966, N-1—N-17.
65. Block, D. L.; Card, M. F.; and Mikulas, M. M., Jr.: Buckling of Eccentrically Stiffened Orthotropic Cylinders. NASA TN D-2960, Aug. 1965.
66. Roark, R. J.: Formulas for Stress-Strain. McGraw-Hill Book Co., Inc. (New York), 1954.

Bibliography

STIFFENED-SKIN STRUCTURES

1956

BENNEY, E. A.; ZUCROW, M. J.; AND BESSERER, C. W.: Aerodynamics, Propulsion, Structures. D. Van Nostrand Co., Inc., 1956.

1957

Advances in the Astronautical Sciences, vols. 1–6. Am. Astronaut. Soc., 1957–61.

BIJLAARD, P. P.: Buckling Under External Pressure of Cylindrical Shells Evenly Stiffened by Rings Only. J. Aeron. Sci., vol. 24, no. 6, June 1957, p. 437.

BODNER, S. R.: General Instability of a Ring-Stiffened, Circular Cylinder Shell Under Hydrostatic Pressure. J. Appl. Mech. (Trans. ASME), vol. 79, 1957, pp. 269–277.

GERARD, G.; AND BECKER, H.: Handbook of Structural Stability.
- Pt. I, Buckling of Flat Plates. NACA TN 1781, July 1957.
- Pt. II, Buckling of Component Elements. NACA TN 3782, July 1957.
- Pt. III, Buckling of Curved Plates and Shells. NACA TN 3783, Aug. 1957.
- Pt. IV, Failure of Plates and Component Elements. NACA TN 3784, Aug. 1957.
- Pt. V, Compressive Strength of Flat Unstiffened Panels. NACA TN 3785, Aug. 1957.
- Pt. VI, Strength of Stiffened Curved Plates and Shells. NACA TN 3786, July 1957.

SEIDE, P.: A Donnell Type Theory for Asymmetrical Bending and Buckling of Thin Conical Shells. J. Appl. Mech., vol. 24, no. 4, Dec. 1957, p. 547.

1958

ADAMS, C. C.: Space Flight. McGraw-Hill Book Co., Inc., 1958.

BECKER, H.: General Instability of Stiffened Cylinders. NACA TN 4237, July 1958.

BESSERER, C. W.: Missile Engineering Handbook. D. Van Nostrand Co., Inc., 1958.

GALLETLY, G. D.; SLANKARD, R. C.; AND WENK, E.: General Instability of Ring-Stiffened Cylindrical Shells Subject to External Hydrostatic Pressure—Comparison of Theory and Experiment. J. Appl. Mech., vol. 25, no. 2, June 1958, p. 259.

KOLLBRUNNER, C. F.; AND MEISTER, M.: Ausbeulen, Theorie und Berechnung von Blechen. Springer-Verlag (Berlin), 1958, p. 325.

RADKOWSKI, P. P.: Buckling of Thin Single- and Multi-Layer Conical and

Cylindrical Shells With Rotationally Symmetric Stresses. Proc. 3d U.S. Natl. Cong. Appl. Mech., ASME, 1958, p. 443.

SUJATA, H. L.: Plastic Buckling of Simply Supported Plates With Longitudinal Stiffeners Subjected to Uniform Compression. Ph. D. thesis, Cornell Univ., Sept. 1958.

TEREBUSHKO, O. I.: Analysis of Carrying Capacity of Circular Cylindrical Panel Reinforced by Ribs. *In:* Collect. Analysis of Spatial Structures, Gosstroiizdat (Moscow), no. 4, 1958, pp. 531–554.

1959

ARKHANGORODSKIY, A. G.; AND POPOV, V. G.: Stability of Cylindrical Shells With Oblique Transverse Stiffeners. Izv. Vysshckh Uchebn. Zavedenii Mashinostr., no. 6, 1959, pp. 61–68.

GERARD, G.; AND BECKER, H.: Handbook of Structural Stability. Pt. VII—Strength of Thin-Wing Construction. NASA TN D–162, Sept. 1959.

HOMEWOOD, R. H.; BRINE, A. C.; AND JOHNSON, A. E.: Buckling Instability of Monocoque Shells. Avco TR RAD–TR–9–59–20, Aug. 1959.

PETERSON, J. P.; AND DOW, M. B.: Compression Tests on Circular Cylinders Stiffened Longitudinally by Closely Spaced Z-Stringers. NASA Memo 2–12–59L, 1959.

REYNOLDS, T. E.; AND BLUMENBERG, W. F.: General Instability of Ring-Stiffened Cylindrical Shells Subject to External Hydrostatic Pressure. DTMB Rept. 1324, June 1959.

SANDERS, J. L., JR.: An Improved First-Approximation Theory for Thin Shells. NASA TR R–24, 1959.

SEIDE, P.: On the Buckling of Truncated Conical Shells Under Uniform Hydrostatic Pressure. Proc. IUTAM Symp. of the Theory of Thin Elastic Shells (Delft, Holland), Aug. 1959, 1960, p. 363.

TURKIN, K. D.: General Stability of a Reinforced Cylindrical Shell Under Transverse Bending. *In:* Collect. Analysis of Spatial Structures, Gosstroiizdat (Moscow), no. 5, 1959, pp. 450–474.

1960

ARKHANGORODSKIY, A. G.; AND POPOV, V. G.: Stability of Cylindrical Shells, Reinforced by Transverse and Longitudinal Stiffeners. Izv. Vysshikh Uchebn. Zavedenii Mashinostr., no. 3, 1960, pp. 3–13.

DAREVSKIY, V. M.; AND KSHNIAKIN, R. I.: Stability Under External Pressure of a Cylindrical Shell Reinforced by Rings. Dokl. Akad. Nauk SSSR, vol. 134, no. 3, 1960, pp. 548–551.

HOFF, N. J.; AND SINGER, J.: Buckling of Circular Conical Shells Under External Pressure. Proc. IUTAM Symp. on the Theory of Thin Elastic Shells (Delft, Holland), Aug. 1959, p. 389.

SINGER, J.: The Effect of Axial Constraint on the Instability of Thin Circular Cylindrical Shells Under External Pressure. J. Appl. Mech., vol. 27, no. 4, Dec. 1960, p. 737.

STEIN, M.: Behavior of Buckled Rectangular Plates. J. Eng. Mech. Div., vol. 86, EM2, 1960, pp. 59–76.

THIELEMANN, W. F.: New Developments in the Nonlinear Theories of the Buckling of Thin Cylindrical Shells. *In:* Aeronautics and Astronautics, Pergamon Press, Inc., 1960, pp. 76–119.

TRAPEZIN, I. I.: Critical Load and Natural Frequency of an Orthotropically

Constructed Conical Shell, Clamped at the Apex, Under Uniform Hydrostatic Pressure. *In:* Strength of Aeronautical Structures, p. 5. Inst. of Aviation—Sergo Orgonikidze, Oborongiz, Moscow, 1960. (In Russian.)

1961

BECKER, H.; AND GERARD, G.: Measurement of Torsional Rigidity of Stiffened Plates. NASA TR SM 61–14, Oct. 1961.

CZERWENKA, G.: Untersuchungen von Dünnen Kurzen Zylindern, die Durch Ringkleinstprofile Enger und Mittlerer Teilung Verstarkt Sind und Unter Manteldruck Stehen. Z. Flugwissensch., vol. 10, no. 6, June 1961, pp. 163–202.

EAKIN, F.: Honeycomb Cylinder Tests. Douglas Aircraft 8M–37719, 1961.

GERARD, G.: Cylinder Instability. NASA Tech. Status Rept. no. 5, May 15, 1961–Nov. 15, 1961.

GERARD, G.: Plastic Stability Theory of Geometrically Orthotropic Plates and Cylindrical Shells. New York Univ. TR SM 61–11, July 1961.

HESS, T. E.; AND VINSON, J. R.: Behavior of Orthotropic Structures. Doc. no. 61SD46, General Electric Co., Mar. 1961.

SERPICO, J. C.: Elastic Stability of Orthotropic Conical and Cylindrical Shells Subjected to Axisymmetric Loading Conditions. Avco RAD TR 61–7, Apr. 1961.

SINGER, J.: Buckling of Circular Conical Shells Under Axisymmetric External Pressure. J. Mech. Eng. Sci., vol. 3, no. 4, Dec. 1961, p. 330.

TENNYSON, R. C.: An Experimental Investigation of the Behavior of Stiffened Plates in Axial Compression. UTIA TN 57, Sept. 1961.

TIMOSHENKO, S.; AND GERE, J.: Theory of Elastic Stability, 2d ed., McGraw-Hill Book Co., Inc., 1961.

1962

BECKER, H.; AND GERARD, G.: Elastic Stability of Orthotropic Shells. J. Aerospace Sci., vol. 29, no. 5, May 1962, pp. 505–512, 520.

BECKER, H.; GERARD, G.; AND WINTER, R. W.: Experiments on Axially Compressive General Instability of Monolithic Circumferentially Stiffened Circular Cylindrical Shells. New York Univ. TR SM 62–5, May 1962.

CRAWFORD, R. F.; AND BURNS, A. B.: Strength Efficiency and Design Data for Beryllium Structures. Aeronautical Systems Div., ASD TR 61–692, WPAFB, 1962.

CRAWFORD, R. F.; AND HOLMES, A. M. C.: RIFT Interstage Structural Development Program. Structures Study 1017, Lockheed Missiles & Space Co., June 1962.

FLÜGGE, W.: Stresses in Shells. Springer-Verlag (Berlin), 1962.

GERARD, G.: Introduction to Structural Stability. McGraw-Hill Book Co., Inc., 1962.

GERARD, G.: Compressive Stability of Orthotropic Cylinders. New York Univ. TR SM 62–4, May 1962.

GERARD, G.: On the Role of Initial Imperfections in Plastic Buckling of Cylinders Under Axial Compression. J. Aerospace Sci., vol. 29, no. 6, June 1962, pp. 744–745.

GRESZCZUK, L. B.: Effect of Reinforcement Geometry on the Stresses in Spherical Shells. Paper 578C, Soc. Automotive Engrs., Natl. Aerospace Eng. & Mfg. Mtg. (Los Angeles, Calif.), Oct. 8–12, 1962.

HEDGEPETH, J. M.: Design of Stiffened Cylinders in Axial Compression. *In:* Collected Papers on Instability of Shell Structures—1962. NASA TN D–1510, Dec. 1962, pp. 77–83.

273-140 O - 67 - 9

HESS, T. E.: Stability of Orthotropic Cylindrical Shells Under Combined Loading. J. Am. Rocket Soc., vol. 31, no. 2, Feb. 1961, p. 237.

KORDASHENKO, A. B.: On Stability of Shells Reinforced by Ribs. Izv. Akad. Nauk SSSR, Otd. Tekhn. Nauk, Mekhan. i Mashinostr., no. 1, 1962, pp. 115–120.

MAZURKIEWICZ, Z.: Bending and Buckling of Rectangular Plate Reinforced Transversely by Ribs With Variable Rigidities. Bull. Acad. Polon. Sci. Ser. Sci. Tech., vol. 10, no. 8, 1962, pp. 329–339.

NEUT, A. VAN DER: General Instability of Orthogonally Stiffened Circular Shells. *In:* Collected Papers on Instability of Shell Structures—1962. NASA TN D–1510, Dec. 1962, pp. 309–321.

NICKELL, F. H.; AND CRAWFORD, R. F.: Optimum Ring Stiffened Cylinders Subjected to Uniform Hydrostatic Pressure. Soc. Automotive Engrs., preprint 578F, 1962.

PIERCE, C. M.: Post-Buckling Behavior of Stiffened Cylinders and Curved Panels: An Annotated Bibliography (Period Covered, 1950—June 1962). SB–62–37, Rept. 3–77–62–22, Sept. 1962.

REYNOLDS, T. E.: Elastic Lobar Buckling of Ring-Supported Cylindrical Shells Under Hydrostatic Pressure. U.S. Navy DTMB Rept. 1614, Sept. 1962.

SEIDE, P.; AND WEINGARTEN, V. I.: On the Buckling of Circular Cylindrical Shells Under Pure Bending. Trans. ASME, vol. 28, 1961, pp. 112–116.

SINGER, J.: Buckling of Orthotropic and Stiffened Conical Shells. Collected Papers on Instability of Shell Structures. NASA TN D–1510, Dec. 1962, pp. 463–479.

SINGER, J.; AND ECKSTEIN, A.: Experimental Investigations of the Instability of Conical Shells Under External Pressure. Proc. 4th Israel Ann. Conf. Aviation Astronaut., Feb. 1962. Bull. Res. Council Israel, vol. 11C, no. 1, Apr. 1962, p. 97.

SINGER, J.; AND FERSHT-SCHER, R.: Buckling of Orthotropic Conical Shells Under External Pressure. TAE Rept. 22, Technion Res. and Develop. Found., Haifa, Israel (to be published in the Aeronaut. Quart.), 1962. (See 1964.)

1963

BAILEY, F. C.: Buckling and Failure Strength of Plating Panels Loaded in Shear. Final Rept., Contract NObs–72290 (TR–602/51(d)), Sept. 20, 1963.

BARUCH, M.: General Instability of Stiffened Circular Conical Shells Under Hydrostatic Pressure. Unpubl. Ph.D. thesis, Technion-Israel Inst. Technol. (Haifa), Jan. 1963. (In Hebrew.)

BARUCH, M.; AND SINGER, J.: Effect of Eccentricity of Stiffeners on the General Instability of Stiffened Cylindrical Shells Under Hydrostatic Pressure. J. Mech. Eng. Sci., vol. 5, no. 1, 1963, pp. 23–27.

BARUCH, M.; AND SINGER, J.: General Instability of Stiffened Circular Conical Shells Under Hydrostatic Pressure. Technion Res. and Develop. Found. (Haifa, Israel), TAE Rept. 28, May 1963.

BECKER, H.; AND GERARD, G.: Measurement of Torsional Rigidity of Stiffened Plates. NASA TN D–2007, July 1963.

BLUHM, J. I.; AND MARDIROSIAN, M. M.: Fracture-Arrest Capabilities of Annularly Reinforced Cylindrical Pressure Vessels. Exptl. Mech., vol. 3, Dec. 1963, pp. 305–306.

BUDIANSKY, B.; AND RADKOWKI, P. P.: Numerical Analysis of Unsymmetrical Bending of Shells of Revolution. AIAA J., vol. 1, no. 1, 1963, pp. 1833–1842.

CRAWFORD, R. F.; AND BURNS, A. B.: Minimum Weight Potentials for Stiffened Plates and Shells. AIAA J., vol. 1, no. 4, Apr. 1963, pp. 879–886.

HOM, K.; BUHL, J. E., JR.; AND COUCH, W. P.: Hydrostatic Pressure Tests of Unstiffened and Ring-Stiffened Cylindrical Shells Fabricated of Glass-Filament Reinforced Plastics. DTMB-1745, Sept. 1963.

MINKARAH, I. A.; AND HOPPMANN, W. H.: Flexural Vibrations of Cylindrically Aeolotropic Circular Plates. Contract DA-30-069-AMC-195(R), Rensselaer Polytech. Inst., Oct. 1963.

SERPICO, J. C.: Elastic Stability of Orthotropic Conical and Cylindrical Shells Limited to Axisymmetric Loading Conditions. AIAA J., vol. 1, no. 1, Jan. 1963, pp. 128–137.

SINGER, J.: A Donnell Type Equation for Bending and Buckling of Orthotropic Conical Shells. J. Appl. Mech., vol. 30, no. 2, 1963, pp. 303–305. (Also TAE Rept. 18, Technion Res. and Develop. Found. (Haifa, Israel), 1961.)

SINGER, J.; AND ECKSTEIN, A.: Recent Experimental Studies of Buckling of Conical Shells Under Torsion and External Pressure. Proc. 5th Israel Ann. Conf. Aviation Astronaut., Feb. 1963, p. 135.

SINGER, J.; ECKSTEIN, A.; FERSHT-SCHER, R.; AND BERKOVITS, A.: Buckling of Isotropic, Orthotropic and Ring-Stiffened Conical Shells. Technion Res. and Develop. Found. (Haifa, Israel), TAE Rept. 30, sec. 1, Sept. 1963.

ŠKALOUD, MIROSLAV: The Supercritical Behavior of Compressed Plates Reinforced by Elastic Stiffeners (Überkritisches Verhaletn Gedrückter, Mit Nachgiebigen Rippen Versteifter Platten). Acta Tech. ČSAV, vol. 8, no. 5, 1963, pp. 459–474.

STACHIW, J. D.: The Effects of Shell Joints and Bonding on the Stability of Acrylic Resin Cellular Shells. Pennsylvania State Univ., Contract NORd-16597-97, Sept. 25, 1963.

TENNYSON, R. C.: A Note on the Classical Buckling Load of Circular Cylindrical Shells Under Axial Compression. AIAA J., vol. 1, 1963, pp. 473–476.

TURKIN, K. D.: Stability of a Reinforced Circular Cylindrical Shell Subjected to Compression and Pure Bending (Ustoychivost Podkreplennoy Krugovoy Tsilindricheskoy Obolochki Pri Szhatii I Chistom Izgibe). NASA TT F-8298, July 1963.

1964

ALMROTH, B. O.: Postbuckling Behavior of Orthotropic Cylinders Under Axial Compression. AIAA J., vol. 2, no. 10, 1964, pp. 1795–1799.

AMBARTSUMYIAN, S. A.: Theory of Anisotropic Shells. NASA TT F-118, May 1964.

ANON.: Theoretical and Experimental Analysis of Orthotropic-Shell Stability. LMSC-A701014, Lockheed Missiles & Space Co., Sept. 11, 1964.

BARUCH, M.: Equilibrium and Stability Equations for Stiffened Shells. Proc. 6th Israel Ann. Conf. Aviation Astronaut. Israel J. Technol., vol. 2, no. 1, Feb. 1964, p. 117.

BARUCH, M.; ET AL.: General Instability of Conical Shells With Non-Uniformly Spaced Stiffeners Under Hydrostatic Pressure. Sci. Rept. no. 1, TAE-37, Dept. Aeronaut. Eng., Israel Inst. Technol., Dec. 1964.

BURNS, A. B.: Structural Optimization of Axially Compressed Cylinders Stiffened With Corrugation and Rings, Including the Effects of Corrugation/Ring Eccentricity. LMSC 6-65-65-12, Lockheed Missiles & Space Co., 1964.

CARD, M. F.: Bending Tests of Large-Diameter Stiffened Cylinders Susceptible to General Instability. NASA TN D-2200, 1964.

CARD, M. F.: Preliminary Results of Compression Tests on Cylinders With Eccentric Longitudinal Stiffeners. NASA TM X-1004, 1964.

DURGAR'IAN, S. M.: A Nonsteady Thermal Problem for a Cylindrical Shell Reinforced by a Ring (Ob Odnoi Nestatsionarnoi Temperaturnoi Zadache Dlia Tsilindricheskoi Obolochki, Podkreplennoi Kol'tsom). Theory of Shells and Plates: Proc. 4th All-Union Conf. (Yerevan, Armenian SSR), Oct. 24–31, 1962. Yerevan, Izdatel'stvo Akad. Nauk Armianskoi SSR, 1964, pp. 452–460.

GALKIN, S. I.: Calculation of a Cylindrical Shell Reinforced by Elastic Ribs, Under the Effect of a Concentrated Force (Raschet Tsilindricheskoi Obolochki, Podkreplennoi Uprugimi Shpangoutami, Na Deistvie Sosredotochennoi Sily). Theory of Shells and Plates: Proc. 4th All-Union Conf. (Yerevan, Armenian SSR), Oct. 24–31, 1962. Yerevan, Izdatel'stvo Akad. Nauk Armianskoi SSR, 1964, pp. 362–372. (In Russian.)

HOFF, N. J.: The Effect of the Edge Conditions on the Buckling of Thin-Walled Circular Cylindrical Shells in Axial Compression. Stanford Univ., Dept. of Aerospace Eng. Res. SUDAR 205, U.S. Air Force Office Sci. Res., AFOSR 64–1724, Aug. 1964.

KHUAN, K.-C.: Limiting Equilibrium of Cylindrical Reinforced Shells With Thin Claddings (O Predel'nom Ravnovesii Tsilindricheskikh Podkreplennykh Obolochek S. Tonkimi Obshivkami). Acta Mech. Sinica, vol. 7, June 1964, pp. 91–108. (In Russian.)

KIZIMA, G. A.: Investigation of the Stressed State of Ribbed Shells of Zero Gaussian Curvature (Issledovanie Napriazhennago Sostoianiia Rebristykh Obolochek Nulevoi Guassovoi Krivizny). Theory of Shells and Plates: Proc. 4th All-Union Conf. (Yerevan, Armenian SSR), Oct. 24–31, 1962. Yerevan, Izdatel'stvo Akad. Nauk Armianskoi SSR, 1964, pp. 507–514. (In Russian.)

KOMISSAROVA, G. L.: Stability of a Longitudinally Corrugated Cylindrical Shell With and Without Reinforcing Ribs (Ustoichivost'Prodol'no Gofrirovannoi Tsilindricheskoi Obolochki, Neprodkreplennoi I Podkreplennoi Shpangoutami). Theory of Shells and Plates: Proc. 4th All-Union Conf. (Yerevan, Armenian SSR), Oct. 24–31, 1962. Yerevan, Izdatel'stvo Akad. Nauk Armianskoi SSR, 1964, pp. 563–571. (In Russian.)

MAEDA, T.; AND NISHIMURA, T.: Shear Buckling of Simply-Supported Infinitely Long Plates Reinforced by Oblique Stiffeners. Nagoya Univ., Fac. Eng., Mem., vol. 16, Nov. 1964, pp. 110–118.

MECK, H. R.: A Survey of Methods of Stability Analysis of Ring-Stiffened Cylinders Under Hydrostatic Pressure. ASME Winter Ann. Mtg. (New York), Nov. 29–Dec. 4, 1964. Paper 64–WA/UnT–2. Trans. ASME, Ser. B–J. of Eng. for Industry, vol. 87, Aug. 1965, pp. 385–390.

MEINCKE, H.: Reinforcement of Cylindrical Vessels Subject to External Pressure (Die Versteifung von Zylindrischen Aussendruckbehältern). VDI Z., vol. 106, Jan. 1964, pp. 19–22. (In German.)

MEYER, R. R.; AND BALLIFANTE, R. J.: Fabrication and Experimental Evaluation of Common Domes Having Waffle-Like Stiffening. Pt. I, Rept. no. SM–47742, Douglas Aircraft Corp., Nov. 1964.

PAPIRNO, R.; AND BECKER, H.: Stresses in Reinforced, Perforated Rectangular Plates Loaded by Transverse Pressure. Final Rept., ARA–F–239–4: Allied Research Associates, Inc., Dec. 1964.

ROCKEY, K. C.; AND COOK, I. T.: Influence of the Torsional Rigidity of Transverse Stiffeners Upon the Shear Buckling of Stiffened Plates. Aeronaut. Quart., vol. 15, May 1964, pp. 198–202.

SCHMIT, L. A., JR.; AND FOX, R. L.: An Integrated Approach to Structural Synthesis and Analysis. Proc. AIAA/ASME 5th Ann. Structures and Materials Conf., AIAA (New York), 1964, pp. 294–315.

SINGER, J.; AND FERSHT-SCHER, R.: Buckling of Orthotropic Conical Shells Under External Pressure. Aeronaut. Quart., vol. XV, May 1964, p. 151.

SKALOND, M.; AND NOVOTNY, R.: Postbuckling Behavior of a Uniformly Compressed Plate Stiffened by Two Longitudinal Ribs Subdividing the Web Into Three Equal Panels. Acta Tech. CSAV, vol. 9, no. 6, 1964, pp. 542–566. (In German.)

TENNYSON, R. C.: Buckling of Circular Cylindrical Shells in Axial Compression. AIAA J., vol. 2, 1964, pp. 1351–1353.

WAH, T.: Circular Symmetric Vibrations of Ring-Stiffened Cylindrical Shells. J. Soc. Ind. Appl. Math., vol. 12, Sept. 1964, pp. 649–662.

1965

ARMSTRONG, H. H.: Weight Saving Through Use of Beryllium-Aluminum Alloys for Saturn-Type Vehicle Structures. LMSC/HREC A712431, Lockheed Missiles & Space Co., Huntsville, Ala., Dec. 1965.

BARUCH, M.; AND SINGER, J.: General Instability of Stiffened Circular Conical Shells Under Hydrostatic Pressure. Aeronaut. Quart., vol. 16, pt. 2, May 1965, pp. 187–204.

BECKER, H.; AND GERARD, G.: Elastic Stability of Orthotropic Shells. J. Aerospace Sci., vol. 29, no. 5, May 1965, pp. 501–512.

BIRON, A.; AND SAWCZUK, A.: Plastic Analysis of Rib-Reinforced Cylindrical Shells. DOMIIT–1–29, IIT Res. Inst., Dept. Mech., July 1965.

CRAWFORD, R. F.: Effects of Asymmetric Stiffening on Buckling of Shells. Paper 65–371. AIAA 2d Ann. Mtg. (San Francisco, Calif.), July 26–29, 1965.

CRAWFORD, R. F.; AND SCHWARTZ, D. B.: General Instability and Optimum Design of Grid-Stiffened Spherical Domes. AIAA J., vol. 3, no. 3, Mar. 1965, pp. 511–515.

DELUZIO, A. J.; STUHLMAN, C. E.; AND ALMROTH, B. O.: Influence of Stiffener Eccentricity and End Moment on the Stability of Cylinders in Compression. Proc. AIAA/ASME 6th Ann. Structures and Materials Conf., Apr. 1965.

DOOLEY, J. F.: On the Torsional Stiffness of Closed-Section Web Stiffeners. Intern. J. Mech. Sci., vol. 7, 1965, pp. 183–196.

DOW, D. A.: Buckling and Post-Buckling Tests of Ring-Stiffened Cylinders Loaded by Uniform External Pressure. NASA TN D–3111, Nov. 1965.

FENN, R. W., JR.; ET AL.: A Mechanical Property Evaluation of Be-38% Al Alloy From −320° to 800° F. Metal Eng. Quart., vol. 5, no. 4, Nov. 1965, p. 37.

FENN, R. W., JR.; ET AL.: Beryllium-Aluminum Alloys. J. Spacecraft Rockets, vol. 2, no. 1, Jan.–Feb. 1965, p. 87.

FRASER, W. B.: Buckling of a Structure With Random Imperfections. Ph. D. thesis, Harvard Univ., 1965.

GUTKOWSKI, W.: Cylindrical Grid Shells. Bull. Acad. Polon. Sci., Ser. Sci. Tech., vol. 13, no. 3, 1965, pp. 191–199.

HEALY, J. J.: Parametric Study of Unstiffened and Stiffened Prolate Spheroidal Shells Under External Hydrostatic Pressure. DTMB–2018, Struc. Mech. Lab., David Taylor Model Basin (Washington, D.C.), Aug. 1965.

HEDGEPETH, J. M.; AND HALL, D. B.: Stability of Stiffened Cylinders. 2d AIAA Aerospace Sci. Mtg. (New York), Jan. 25–27, 1965, Paper 65–79. AIAA J., vol. 3, Dec. 1965, pp. 2275–2286.

HUFFINGTON, N. J., JR.; AND SCHUMACHER, R. N.: Flexure of Parallel-Stiffened Plates. Martin Co. (Baltimore, Md.), Jan. 1965.

IL'YIN, L. O.: Differential Equations of Elastic Equilibrium of Shells of Rotation With Meridional Ribs at Power and Temperature Loads. Prikl. Mekhan. (Ukr. SSR), vol. 10, no. 3, Dec. 15, 1965, pp. 254–262.

JOHNSTON, G. S.; AND LANTZ, R. B.: Efficiency of Waffle Grid Panels Under Compressive Loading. J. Aircraft, vol. 2, Sept.–Oct. 1965, pp. 437–438.

KOWDO, J.: Ultimate Strength of Longitudinally Stiffened Plate Panels Subjected to Combined Axial and Lateral Loading. Rept. 248.13 Lehigh Univ. (Bethlehem, Pa.), Aug. 1965.

McELMAN, J. A.; MIKULAS, M. M., JR.; AND STEIN, M.: Static and Dynamic Effects on Eccentric Stiffening of Plates and Cylindrical Shells. AIAA Preprint Paper 65–370, AIAA 2d Annual Mtg. (San Francisco, Calif.), July 26–29, 1965.

MANEVICH, A. E.: Stability Under External Load of a Cylindrical Shell Reinforced by Ribs (Ustoichivost' Tsilindricheskoi Obolochki Podkreplennoi Shpangoutami I Nagruzhennoi Vneshnim Davleniem). Akad. Nauk SSSR, Izv., Mekhan., Nov.–Dec. 1965, pp. 106–110. (In Russian.)

MEAD, D. J.: The Damping of Stiffened Plate Structures. Acoustical Fatigue in Aerospace Structures: Proc. 2d Intern. Conf. (Dayton, Ohio), Apr. 29–May 1, 1964.

MORDELET, R.: New Considerations on Elastic Stability (Considerations Nouvelles Sur la Stabilite Elastique). Assoc. Francaise des Ingénieurs et Techniciens de l'Aéronautique et de l'Espace, Congrès Intern. Aéronautique, 7th, Problèmes de Structures d'Avions et d'Engins (Paris, France), June 14–16, 1965. (In French.)

NICKELL, E. H.; AND SKOGH, J.: An Investigation of the Panel Collapse Strength of Stiffened Plastic Hemispheres. LMSC–6–75–65–53, Lockheed Missiles & Space Co., Aerospace Sci. Lab., Oct. 1, 1965.

SINGER, J.; ET AL.: Further Remarks on the Effect of Eccentricity of Stiffeners on the General Instability of Stiffened Cylindrical Shells. Sci. Rept. no. 2, Israel Inst. of Tech., Dept. of Aeronaut. Eng., TAE–42: SR–2, Aug. 1965.

TSUIJI, T.: Built-Up Members in Plastic Design: Strength of Longitudinally Stiffened Plate Panels With Large b/t. Contract NObs–9004, Lehigh Univ., Fritz Eng. Lab. Rept. no. 248.14, June 1965.

WAH, T.: The Buckling of Gridworks. J. Mech. Phys. Solids, vol. 13, Feb. 1965, pp. 1–16.

WOOD, W. G.: The Collapse Under End Load of Pressurized, Axially Stiffened, Thin Cylinders. J. Mech. Eng. Sci., vol. 7, Dec. 1965, pp. 469–481.

ZARUTSKIY, V. A.; AND PRIADKO, A. A.: Analysis of Reinforced Shells. Prikl. Mekhan. i Tekhn. Fiz. (USSR), vol. 1, no. 7, 1965, pp. 50–56.

1966 (Partial Listing)

BURNS, A. B.; AND ALMROTH, B. O.: Structural Optimization of Axially Compressed, Ring-Stringer Stiffened Cylinders. AIAA J. Spacecraft Rockets, vol. 3, 1966, pp. 19–25.

CHILDERS M. G.: Structures and Materials for Hypersonic Vehicles. In Lockheed Horizons.

DICKSON, J. N.; AND BROLLIAR, R. H.: The General Instability of Ring-Stiffened Corrugated Cylinders Under Axial Compression. NASA TN D–3089, Jan. 1966.

Structural Systems and Program Decision, vol. 1, NASA SP–6008, 1966.

LAYERED STRUCTURES OTHER THAN SANDWICH

1939

GREEN, A. E.; AND TAYLOR, G. I.: Stress Systems in Aeolotropic Plates, I. Proc. Royal Soc. (London), vol. 173A, 1939, pp. 162–172.

GREEN, A. E.: Stress Systems in Aeolotropic Plates, II. Proc. Royal Soc. (London), vol. 173A, 1939, pp. 173–192.

1942

GREEN, A. E.: Stress Systems in Aeolotropic Plates, IV. Proc. Royal Soc. (London), vol. 180A, 1942, pp. 173–208.

1944

GOLAND, M.; AND REISSNER, E.: The Stresses in Cemented Joints. J. Appl. Mech., vol. 11, 1944, pp. 17–27.

1945

GREEN, A. E.; AND TAYLOR, G. I.: Stress Systems in Aeolotropic Plates, III. Proc. Royal Soc. (London), vol. 184A, 1945, pp. 181–195.

GREEN, A. E.: Stress Systems in Aeolotropic Plates, V. Proc. Royal Soc. (London), vol. 184A, 1945, pp. 231–252.

GREEN, A. E.: Stress Systems in Aeolotropic Plates, VI. Proc. Royal Soc. (London), vol. 184A, 1945, pp. 289–300.

GREEN, A. E.: Stress Systems in Aeolotropic Plates, VII. Proc. Royal Soc. (London), vol. 184A, 1945, pp. 301–345.

1947

BIRCHALL, H.; AND LAKE, G. F.: An Alternative Form of Pressure Vessel of Novel Laminar Construction. Proc. Inst. Mech. Eng., vol. 156, 1947, pp. 349–367.

1949

DIETZ, A. G. H., ED.: Engineering Laminates. John Wiley & Sons, Inc., 1949.

DIETZ, A. G. H.: Two Species Laminated Beams. Trans. ASME, vol. 71, May 1949, pp. 401–405.

1951

KEYLWERTH, R.: The Anisotropic Elasticity of Wood and of Laminated Wood. VDI-Forschungsh., vol. 430, 40 pp., 1951. (In German.)

1952

HEARMON, R. F. S.; AND ADAMS, E. H.: The Bending and Twisting of Anisotropic Plates. Brit. J. Appl. Phys., vol. 3, no. 5, May 1952, pp. 150–156.

1953

SMITH, C. B.: Some New Types of Orthotropic Plates Laminated of Orthotropic Materials. J. Appl. Mech., vol. 20, June 1953, pp. 268–271.

STRUB, R. A.: Distribution of Mechanical and Thermal Stresses in Multilayer Cylinders. Trans. ASME, vol. 75, no. 1, Jan. 1953, pp. 73–79.

WITT, R. K.; HOPPMAN, W. H., II; AND BUXBAUM, R. S.: Determination of Elastic Constants of Orthotropic Materials with Special Reference to Laminates. ASTM Bull., vol. 194, Dec. 1953, p. 53.

1954

CLARK, L. G.: Deflection of Laminated Beams. Trans. Am. Soc. Civil Engrs., vol. 119, 1954, p. 721.

GREEN, A. E.; AND ZERNA, W.: Theoretical Elasticity. Oxford University Press (London), 1954.

TRENKS, K.: Contributions to the Calculation of Orthogonal Anisotropic Rectangular Plates. Bauingenieur, vol. 29, no. 10, 1954, pp. 372–377. (In German.)

1955

Clark, L. G.: Buckling of Laminated Columns. J. Appl. Mech., vol. 22, 1955, pp. 553–556.

Durelli, A. J.; and Tsao, C. H.: J. Appl. Mech. (Trans. ASME), vol. 22, 1955, p. 190.

Erickson, E. C. O.; and Norris, C. B.: Tensile Properties of Glass-Fabric Laminates With Laminations Oriented in Any Way. Forest Prod. Lab. Rept. no. 1853, Nov. 1955.

1956

Anon.: An Appraisal of the Properties and Methods of Production of Laminated or Composite Ship Steel Plate. Comm. on Ship Steel, Ship Structure Comm. Rept. No. 84, Jan. 1956.

Jasper, T. M.; and McLean, J. T.: Multilayer Vessels for High Pressure and Temperature. Chem. Eng. Progr., vol. 52, no. 12, Dec. 1956, pp. 521–526.

1957

Marin, J.: Theories of Strength for Combined Stresses and Nonisotropic Materials. J. Aeron. Sci., vol. 24, no. 4, 1957, pp. 265–269.

1958

Micks, W. R.: Bibliography of Literature on Optimum Design of Structures and Related Topics. The Rand Corp., RM-2304, ASTIA AD-215771, Dec. 15, 1958.

Radkowski, P. P.: Buckling of Thin Single and Multilayer Conical and Cylindrical Shells With Rotationally Symmetric Stresses. Proc. 3d U.S. Natl. Congr. Appl. Mech. (Providence, R.I.), June 1958, pp. 443–449.

Suchar, M.: Computation by Means of Polynomials of Influence Surfaces for Anisotropic Plates With Finite Dimensions. Arch. Mech. Stosowanej, vol. 10, no. 5, 1958, pp. 615–634. (In English.)

Vinson, J. R.: Thermal Stresses in Laminated Circular Plates. Proc. 3d U.S. Natl. Congr. Appl. Mech., June 1958, ASME, 1958, pp. 471–476.

Zophres, W.: Rept. EM 8-27, Space Technology Labs. (Redondo Beach, Calif.), 1958.

1959

Baird, R. B.; Forbes, F. W.; and Lipsitt, H. A.: Tensile and Fatigue Properties of Laminated Sheet Structures. ASTM Proc., vol. 59, 1959, pp. 755–766.

Becker, S. J.; and Mollick, L.: The Theory of the Ideal Design of a Compound Vessel. J. Eng. Ind., ASME Paper 59-A-125, 1959.

Clark, L. G.: General Small Deflection Theory of Elastic Bending and Buckling of Laminated Plates. Proc. 4th Midwestern Conf. Solid Mech., Univ. of Texas, 1959.

Composite Materials and Composite Structures. Proc. 6th Sagamore Ordnance Mater. Res. Conf., Aug. 1959.

Gottenberg, W. G.: TR-59-0000-09951, Space Technology Labs. (Redondo Beach, Calif.), 1959.

Librescu, L.: Studii Cercetari Mecan., Appl., Inst. Mecan. Appl., Acad. Repub. Pop. Romine, vol. 10, no. 1, 1959, p. 187.

Librescu, L.: Studii Cercetari Mecan., Appl., Inst. Mecan. Appl., Acad. Repub. Pop. Romine, vol. 10, no. 4, 1959, p. 1235.

Miller, R. E.; et al.: TAM Rept. no. 143, Univ. Illinois (Urbana, Ill.), 1959.

PISTER, K. S.: Flexural Vibration of Thin Laminated Plates. J. Acoust. Soc. Am., Feb. 1959.

PISTER, K. S.; AND DONG, S. B.: Elastic Bending of Layered Plates. J. Eng. Mech. Div., ASCE, vol. 84, Oct. 1959, pp. 1–10.

SIMON, A. W.: Theory of Beams Composed of Two Elastic Materials. Am. J. Phys., vol. 27, no. 7, Oct. 1959, pp. 500–502.

STAVSKY, Y.: On the Theory of Heterogeneous Anisotropic Plates. Progress Rept., AF Contract No. AF 33(616)–6280, MIT, 1959–60.

WAHL, N. E.: Properties of Asbestos Reinforced Laminates at Elevated Temperatures. SAE Preprint Paper 106V, SAE Natl. Aeron. Mtg. (Los Angeles, Calif.), Oct. 5–9, 1959.

1960

GOODMAN, J. W.: Final Report on Pressure Vessel Design Criteria. Space Technology Labs., Inc., Rept. AFBMD–TR–61–9, Dec. 31, 1960.

KOBLER, R. S.; LURIE, W.; AND NUTT, R. L.: Development and Evaluation of an Adhesive System for Structural Bonding of Aluminum. Naval Ordnance Rept. 6953 (pt. I), Apr. 1960.

LIBRESCU, L.: Rev. Mecan. Appl., vol. 5, no. 3, 1960, p. 401.

LIBRESCU, L.: Studii Cercetari Mecan., Appl., Inst. Mecan. Appl., Acad. Repub. Pop. Romine, vol. II, no. 4, 1960, p. 937.

LIBRESCU, L.: Rev. Mecan. Appl., vol. 5, no. 5, 1960, p. 695.

NASH, W. A.: Appl. Mech. Rev., vol. 13, 1960, p. 161.

MCKAIG, H. L.; AND JONES, J. B.: Development Study for Optimization of Structural Properties of Pressure Vessels with Ultrasonic Welding. (U), Aeroprojects, Inc., Rept. RR–60–94 (Confidential), July 1960.

VALENTA, J.: Acta Tech., vol. 31, no. 1/2, 1960, p. 227.

1961

BECKER, S. J.: An Analysis of the Yielded Compound Cylinder. J. Eng. Ind., Feb. 1961, pp. 43–49.

COWPER, G. R.: Struct. Lab., Aeron. Rept. LR–300, Canada NRC, NAE, 1961.

DONG, S. B.; ET AL.: Analysis of Structural Laminates. Aerospace Res. Lab., Office Aerospace Res., USAF, WPAFB (Dayton, Ohio), 1961.

HEARMON, R. F. S.: An Introduction to Applied Anisotropic Elasticity. Oxford University Press (London), 1961.

PALMER, P. J.: Band Reinforced and Layer Built Pressure Vessels. British Welding J., Feb. 1961, pp. 51–57.

RADKOWSKI, P. P.: AVCO RAD TR–61–36, 1961.

WHITE, J. C.: J. Eng. Ind. (Trans. ASME) vol. 83, ser. B, 1961, p. 397.

1962

ABRAHAM, L. H.: Structural Design of Missiles and Spacecraft. McGraw-Hill Book Co., Inc., 1962, p. 108.

AMBARTSUMIAN, S. A.: Appl. Mech. Rev., vol. 15, p. 245, 1962.

BAGDASARIAN, ZH. E.; AND GNUNI, W. Ts.: FTO–TT–61–313, FTD, AFSC, WPAFB (Dayton, Ohio), 1962, ASTIA AD–270785.

BALTRUKONIS, J. H.; ET AL.: TR–3, NsG–125–61, Catholic Univ. Am. (Washington, D.C.), 1962.

BIJLAARD, P. P.; AND DOHRMANN, R. J.: General Procedure for Analyzing Thick Shells of Irregular Shapes for Mechanical and Thermal Loading. ASME Paper 62–WA–226, 1962.

DONG, S. B.; ET AL.: J. Aerospace Sci., vol. 29, p. 969, 1962.

LANE, F.; AND MAGNUS, D.: GASL–TR–284, General Applied Science Labs. (Westbury, N.Y.), 1962.

PAUL, B.: Linear Bending Theory of Laminated Cylindrical Shells Under Axisymmetric Load. ASME Paper 62–WA–83, 1962.

SAITO, H.; AND SATO, K.: J. Appl. Mech. (Trans. ASME), vol. 29, p. 287, 1962.

1963

ANON.: Multilayer Vessels. Chicago Bridge & Iron Co., 1963.

BERT, C. W.: Ellipsoidal Closures for Minimum-Weight Pressure Vessels. J. Can. Aeron. Space Inst., May 1963, pp. 133–136.

BUDIANSKY, B.; AND RADKOWSKI, P. P.: AIAA J., vol. 1, p. 1833, 1963.

CITRIN, G.: Fourth Quarterly Progress Report—Evaluation of High-Strength Lightweight Laminated Pressure Vessels of Lap-Joint Construction. Republic Aviation Corp., RAC 1160, 244–3004; Watertown Arsenal Rept. no. WAL TR 766.2/3–3, Jan. 16, 1963.

CITRIN, G.: Fifth Quarterly Progress Report—Evaluation of High-Strength Lightweight Laminated Pressure Vessels of Lap-Joint Construction. Republic Aviation Corp., RAC 1313, 244–3005; Watertown Arsenal Rept. no. WAL TR 677.2/3–4, Apr. 10, 1963.

CLARK, S. K.: Textile Res. J., vol. 33, p. 295, 1963.

CLARK, S. K.: Textile Res. J., vol. 33, p. 935, 1963.

DONG, R. G.; AND DONG, S. B.: AIAA J., vol. 1, p. 2565, 1963.

CHENG, S.; AND HO, B. P. C.: AIAA J., vol. 1, p. 892, 1963.

CHENG, S.; AND HO, B. P. C.: AIAA J., vol. 1, p. 1603, 1963.

ERINGEN, A. C.: Proc. 1st Midwestern Conf. Solid Mech., Univ. Illinois (Urbana, Ill.), p. 66, 1963.

HAYASHI, T.; AND HIRANO, Y.: Trans. Japan Soc. Aeron. Space Sci. vol. 6, no. 9, p. 18, 1963.

LOVE, G. G.: AIAA J., vol. 1, p. 1843, 1963.

RUBO, E.: New Vistas Opened Up by Welded Multi-Wall Pressure Vessels Having One Wall Isolated From the Next. Welding Research Abroad, Dec. 1963.

1964

AMBARTSUMIAN, S. A.: Theory of Anisotropic Shells. NASA TT F–118, 1964.

CHENG, S.: MRC–483, Math. Res. Center, Univ. Wisconsin (Madison, Wis.), 1964. ASTIA AD–603288.

DONG, S. B.: J. Eng. Mech. Div. (Proc. Am. Soc. Civil Engr. EM), vol. 90, pt. 1, p. 53, 1964.

IYENGAR, K. T. S. R.; AND YOGANANDA, C. V.: Z. Angew, Math. Mech., vol. 44, p. 270, 1964.

PAUL, B.: J. Appl. Mech. (Trans. ASME), vol. 30, p. 98, 1963, and vol. 31, p. 155, 1964.

STAVSKY, Y.: Aeron. Quart., vol. 15, p. 29, 1964.

WEINGARTEN, V. I.: J. Soc. Exptl. Stress Anal., vol. 4, p. 200, 1964.

1965

TSAI, S. N.; AND AZZI, V. D.: The Strength of Laminated Anisotropic Composites. AIAA Preprint Paper 65–75, AIAA 2d Aerospace Sci. Mtg., New York, Jan. 25–27, 1965.

LAYERED STRUCTURES OTHER THAN SANDWICH—RUSSIAN WORKS

1949

Ambartsumian, S. A.: Dokl. Akad. Nauk. Arm. SSR 11, no. 2, 1949.

1950

Fridman, M. M.: The Mathematical Theory of Elasticity of Anisotropic Media. Grikladn. Mat. Mekhan., vol. 14, 1950, pp. 321–340. (In Russian.)

Lekhnitskiy, S. G.: Theory of Elasticity of Anisotropic Bodies. *In:* Gosudarstvennoe Izdatel'stvo, Tekhniko-Theoretichskoi Literaturi (Moscow-Leningrad), 1950. (In Russian.)

1951

Ambartsumian, S. A.: Izv. Akad. Nauk Arm. SSR, Fiz. Mat., Estestv. i Tekhn. Nauki, vol. 4, no. 5, 1951.

1952

Ambartsumian, S. A.: Izv. Akad. Nauk Arm. SSR, Fiz. Mat., Estestv. i Tekhn. Nauki, vol. 5, no. 6, 1952.

1953

Ambartsumian, S. A.: The Calculation of Laminated Anisotropic Shells. Izv. Akad. Nauk Arm. SSR, Ser. Fiz. Mat. Nauk, vol. 6, no. 3, 1953, pp. 15–35. (In Russian.)

Grigolyuk, E. I.: Strength and Stability of Cylindrical Bimetallic Shells. Inzh. Sb., vol. 16, 1953, pp. 119–148. (In Russian.)

Grigolyuk, E. I.: Thin Bimetallic Shells and Plates. Inzh. Sb., vol. 17, 1953, pp. 69–120. (In Russian.)

Uzdalev, A. I.: Bending Stresses in Anisotropic Two-Layered Cylinders by Transverse Forces. Inzh. Sb., vol. 15, 1953, pp. 35–42. (In Russian.)

1954

Grigolyuk, E. I.: Izv. Akad. Nauk SSSR, Otd., Tekhn. Nauk, vol. 8, p. 89, 1954.

Grigolyuk, E. I.: Equations of Axially Symmetric Bimetallic Elastic Shells. Inzh. Sb., vol. 18, 1954, pp. 89–98. (In Russian.)

Grigolyuk, E. I.: Inzh. Sb., vol. 19, p. 73, 1954 (trans. D. Taylor).

1955

Grigolyuk, E. I.: Gos. izd-vo lit. po str-vu i arkhit., no. 3, p. 375, 1955.

Korolev, V. I.: Inzh. Sb., vol. 22, p. 98, 1955 (trans. Avco Mfg. Co., 1958).

1956

Grigolyuk, E. I.: Inzh. Sb., vol. 23, p. 28, 1956.

Peshlmaldzhyan, D. V.: The Calculation of Symmetrically Loaded Laminated Anisotropic Rotating Shells. Izv. Akad. Nauk Arm. SSR, Ser. Fiz. Mat. Nauk, vol. 10, no. 2, 1956, pp. 39–54. (In Armenian.)

1957

AKSEL'RAD, E. L.: Tr. Leningr. In-ta Aviats. Prib., no. 24, p. 41, 1957.

AMBARTSUMIAN, S. A.: Izv. Akad. Nauk SSSR, Otd. Tekhn. Nauk, no. 1, p. 77, 1957.

AMBARTSUMIAN, S. A.: Izv. Akad. Nauk Arm. SSR, Fiz. Mat., Estestv. Tekhn. Nauki, vol. 10, no. 2, p. 17, 1957.

ARBARTSUMIAN, S. A.: On the Analysis of Two-Layered Orthotropic Shells. Izv. Akad. Nauk SSSR, Otd. Tekhn. Nauk, no. 7, July 1957, pp. 57–64. (In Russian.)

LEKHKITSKIY, G. S.: Anisotropic Plates. *In:* Gosudarstvennoe Izkatel'stvo, Tekhniko-Teoreticheskoi Literaturi (Moscow-Leningrad), 1957. (In Russian.)

PESHTMALDZHYAN, D. V.: Izv. Akad. Nauk Arm. SSR, Fiz. Mat., Estestv. Tekhn. Nauki, vol. 10, no. 2, p. 39, 1957.

1958

AKSEL'RAD, E. L.: Izv. Akad. Nauk SSSR, Otd. Tekhn. Nauk, no. 6, p. 73, 1958.

AKSEL'RAD, E. L.: Izv. Akad. Nauk SSSR, Otd. Tekhn. Nauk, no. 8, p. 48, 1958.

GRIGOLYUK, E. I.: On Stability of Three-Layered Shells and Plates Beyond the Elastic Limit. Izv. Akad. Nauk SSSR, Otd. Tekhn. Nauk, no. 6, 1958, pp. 68–72.

KURSHIN, L. M.: On Stability of a Three-Layered Shallow Cylindrical Shell Under Compression. Izv. Akad. Nauk SSSR, Otd. Tekhn. Nauk, no. 8, 1958, pp. 97–100.

1959

GAZIZOV, B. G.: Izv. Vysshikh. Uchebn. Zavedenii, Aviats. Tekhn., vol. 2, no. 4, p. 79, 1959.

1960

AKSEL'RAD, E. L.: Izv. Akad. Nauk SSSR, Otd. Tekhn. Nauk, Mekhan. i Mashinostr., no. 6, p. 158, 1960.

BAGDASARIAN, ZH. E.; AKD GNUNI, V. Ts.: Izv. Akad. Nauk Arm. SSR, Fiz. Mat., Estestv. Tekhn. Nauki, vol. 13, no. 5, p. 27, 1960.

BURMISTROV, E. F.: Inzh. Sb., vol. 27, p. 185, 1960.

GNUNI, V. Ts: Izv. Akad. Nauk Arm. SSR, Fiz. Mat., Estestv. Tekhn. Nauki, vol. 13, no. 1, p. 47, 1960.

SVIRSKII, I. V.; AND GALIMOY, N. K.: Izv. Kazansk. Filiala Akad. Nauk SSSR, Ser. Fiz. Mat. i Tekhn. Nauk, no. 14, p. 71, 1960.

UZDALEV, A. I.: Izv. Vysshikh Uchebn. Zavedenii, Stroit. i Arkhitek., no. 1, p. 30, 1960.

1961

AKSEL'RAD, E. L.: Izv. Akad. Nauk SSSR, Otd. Tekhn. Nauk, Mekhan. i Mashinostr., no. 2, p. 164, 1961.

1962

KOROLEV, V. I.; SMIRNOV, I. G.; AND STOMMA, R. P.: Study of Stability of Bimetallic Cylindrical Shells Under Axial Compression Beyond the Elastic Limit. Inzh. Zh., vol. 2, no. 1, 1962, pp. 98–110.

SACHENKOV, A. V.: Trans. Conf. Theory of Plates and Shells, L'vov, 128. Akad. Nauk Ukr. SSR Press (Kiev), 1962.

SANDWICH STRUCTURES

1933

DONNELL, L. H.: Stability of Thin-Walled Tubes Under Torsion. NACA Rept. 479.

NEUT, A. VAN DER: The Three-Point Bending Test of Wooden Box Beams. NLL Rept. S. 72. (In Dutch.)

1936

TIMOSHENKO, S.: Theory of Elastic Stability. McGraw-Hill Book Co., Inc., 2d ed. (New York), 1961.

1940

GOUGH, G. S.; ELAM, C. F.; AND DE BRUYNE, N. D.: The Stabilization of a Thin Sheet by a Continuous Supporting Medium. J. Roy. Aeron. Soc., vol. 44, no. 349, Jan. 1940, pp. 12–43.

TIMOSHENKO, S.: Theory of Plates and Shells. McGraw-Hill Book Co., Inc., 2d ed. (New York), 1959.

1941

BARWELL, F. T.: Some Tests on Sandwich Structures Loaded in Compression. Aeron. Res. Counc., Rept. 5401.

1942

LEGGETT, D. M. A.; AND HOPKINS, H. G.: Sandwich Panels and Cylinders Under Compressive End Loads. RAE Rept. SME 3203; also R. and M. 2262, 1949.

1943

NEUT, A. VAN DER: Die Stabilität geschichteter Streifen. NLL Rept. S. 284.

NEUT, A. VAN DER: Die Stabilität geschichteter Platten. NLL Rept. S. 286.

1944

FLÜGGE, W.; AND MARGUERRE, K.: Die optimale Knicklast eines Stabes der aus zwei durch einen leichten Fuellstoff verbundenen Blechen besteht. DVL, ZWB UM 1360.

HOPKINS, H. G.; AND PEARSON, S.: The Behaviour of Flat Sandwich Panels Under Uniform Transverse Load. RAE Rept. SME 3277.

MARCH, H. W.: Buckling Loads of Panels Having Light Cores and Dense Faces. FPL Rept. 1504.

MARCH, H. W.; SMITH, C. B.; AND KOMMERS, W. J.: Flexural Rigidity of a Rectangular Strip of Sandwich Construction. FPL Repts. 1505 and 1505–A.

MARGUERRE, K.: The Optimum Buckling Load of a Flexibly Supported Plate Composed of Two Sheets Joined by a Lightweight Filler, When Under Longitudinal Compression. Ministry of Supply, TPA 3/TIB Transl. 3477, GDC 10/5739 T.

1945

COX, H. L.; AND RIDDELL, J. R.: Sandwich Construction and Core Materials. pt. III. Instability of Sandwich Struts and Beams. R. and M. 2125.

DALE, F. A.; AND SMITH, R. C. T.: Grid Sandwich Panels in Compression. Australian Council Aeron., Rept. ACA–16.

HEMP, W. S.: The Theory of Sandwich Construction. Bristol Aeroplane Co., Ltd., Tech. Office, Rept. 29.

HOFF, N. J.; AND MAUTNER, S. E.: The Buckling of Sandwich Type Panels. J. Aeron. Sci., vol. 12, no. 3, July 1945, pp. 285–297.

MARCH, H. W.; AND SMITH, C. B.: Buckling Loads of Flat Sandwich Panels in Compression. FPL Rept. 1525.

REISSNER, E.: The Effect of Transverse Shear Deformation on the Bending of Elastic Plates. J. Appl. Mech., vol. 12, no. 2, June 1945, pp. A 69–77.

WITTRICK, W. H.: A Theoretical Analysis of the Efficiency of Sandwich Construction Under Compressive End Load. R. and M. 2016.

1946

BARWELL, F. T.; AND RIDDELL, J. R.: The Wrinkling of Sandwich Struts. R. and M. 2143.

BARWELL, F. T.: Sandwich Construction and Core Materials. Pt. IV. Notes on Some Methods of Testing Core Materials. R. and M. 2467.

BIJLAARD, P. P.: On the Elastic Stability of Thin Plates, Supported by a Continuous Medium. Proc. Koninkl. Ned. Akad. v. Wetanschap., vol. 44, no. 10, pp. 1189–1199.

COX, H. L.: The Requirements for Core Materials for Sandwich Construction. Aeron. Res. Council, Rept. 9774.

DONNELL, H. L.: The Effect of Transverse Shear Deformation on the Bending of Elastic Plates. (Discussion.) J. Appl. Mech., vol. 13, no. 3, Sept. 1946, pp. A 249–250.

GARRARD, A.: Theory of Sandwich Construction. Brit. Plastics, vol. 18, nos. 208 and 209, Sept. and Oct. 1946, pp. 380–388 and 451–458.

GOODIER, J. N.: Cylindrical Buckling of Sandwich Plates. J. Appl. Mech., vol. 13, no. 4, Dec. 1946, pp. A 253–260.

HU, C. PAI; LUNDQUIST, E. E.; AND BATDORF, S. B.: Effect of Small Deviations from Flatness on Effective Width and Buckling of Plates in Compression. NACA TN 1124.

KUENZI, E. W.: Design Criteria for Long Curved Panels of Sandwich Construction in Axial Compression. FPL Rept. 1558. Repr. Jan. 1959.

1947

BIJLAARD, P. P.: On the Elastic Stability of Sandwich Plates, I and II. Proc. Koninkl. Ned. Akad. v. Wetanschap., vol. 50, nos. 1 and 2, pp. 79–87 and 186–193.

BOLLER, K. H.: Buckling Loads of Flat Sandwich Panels in Compression. FPL Repts. 1525–A, B, C, and D.

BOLLER, K. H.: Preliminary Report on the Strength of Flat Sandwich Panels in Edgewise Compression. FPL Rept. 1561.

HOWARD, H. B.: A Note on Initial Irregularities in Sandwich Construction. Aeron. Res. Counc., Rept. 10371, Feb. 1947, and Amendment, Apr. 1947.

KUENZI, E. W.: Stability of a Few Flat Sandwich Panels Subjected to Shear. FPL Rept. 1560.

KUENZI, E. W.: Stability of a Few Curved Panels Subjected to Shear. FPL Rept. 1571. Repr. Aug. 1962.

NORRIS, C. B.: An Analysis of the Compressive Strength of Honeycomb Cores for Sandwich Construction. NACA TN 1251.

REISSNER, E.: On Bending of Elastic Plates. Quart. Appl. Math., vol. 5, no. 1, Apr. 1947, pp. 55–68.

TROXELL, W. W.; AND ENGEL, H. C.: Column Characteristics of Sandwich Panels Having Honeycomb Cores. J. Aeron. Sci., vol. 14, no. 7, July 1947, pp. 413–421.

WAN, C. C.: Face Buckling and Core Strength Requirements in Sandwich Construction. J. Aeron. Sci., vol. 14, no. 9, Sept. 1947, pp. 531–539.

WILLIAMS, D.: Sandwich Construction—A Practical Approach for the Use of Designers. RAE Rept. no. Structures 2.

1948

ANON.: Theory and Practice of Sandwich Construction in Aircraft. Inst. Aeron. Sci., Prepr. 165.

(a) NORRIS, C. B.: Research on Sandwich Constructions at the Forest Products Laboratory, pp. 4–12.

(b) HOFF, N. J.: The Buckling of Sandwich Structural Elements, pp. 13–20.

(c) REISSNER, E.: Contributions to the Problem of Structural Analysis of Sandwich-Type Plates and Shells, pp. 21–48.

(d) LIBOVE, CH.: A Small-Deflection Theory for Flexurally Orthotropic Flat Sandwich Plates, pp. 49–56.

ANON.: Methods of Test for Determining Strength Properties of Core Materials for Sandwich Construction at Normal Temperatures. FPL Rept. 1555.

ANON.: Methods for Conducting Mechanical Tests of Sandwich Constructions at Normal Temperatures. FPL Rept. 1556.

BENSCOTER, S. U.: Shear Flows in Multicell Sandwich Sections. NACA TN 1749.

BOLLER, K. H.: Buckling Loads of Flat Sandwich Panels in Compression. FPL Rept. 1525–E.

HEMP, W. S.: On a Theory of Sandwich Construction. College of Aeronautics, Cranfield, Rept. 15.

HOFF, N. J.; AND MAUTNER, S. E.: Bending and Buckling of Sandwich Beams. J. Aeron. Sci., vol. 15, no. 12, Dec. 1948, pp. 707–720.

KOMMERS, W. J.; AND NORRIS, C. B.: Effects of Shear Deformation in the Core of a Flat Rectangular Sandwich Panel. Stiffness of Flat Panels of Sandwich Construction Subjected to Uniformly Distributed Loads Normal to Their Surfaces—Simply Supported Edges. FPL Rept. 1583–A, Repr. Apr. 1956.

LIBOVE, C.; AND BATDORF, S. B.: A General Small Deflection Theory for Flat Sandwich Plates. NACA TN 1526; also NACA Rept. 899.

MARCH, H. W.: Effects of Shear Deformation in the Core of a Flat Rectangular Sandwich Panel. FPL Rept. 1583.

PLANTEMA, F. J.: Some Investigations on the Euler Instability of Flat Sandwich Plates With Simply-Supported Edges. Proc. VII Intnatl. Congr. Appl. Mech., I, pp. 200–213.

PULLEN, W. J.; CHAPMAN, R. G.; PEARSON, S.; AND OAKS, J. K.: Sandwich Construction and Core Materials. Pt. VI. R. and M. 2687.

REISSNER, E.: Finite Deflections of Sandwich Plates. J. Aeron. Sci., vol. 15, no. 7, July 1948, pp. 435–440. (Erratum in J. Aeron. Sci., vol. 17, no. 2, Feb. 1950, p. 125.)

REUTTER, F.: Ueber die Stabilität dreischichtiger Stäbe und Platten, deren mittlere aus einem Leichtstoff bestehende Schicht einem in Dickenrichtung veränderlichen Elastizitätsmodul hat. ZAMM, vol. 28, nos. 1 and 5, Jan. and May 1948, pp. 1–12 and 132–148.

TAYLOR, L. J.: Strength of Sandwich Panels. Proc. VII Intnatl. Congr. Appl. Mech., I, pp. 187–199.

1949

BIJLAARD, P. P.: Stability of Sandwich Plates. J. Aeron. Sci., vol. 16, no. 9, Sept. 1949, pp. 573–574.

BOLLER, K. H.; AND NORRIS, C. B.: Elastic Stability of the Facings of Flat Sandwich Panels When Subjected to Combined Edgewise Stresses. FPL Rept. 1802. Repr. Aug. 1955.

FLÜGGE, W.: Determination des Dimensions Optima des Plaques-Sandwichs. Rech. Aeron., Jan., Feb. 1949, no. 7, pp. 43–49.

HEEBINK, B. G.; AND MOHAUPT, A. A.: Effect of Defects on Strength of Aircraft Type Sandwich Panels. FPL Rept. 1809. Repr. 1962.

HOFF, N. J.: The Strength of Laminates and Sandwich Structural Elements. Engineering Laminates, A. G. H. Dietz, ed., John Wiley & Sons, Inc. (New York), pp. 6–88.

HUNTER-TOD, J. H.: The Elastic Stability of Sandwich Plates. College of Aeronautics, Cranfield, Rept. 25; also R. and M. 2778.

JACOBI, H.: Untersuchungen an stuetzstoffversteiften Verbundstaeben. Kunststoffe, vol. 39, nos. 11 and 12, Nov. and Dec. 1949, pp. 269–278 and 315–320.

NORRIS, C. B.; ET AL.: Wrinkling of the Facings of Sandwich Construction Subjected to Edgewise Compression. FPL Rept. 1810. Repr. Apr. 1961.

PLANTEMA, F. J.; AND KOCK, DE A. C.: The Elastic Overall Instability of Sandwich Plates With Simply-Supported Edges. NLL Rept. S. 346, Repr. and Trans. NLL XV, pp. S17–34.

REISSNER, E.: Small Bending and Stretching of Sandwich-Type Shells. NACA TN 1832; also NACA Rept. 975.

SCHWARTZ, R. T.: Structural Sandwich Construction. *In* Engineering Laminates, A. G. H. Dietz, ed., John Wiley & Sons, Inc. (New York), pp. 654–707.

SEIDE, P.; AND STOWELL, E. Z.: Elastic and Plastic Buckling of Simply Supported Metalite Type Sandwich Plates in Compression. NACA TN 1822; also NACA Rept. 967.

SEIDE, P.: Compressive Buckling of Flat Rectangular Metalite Type Sandwich Plates With Simply Supported Loaded Edges and Clamped Unloaded Edges. NACA TN 1886.

SEIDE, P.: Shear Buckling of Infinitely Long Simply Supported Metalite Type Sandwich Plates. NACA TN 1910.

TAYLOR, L. J.: Notes on Sandwich Construction. Aircraft Eng., vol. 21, no. 244, June 1949, p. 196.

1950

BIJLAARD, P. P.: Stability of Sandwich Plates in Combined Shear and Compression. J. Aeron. Sci., vol. 17, no. 1, Jan. 1950, p. 63.

BIJLAARD, P. P.: Investigation of the Optimum Distribution of Material in Sandwich Plates Loaded in Their Plane. Cornell Aeron. Lab. Rept. SA–247–S–8.

BOLLER, K. H.; AND NORRIS, C. B.: Effect of Shear Strength on Maximum Loads of Sandwich Columns. FPL Rept. 1815.

ERICKSEN, W. S.; AND MARCH, H. W.: Effects of Shear Deformation in the Core of a Flat Rectangular Sandwich Panel. Compressive Buckling of Sandwich Panels Having Facings of Unequal Thickness. FPL Rept. 1583–B. Rev. Nov. 1958.

ERICKSEN, W. S.: Effects of Shear Deformation in the Core of a Flat Rectangular Sandwich Panel. Deflection Under Uniform Load of Sandwich Panels Having Facings of Unequal Thickness. FPL Rept. 1583–C.

HOFF, N. J.: Bending and Buckling of Rectangular Sandwich Plates. NACA TN 2225.

MARCH, H. W.: Elastic Stability of the Facings of Sandwich Columns. Proc. Symp. Appl. Math., vol. 3, Elasticity. McGraw-Hill Book Co., Inc. (New York), pp. 85–106.

NORRIS, C. B.; AND KOMMERS, W. J.: Short-Column Compressive Strength of Sandwich Constructions as Affected by the Size of the Cells of Honeycomb-Core Materials. FPL Rept. 1817.

RINGELSTETTER, L. A.; VOSS, A. W.; AND NORRIS, C. B.: Effect of Cell Shape on Compressive Strength of Hexagonal Honeycomb Structures. NACA TN 2243.

STEIN, M.; AND MAYERS, J.: A Small-Deflection Theory for Curved Sandwich Plates. NACA TN 2017; also NACA Rept. 1008, 1951.

WERREN, F.; AND NORRIS, C. B.: Analysis of Shear Strength of Honeycomb Cores for Sandwich Construction. NACA TN 2208.

1951

BIJLAARD, P. P.: On the Optimum Distribution of Material in Sandwich Plates Loaded in Their Plane. Proc. 1st U.S. Natl. Congr. Appl. Mech., pp. 373–380.

BIJLAARD, P. P.: Analysis of the Elastic and Plastic Stability of Sandwich Plates by the Method of Split Rigidities. Inst. Aeron. Sci. Preprint 259; also J. Aeron. Sci., vol. 18, nos. 5 and 12, May and Dec. 1951, pp. 339–349 and 790–796, 829.

ERICKSEN, W. S.: Supplement to: Effects of Shear Deformation in the Core of a Flat Rectangular Sandwich Panel. Deflection Under Uniform Load of Sandwich Panels Having Facings of Moderate Thickness. FPL Rept. 1583–D.

ERINGEN, A. C.: Bending and Buckling of Rectangular Sandwich Plates. Proc. 1st U.S. Natl. Congr. Appl. Mech., pp. 381–390.

ERINGEN, A. C.: Buckling of a Sandwich Cylinder Under Uniform Axial Compressive Load. J. Appl. Mech., vol. 18, no. 2, June 1951, pp. 195–202.

GERARD, G.: Torsional Instability of a Long Sandwich Cylinder. Proc. 1st U.S. Natl. Congr. Appl. Mech., pp. 391–394.

GERARD, G.: Compressive and Torsional Instability of Sandwich Cylinders. Symp. on Structural Sandwich Constructions. ASTM Spec. Tech. Publ. 118, pp. 56–69.

GERARD, G.: Note on Bending of Thick Sandwich Plates. J. Aeron. Sci., vol. 18, no. 6, June 1951, pp. 424–427.

GOODIER, J. N.: Some Observations on Elastic Stability. Proc. 1st U.S. Natl. Congr. Appl. Mech., pp. 193–202.

GOODIER, J. N.; AND NEOU, I. M.: The Evaluation of Theoretical Critical Compression in Sandwich Plates. J. Aeron. Sci., vol. 18, no. 10, Oct. 1951, pp. 649–656.

HUBKA, R. E.; DOW, N. F.; AND SEIDE, P.: Relative Structural Efficiencies of Flat Balsa-Core Sandwich and Stiffened-Panel Construction. NACA TN 2514.

KUENZI, E. W.; AND ERICKSEN, W. S.: Shear Stability of Flat Panels of Sandwich Construction. FPL Rept. 1560, Rev.

KUENZI, E. W.: Edgewise Compressive Strength of Panels and Flatwise Flexural Strength of Strips of Sandwich Constructions. FPL Rept. 1827.

KUENZI, E. W.: Flexure of Structural Sandwich Construction. FPL Rept. 1829.

LIBOVE, CH.; AND HUBKA, R. E.: Elastic Constants for Corrugated Core Sandwich Plates. NACA TN 2289.

MARCH, H. W.: Sandwich Construction in the Elastic Range. Symp. on Structural Sandwich Constructions. ASTM Spec. Tech. Publ. 118, pp. 32–45.

MOHAUPT, A. A.; AND HEEBINK, B. C.: Effect of Defects on Strength of Aircraft Type Sandwich Panels. FPL Rept. 1809–A.

NORRIS, C. B.: Strength of Sandwich Construction. Symp. on Structural Sandwich Constructions. ASTM Spec. Tech. Publ. 118, pp. 46–53.

273-140 O - 67 - 10

TEICHMANN, F. K.; WANG, C. T.; AND GERARD, G.: Buckling of Sandwich Cylinders Under Axial Compression. J. Aeron. Sci., vol. 18, no. 6, June 1951, pp. 398–406.

TEICHMANN, F. K.; AND WANG, C. T.: Finite Deflections of Curved Sandwich Plates and Sandwich Cylinders. Inst. Aeron. Sci., Fairchild Publ. Fund, FF.4.

YEN, T. K.; GUNTURKUN, S.; AND POHLE, F. V.: Deflections of a Simply Supported Rectangular Sandwich Plate Subjected to Transverse Loads. NACA TN 2581.

1952

BIJLAARD, P. P.: Analysis of the Elastic and Plastic Stability of Sandwich Plates by the Method of Split Rigidities. III. J. Aeron. Sci., vol. 19, no. 7, July 1952, pp. 502–503.

BLEICH, F.: Buckling Strength of Metal Structures. McGraw-Hill Book Co., Inc. (New York).

ERINGEN, A. C.: Ripple-Type Buckling of Sandwich Columns. J. Aeron. Sci., vol. 19, no. 6, June 1952, pp. 409–417.

FLÜGGE, W.: The Optimum Problem of the Sandwich Plate. J. Appl. Mech., vol. 19, no. 1, Mar. 1952, pp. 104–108.

GERARD, G.: Linear Bending Theory of Isotropic Sandwich Plates by an Order-of-Magnitude Analysis. J. Appl. Mech., vol. 19, no. 1, Mar. 1952, pp. 13–15.

HORVEY, G.: Bending of Honeycombs and of Perforated Plates. J. Appl. Mech., vol. 19, no. 1, Mar. 1952, pp. 122–123.

MARCH, H. W.; AND KUENZI, E. W.: Buckling of Cylinders of Sandwich Construction in Axial Compression. FPL Rept. 1830. Rev. Dec. 1957.

MARCH, H. W.: Behavior of a Rectangular Sandwich Panel Under a Uniform Lateral Load and Compressive Edge Loads. FPL Rept. 1834.

NEUBER, H.: Theorie der Druckstabilität der Sandwichplatte. ZAMM, 32, 11/12, Nov./Dec. 1952, pp. 325–337 and 33, 1/2, Jan./Feb. 1953, pp. 10–26.

NORRIS, C. B.; AND KOMMERS, W. J.: Critical Loads of a Rectangular, Flat Sandwich Panel Subjected to Two Direct Loads Combined With a Shear Load. FPL Rept. 1833.

NORRIS, C. B.: Effect of Unbonded Joints in an Aluminum Honeycomb-Core Material for Sandwich Constructions. FPL Rept. 1835.

PLANTEMA, F. J.: Theory and Experiments on the Elastic Overall Instability of Flat Sandwich Plates. Ph. D. thesis, Delft; also NLL Rept. S. 402, Repr. and Trans. NLL, 17.

SEIDE, P.: Compressive Buckling of Flat Rectangular Metalite Type Sandwich Plates With Simply Supported Loaded Edges and Clamped Unloaded Edges. NACA TN 2637, rev.

SEIDE, P.: The Stability Under Longitudinal Compression of Flat Symmetric Corrugated-Core Sandwich Plates With Simply Supported Loaded Edges and Simply Supported or Clamped Unloaded Edges. NACA TN 2679.

STEIN, M.; AND MAYERS, J.: Compressive Buckling of Simply Supported Curved Plates and Cylinders of Sandwich Construction. NACA TN 2601.

VOSS, A. W.: Mechanical Properties of Some Low-Density Core Materials. FPL Rept. 1826.

WANG, C. T.; AND RAO, G. V. R.: A Study of an Analogous Model Giving the Non-Linear Characteristics in the Buckling Theory of Sandwich Cylinders. J. Aeron. Sci., vol. 19, no. 2, Feb. 1952, pp. 93–100.

WANG, C. T.; AND SULLIVAN, D. P.: Buckling of Sandwich Cylinders Under Bending and Combined Bending and Axial Compression. J. Aeron. Sci., vol. 19, no. 7, July 1952, pp. 468–470.

WANG, C. T.: Principle and Application of Complementary Energy Method for Thin Homogeneous and Sandwich Plates and Shells With Finite Deflections. NACA TN 2620.

WERREN, F.; AND LEWIS, W. C.: Fatigue of Sandwich Constructions for Aircraft. FPL Rept. Ser. 1559 and 1559–A through K, Dec. 1946–July 1952.

WERREN, F.: Shear Fatigue Properties of Various Sandwich Constructions. FPL Rept. 1837.

YEN, KO TO; SALERNO, V. L.; AND HOFF, N. J.: Buckling of Rectangular Sandwich Plates Subject to Edgewise Compression With Loaded Edges Simply Supported and Unloaded Edges Clamped. NACA TN 2556.

1953

ERICKSEN, W. S.: The Bending of a Circular Sandwich Panel Under Normal Load. FPL Rept. 1828, Rev.

GERARD, G.: Bending Tests on Thin-Walled Sandwich Cylinders. J. Aeron. Sci., vol. 20, no. 9, Sept. 1952, pp. 639–641.

KROLL, W. D.; MORDFIN, L.; AND GARLAND, W. A.: Investigation of Sandwich Construction Under Lateral and Axial Loads. NACA TN 3090.

LINDOP, R.: Analysis of Test Results on Flat Long Column Sandwich Panels. A. V. Roe & Co., Ltd., Rep. Struct. 10. (Unpub.)

MARCH, H. W.; AND KUENZI, E. W.: Buckling of Sandwich Cylinders in Torsion. FPL Rept. 1840. (Also see ref. 58–13.)

NARDO, S. V.: An Exact Solution for the Buckling Load of Flat Sandwich Panels With Loaded Edges Clamped. J. Aeron. Sci., vol. 20, no. 9, Sept. 1953, pp. 605–612.

NORRIS, C. B.; BOLLER, K. H.; AND VOSS, A. W.: Wrinkling of the Facings of Sandwich Construction Subjected to Edgewise Compression. Sandwich Constructions Having Honeycomb-Cores. FPL Rept. 1810–A.

NORRIS, C. B.; AND KOMMERS, W. J.: Stresses Within a Rectangular, Flat Sandwich Panel Subjected to a Uniformly Distributed Normal Load and Edgewise Direct and Shear Loads. FPL Rept. 1838.

PLANTEMA, F. J.; AND VAN ALPHEN, W. J.: Compressive Buckling of Sandwich Plates Having Various Edge Conditions. C. B. Biezeno Anniv. vol. on Appl. Mech., Stam, Haarlem (Holland), pp. 132–148; also NLL Rept. S. 404.

VODICKA, V.: Durchbiegung von geschichteten Platten mit leichter Fuellung. ZAMM, 33, 5/6, May/June 1953, pp. 188–189.

1954

GOODIER, J. N.; AND HSU, C. S.: Nonsinusoidal Buckling Modes of Sandwich Plates. J. Aeron. Sci., vol. 21, no. 8, Aug. 1954, pp. 525–532.

RAVILLE, M. E.: Analysis of Long Cylinders of Sandwich Construction Under Uniform External Lateral Pressure. FPL Rept. 1844.

SEIDE, P.: Comments on Ripple-Type Buckling of Sandwich Columns. J. Aeron. Sci., vol. 21, no. 4, Apr. 1954, pp. 282–286.

1955

ANON.: Sandwich Construction for Aircraft. ANC–23, pts. I and II. Munitions Bd.; Aircraft Comm. U.S. Government Printing Office (Washington, D.C.).

ERINGEN, A. C.: Some Corrections and Further Contributions to "Ripple-Type Buckling of Sandwich Columns." J. Aeron. Sci., vol. 22, no. 2, Feb. 1955, pp. 142–144.

HAFT, E. E.: Elastic Stability of Cylindrical Sandwich Shells Under Axial and Lateral Load. FPL Rept. 1852.

IKEDA, K.: Theory of Bending of Isotropic Flat Sandwich Plates and Its Applications. Proc. 5th Japan Natl. Congr. Appl. Mech., pp. 103–106.

KUENZI, E. W.: Mechanical Properties of Aluminum Honeycomb Cores. FPL Rept. 1849.

NORRIS, C. B.; AND BOLLER, K. H.: Transfer of Longitudinal Load From One Facing of a Sandwich Panel to the Other by Means of Shear in the Core. FPL Rept. 1846.

RAVILLE, M. E.: Supplement to: Analysis of Long Cylinders of Sandwich Construction Under Uniform External Lateral Pressure. FPL Rept. 1844–A.

RAVILLE, M. E.: Buckling of Sandwich Cylinders of Finite Length Under Uniform External Lateral Pressure. FPL Rept. 1844–B.

RAVILLE, M. E.: Deflection and Stresses in a Uniformly Loaded, Simply Supported, Rectangular Sandwich Plate. FPL Rept. 1847, Repr. Sept. 1962.

ROBINSON, J. R.: The Buckling and Bending of Orthotropic Sandwich Panels With All Edges Simply-Supported. Aeron. Quart., vol. 6, no. 2, May 1955, pp. 125–148.

SOLVEY, J.: Bibliography and Summaries of Sandwich Constructions (1939–1954). Aeron. Res. Lab. (Austr.), Bibl. SM.2.

WANG, C. T.; VACCARO, R. J.; AND DESANTO, D. F.: Buckling of Sandwich Cylinders Under Combined Compression, Torsion, and Bending Loads. J. Appl. Mech., vol. 22, no. 3, Sept. 1955, pp. 324–328.

YOUNGQUIST, W. G.; AND KUENZI, E. W.: Stresses Induced in a Sandwich Panel by Load Applied at an Insert. FPL Repts. 1845, 1845–A, and 1845–B, 1956.

YUSUFF, S.: Theory of Wrinkling in Sandwich Construction. J. Roy. Aeron. Soc., vol. 59, no. 529, Jan. 1955, pp. 30–36.

1956

ANON.: Symposium on Structural Sandwich Constructions. ASTM Spec. Tech. Publ. 201.

ANON.: Fatigue of Sandwich Constructions for Aircraft. USDA, Forest Serv., Forest Products Lab. Rept. No. 1559–H, Madison, Wis., 1956.

BRUNER, G.; AND MONTANTÊME, G.: Les Nids d'Abeille en Alliage Léger et Leur Applications à la Construction Sandwich. Docaéro, no. 37, Mar. 1956, pp. 39–56.

ERINGEN, A. C.: New Numerical Results of the Theory of Buckling of Sandwich Cylinders. J. Appl. Mech., vol. 23, no. 3, Sept. 1956, pp. 476–477.

FRÉTIGNY, G.: Calcul des Caissons Rectangulaires Symetriques avec Remplissage en Nids d'Abeilles. Docaéro, no. 39, July 1956, pp. 13–22.

GERARD, G.: Minimum-Weight Analysis of Compression Structures. New York Univ. Press.

HOFF, N. J.: The Analysis of Structures. John Wiley & Sons, Inc. (New York).

JOHNSON, A. E., JR.; AND SEMONIAN, J. W.: A Study of the Efficiency of High-Strength Steel, Cellular-Core Sandwich Plates in Compression. NACA TN 3751.

KIMEL, W. R.: Elastic Buckling of a Simply-Supported Rectangular Sandwich Panel Subjected to Combined Edgewise Bending and Compression, with supplement. FPL Repts. 1857 and 1857–A, repr. Aug. 1962.

KIMEL, W. R.: Elastic Buckling of a Simply-Supported Rectangular Sandwich Panel Subjected to Combined Edgewise Bending, Compression and Shear. FPL Rept. 1859.

KUENZI, E. W.: Structural Sandwich Construction. USDA, Forest Serv., Forest Products Lab., P and E 121, Madison, Wis., 1956.

KUENZI, E. W.; AND SETTERHOLM, V. C.: Mechanical Properties of Aluminum Multiwave Cores. FPL Rept. 1855.

KUENZI, E. W.: Methods of Testing Sandwich Constructions at Elevated Temperatures. FPL Rept. 2063.

LEWIS, W. C.: Deflection and Stresses in a Uniformly Loaded Simply-Supported Rectangular Sandwich Plate. Experimental Verification of Theory. FPL Rept. 1847-A.

McCOMB, H. G., JR.: Torsional Stiffness of Thin-Walled Shells Having Reinforcing Cores and Rectangular, Triangular, or Diamond Cross Section. NACA TN 3749; also NACA Rept. 1316, 1957.

NICHOLS, R.: Stability of Honeycomb Sandwich Panels in Shear. College of Aeronautics, Note No. 55.

NORRIS, C. B.: Compressive Buckling Design Curves for Sandwich Panels With Isotropic Facings and Orthotropic Cores. FPL Rept. 1854. Rev. Jan. 1958.

SEIDE, P.: On the Torsion of Rectangular Sandwich Plates. J. Appl. Mech., vol. 23, no. 2, June 1956, pp. 191–194.

1957

ANDERSON, R. A.: Weight-Efficiency Analysis of Thin-Wing Construction. Trans. ASME, vol. 79, no. 5, July 1957, pp. 974–979.

ANDERSON, M. S.; AND UPDEGRAFF, R. G.: Some Research Results on Sandwich Structures. NACA TN 4009.

DORLÉAC, B.; AND MONNIER R.: Calcul des Flux de Cisaillement dans un Caisson Cylindrique de Section Symétrique quelconque Rempli de "Nids d'Abeilles" Métalliques. Docaéro, no. 47, Nov. 1957, pp. 17–32.

FREIBERGER, W. F.: On the Minimum Weight Design Problem for Cylindrical Sandwich Shells. J. Aeron. Sci., vol. 24, no. 11, Nov. 1957, pp. 847, 848.

GUEST, J.; AND SOLVEY, J.: Elastic Stability of Rectangular Sandwich Plates Under Bi-Axial Compression. Aeron. Res. Lab. (Austr.), Rept. SM.251.

KAECHELE, L. E.: Minimum-Weight Design of Sandwich Panels. Rand Corp. RM–1895.

KUENZI, E. W.: Mechanical Properties of Glass-Fabric Honeycomb Cores. FPL Rept. 1861.

ONAT, E. T.; SCHUMANN, W.; AND SHIELD, R. T.: Design of Circular Plates for Minimum Weight. J. Appl. Math. Phys. (ZAMP), vol. 8, no. 6, pp. 485–499.

PIAN, T. H. H.; AND JOHNSON, R. I.: On Creep Buckling of Columns and Plates. Aeroelastic and Struct. Lab., Mass. Inst. Technol., TR 25–24.

THURSTON, G. A.: Bending and Buckling of Clamped Sandwich Plates. J. Aeron. Sci., vol. 24, no. 6, June 1957, pp. 407–412.

WILLIS, J. G.: Some Notes on Sandwich Design for Minimum Weight as Applied to Airplane Wings. Aeron. Eng. Rev., Oct. 1957, pp. 44–47.

1958

ASHLEY, H. R.: Sandwich Structure for High-Temperature Vehicles. AGARD Rept. 216.

ANDERSON, M. S.: Local Instability of the Elements of a Truss-Core Sandwich Plate. NACA TN 4292. Also NASA TR R–30, 1959.

BIJLAARD, P. P.: Method of Split Rigidities and Its Application to Various Buckling Problems. NACA TN 4085.

CHANG, C. C.; FANG, B. T.; AND EBCIOGLU, I. K.: Elastic Theory of a Weak-core Sandwich Panel Initially Warped, Simply Supported and Subjected to Combined Loadings. Proc. 3d U.S. Natl. Congr. Appl. Mech., pp. 273–280.

DORLÉAC, B.: De quelques Réalisations Françaises dans le Domaine des Structures Nouvelles. Docaéro, no. 50, May 1958, pp. 3–14.

GERARD, G.: An Evaluation of Structural Sheet Materials in Missile Applications. Jet Propulsion, vol. 28, no. 8, 1958, pp. 511–520.

GRIGOLYUK, E. I.: Buckling of Sandwich Construction Beyond the Elastic Limit. J. Mech. Phys. Solids, vol. 6, no. 4, July 1958, pp. 253–266.

HOFFMAN, G. A.: Poisson's Ratio for Honeycomb Sandwich Cores. J. Aerospace Sci., vol. 25, no. 8, Aug. 1958, pp. 534–535.

KARAVANOV, V. F.: Equations for Slanting Three-Layer Shells With a Light Filler During Finite Displacements. See Appl. Mech. Rev., vol. 14, no. 3, Mar. 1961, p. 193, Rev. 1287. (In Russian.)

KELSEY, S.; GELLATLY, R. A.; AND CLARK, B. W.: The Shear Modulus of Foil Honeycomb Core. Aircraft Eng., vol. 30, no. 356, pp. 294–302, Oct. 1958.

KRIVETSKY, A.: Approximate Core Thickness Required for Sandwich Plate and Shells in Compression. Rept. no. 7–58–0252–1, Bell Aircraft Corp., Aug. 1958.

KURANISHI, M.: The Behavior of Sandwich Structures Involving Stress, Temperature and Time Dependent Factors. IUTAM Symp. Non-Homogeneity in Elasticity and Plasticity (Warsaw), Sept. 1958; Bull. Acad. Polon. Sci., VII, 2–3, p. 169.

MARCH, H. W.; AND KUENZI, E. W.: Buckling of Sandwich Cyclinders in Torsion. FPL Rept. 1840, rev.

NORRIS, C. B.: Compressive Buckling Curves for Simply Supported Sandwich Panels With Glass-Fabric-Laminate Facings and Honeycomb Cores. FPL Rept. 1867.

SCHOELLER, W. C.: Calculating Thermal Stresses in Sandwich Panels. Aviation Age Res. and Devel. Tech. Handbook 1957–1958, pp. B–6 to B–8.

SEMONIAN, J. W.; AND CRAWFORD, R. F.: Some Methods for the Structural Design of Wings for Application Either at Ambient or Elevated Temperatures. Trans. ASME, vol. 80, no. 2, Feb. 1958, pp. 419–426.

SWANN, R. T.: Heat Transfer and Thermal Stresses in Sandwich Panels. NACA TN 4349.

THOMAS, C.: Pratique du Sandwich Nid d'Abeilles Alliage Léger. AGARD Rept. 217.

1959

ANON.: Honeycomb Sandwich Design. Hexcel Products, Inc., 1959.

AKASAKA, T.; AND TAKAGISHI, T.: The Shear Modulus of Foil Honeycomb Sandwich Structures. Trans. Japan Soc. Aeron. Space Sci., vol. 2, no. 3, 1959, pp. 83–90.

ANDERSON, M. S.: Optimum Proportions of Truss-Core and Web-Core Sandwich Plates Loaded in Compression. NASA TN D–98.

BIJLAARD, P. P.: Thermal Stresses and Deflections in Rectangular Sandwich Plates. J. Aerospace Sci., vol. 26, no. 4, Apr. 1959, pp. 210–218.

CHANG, C. C.; AND EBCIOGLU, I. K.: Thermo-Elastic Behavior of Simply-Supported Sandwich Panel Under High Temperature Gradient and Edge Compression. IAS Rept. no. 59–67.

CHENG, S.: Torsion of Rectangular Sandwich Plates. FPL Rept. 1871.

GATEWOOD, B. E.; AND JONES, C. R.: Optimum Design of Stiffened Panels and Sandwich Panels at Elevated Temperature. Proc. 4th Midwestern Conf. Solid Mech., Austin, Tex., Univ. Press, Sept. 1959, pp. 270–297.

GERARD, G.; AND BECKER, H.: Strength of Thin-Wing Construction, pt. VII. *In:* Handbook of Structural Stability. NASA TN D–162.

HEATH, W. G.: The Structural Effects of Kinetic Heating. Some Factors Affecting the Choice of Materials. J. Roy. Aeron. Soc., vol. 63, no. 587, Nov. 1959, pp. 615–619.

JAHNKE, W. E.; AND KUENZI, E. W.: Buckling of Simply Supported Rectangular Sandwich Panels Subjected to Edgewise Bending. FPL Rept. 1868.

KIMEL, W. R.; ET AL.: Natural Frequencies of Vibration of Simply-Supported Sandwich Beams. Proc. 4th Midwestern Congr. Solid Mech., Austin, Tex. Univ. Press, Sept. 1959, pp. 441–456.

KUENZI, E. W.: Effect of Length on the Buckling Stresses of Thin-Walled, Plywood Cylinders in Axial Compressions. USDA, Forest Serv., Forest Products Lab., Rept. no. 1514, Jan. 1959.

KUENZI, E. W.; AND JAHNKE, W. E.: Mechanical Properties of Some Heat-Resistant Metal Honeycomb Cores. FPL Rept. 1872.

KUENZI, E. W.: Structural Sandwich Design Criteria. FPL Rept. 2161. Repr. 1963.

MATHAUSER, E. E.; AND PRIDE, R. A.: Compressive Strength of Stainless-Steel Sandwiches at Elevated Temperature. NASA Memo 6–2–59 L.

NAGAO, H.: The Shear Modulus of Honeycomb Cores. Proc. 9th Japan Natl. Congr. Appl. Mech., 1959, pp. 97–100.

NORRIS, C. B.; AND ZAHN, J. J.: Design Curves for the Buckling of Sandwich Cylinders of Finite Length Under Uniform External Lateral Pressure. FPL Rept. 1869.

ONAT, E. T.; AND YAMANTURK, S.: On Thermally Stressed Elastic-Plastic Shells. Brown Univ., Div. Eng., Tech. Rept. 12.

PRAGER, W.; AND SHIELD, R. T.: Minimum Weight Design of Circular Plates Under Arbitrary Loading. ZAMP, vol. 10, no. 4, July 1959, pp. 421–426.

PRATT, J. S.: The Applicability of Sandwich Construction to Missile Shapes. 1959 First Award Papers, IAS Student Branch Paper Competition. Inst. Aeron. Sci., The Minta Martin Aeronautical Student Fund (1959), pp. 109–119.

SAPOWITH, A. D.: Transverse Shear Stiffness for the Double "V" Corrugated-Core Sandwich Panel. Aero/Space Eng., vol. 18, no. 9, Sept. 1959, pp. 53–56.

SWANN, R. T.: Calculated Effective Thermal Conductivities of Honeycomb Sandwich Panels. NASA TN D–171.

WEIKEL, R. C.; AND KOBAYASHI, A. S.: On the Local Elastic Stability of Honeycomb Face Plate Subjected to Uniaxial Compression. J. Aerospace Sci., vol. 26, no. 10, Oct. 1959, pp. 672–674.

YU, Y.-Y.: A New Theory of Elastic Sandwich Plates. J. Appl. Mech., vol. 26, no. E3, Sept. 1959, pp. 415–421.

YU, Y.-Y.: Simple Thickness-Shear Modes of Vibration of Infinite Sandwich Plates. J. Appl. Mech., vol. 26, no. E4, Dec. 1959, pp. 679–681.

ZAHN, J. J.: Simply-Supported Sandwich Beam. A Non-Linear Theory. FPL Rept. 2157.

1960

ANON.: "Aeroweb" Honeycomb Design Information Sheets, 1 to 8. CIBA (ARL), Ltd. (Duxford, Cambridge).

ANON: Conference on Sandwich Panel Design Criteria, Building Research Institute, NAS–NRC, Publication 798, Washington, D.C.

ARNOLD, L. H.: Design of Sandwich Cylinders Under Axial Compressive Load. Missile and Space Systems Div., Douglas Aircraft Co., TM 18.

CHANG, C. C.; AND EBCIOGLU, I. K.: Elastic Instability of Rectangular Sandwich Panel of Orthotropic Core With Different Face Thicknesses and Materials. J. Appl. Mech., vol. 27, no. 3, Sept. 1960, pp. 474–480.

CHANG, C. C.; AND FANG, B. T.: Initially Warped Sandwich Panel Under Combined Loadings. J. Aerospace Sci., vol. 27, no. 10, Oct. 1960, pp. 779–787.

CHANG, C. C.; AND FANG, B. T.: Flexural Vibrations of a Rectangular Sandwich Panel. IAS Paper 60–21.

CHENG, S.: Torsion of Sandwich Panels of Trapezoidal, Triangular and Rectangular Cross Sections. FPL Rept. 1874; with supplement FPL Rept. 1874–A.

DITTOE, F. A.; AND HAUSRATH, A. H.: Estimated Design (for) Allowable Buckling Stresses for the Stability of Spherical Shells of Conventional Sandwich Construction Under Uniform External Pressure. AS-D-690, Convair Aviation, general.

EASON, G.: The Minimum Weight Design of Circular Sandwich Plates. ZAMP, vol. XI, no. 5, Sept. 25, 1960, pp. 368–375.

FULTON, R. E.: Buckling Analysis and Optimum Proportions of Sandwich Cylindrical Shells Under Hydrostatic Pressure. Civil Eng. Studies, Structural Res. Ser. 199, Dept. Civil Eng., Univ. Illinois.

GOREE, W. S.; AND NASH, W. A.: Elastic Stability of Circular Cylindrical Shell Stabilized By a Soft Elastic Core. Univ. of Florida, Dept. of Eng Mechanics, Eng. and Ind. Expt. Sta., Interim Tech. Rept. 4 for Office of Ordnance Res., U.S. Army. Also see Exptl. Mech., vol. 2, no. 5, May 1962, pp. 142–149.

HARRIS, L. A.; AND AUELMANN, R. A.: Stability of Flat Simply-Supported, Corrugated-Core Sandwich Plates Under Combined Loads. J. Aerospace Sci., vol. 27, no. 7, July 1960, pp. 525–534.

HEATH, W. G.: Sandwich Construction. Correlation and Extension of Existing Theory of Flat Panels Subjected to Lengthwise Compression. Aircraft Eng., vol. 32, nos. 377 and 378, July and Aug. 1960, pp. 186–191 and 230–235.

KAZIMI, M. I.: Sandwich Cylinders. Aerospace Eng., vol. 19, nos. 8 and 9, Aug. and Sept. 1960, pp. 32–37, 46, and 34–45.

MEAD, D. J.; AND FROUD, G. R.: The Damping of Aluminum Honeycomb Sandwich Beams. Univ. Southampton, USAA Rept. 144.

MEAD, D. J.: A Note on the Use of Sandwich Structures in Severe Acoustic Environments. Univ. Southampton, USAA Rept. 145.

NASH, W. A.: Recent Advances in the Buckling of Thin Shells. Appl. Mech. Rev., vol. 13, no. 3, Mar. 1960, pp. 161–164.

NORRIS, C. B.: Compressive Buckling Curves for Flat Sandwich Panels With Dissimilar Facings. FPL Rept. 1875.

NORRIS, C. B.; AND ZAHN, J. J.: Compressive Buckling Curves for Sandwich Cylinders Having Orthotropic Facings. FPL Rept. 1876.

PRAGER, W.: On the Plastic Analysis of Sandwich Structures. Div. Appl. Math., Brown Univ., Tech. Rept. 57; also in: Probl. Continuum Mech., Soc. Ind. and Appl. Math., 1961, pp. 342–349.

SEMONIAN, J. W.: The Bending Strength and Structural Efficiency of Full-Depth-Core Sandwich Wings. WADC TN 59-397.

SHIELD, R. T.: Plate Design for Minimum Weight. Quart. Appl. Math., vol. 18, no. 2, July 1960, pp. 131–144.

SHIELD, R. T.: On the Optimum Design of Shells. J. Appl. Mech., vol. 27, no. 2, June 1960, pp. 316–322.

SHIELD, R. T.: Optimum Design Methods for Structures. *In:* Plasticity. Pergamon Press, 1960.

WEIKEL, R. C.; AND KOBAYASHI, A. S.: Further Studies on the Local Elastic Deformations of Honeycomb Faceplate Subjected to Uniaxial Compression. J. Aerospace Sci., vol. 27, no. 12, Dec. 1960, pp. 961–962.

YUSUFF, S.: Face Wrinkling and Core Strength in Sandwich Construction. J. Roy. Aeron. Soc., vol. 64, no. 591, Mar. 1960, pp. 164–167.

YU, Y.-Y.: Forced Flexural Vibrations of Sandwich Plates in Plane Strain. J. Appl. Mech., vol. 27, no. 3, Sept. 1960, pp. 535–540.

YU, Y.-Y.: Flexural Vibrations of Elastic Sandwich Plates. J. Aerospace Sci., vol. 27, no. 4, Apr. 1960, pp. 272–282, 290.

YU, Y.-Y.: Simplified Vibration Analysis of Elastic Sandwich Plates. J. Aerospace Sci., vol. 27, no. 12, Dec. 1960, pp. 894–900.

YU, Y.-Y.: Vibrations of Elastic Sandwich Cylindrical Shells. J. Appl. Mech., vol. 27, no. E4, Dec. 1960, pp. 653–662.

1961

BIENIEK, M. P.; AND FREUDENTHAL, A. M.: Frequency-Response Functions of Orthotropic Sandwich Plates. J. Aerospace Sci., vol. 28, no. 9, Sept. 1961, pp. 732–735, 752.

CHANG, C. C.; AND EBCIOGLU, I. K.: Effect of Cell Geometry on the Shear Modulus and on Density of Sandwich Panel Cores. J. Basic Eng. (Trans. ASME), vol. 83, no. D4, Dec. 1961, pp. 513–518.

CHANG, C. C.; AND EBCIOGLU, I. K.: Thermoelestic Behavior of a Simply-Supported Sandwich Panel Under Large Temperature Gradients and Edge Compression. J. Aerospace Sci., vol. 28, no. 6, June 1961, pp. 480–482.

CHANG, C. C.; AND FANG, B. T.: Transient and Periodic Response of a Loaded Sandwich Panel. J. Aerospace Sci., vol. 28, no. 5, May 1961, pp. 382–396.

CHENG, S.: A Formula for the Torsional Stiffness of Rectangular Sandwich Plates. J. Appl. Mech., vol. 28, no. 2, June 1961, pp. 307–309.

CHENG, S.: Torsion of Sandwich Plates of Trapezoidal Cross Section. J. Appl. Mech., vol. 28, no. E3, Sept. 1961, pp. 363–366.

CHU, H.-N.: Influence of Transverse Shear on Non-Linear Vibrations of Sandwich Beams With Honeycomb Cores. J. Aerospace Sci., vol. 28, no. 5, May 1961, pp. 405–410.

CHU, H.-N.: On Simple Thickness Vibrations of Thin Sandwich Cylinders. J. Appl. Mech., vol. 28, no. 1, Mar. 1961, pp. 145–146.

CHU, H.-N.: Simple Axisymmetric Thickness Vibrations of a Soft Elastic Cylinder With a Hard, Thin, Elastic Skin. J. Acoust. Soc. Am., vol. 33, no. 10, Oct. 1961, pp. 1293–1295.

CHU, H.-N.: Vibrations of Honeycomb Sandwich Cylinders. J. Aerospace Sci., vol. 28, no. 12, Dec. 1961, pp. 930–939, 944.

COURRIER, G.; AND STRAT, H.: Calculs des Écoulements de Cisailement dans les Élements d'un Caisson Prismatique de Section Trapézoïdale Garni d'un Corps en Nid d'Abeilles. Docaéro, no. 68, May 1961, pp. 11–32.

FALGOUT, TH. E.: A Differential Equation of Free Transverse Vibrations of Isotropic Sandwich Plates. *In:* Developments in Mechanics, vol. 1, Proc. 7th Midwest. Mech. Congr., Michigan State Univ., Sept. 6–8, 1961. Eds. J. E. Lay and L. E. Malvern, North-Holland Publ. Co. (Amsterdam), pp. 223–227.

FORAL, R. F.: Sandwich Construction for Primary Structure of Ballistic Missiles and Space Vehicles. IAS Paper no. 61–1.

FULTON, R. E.: Nonlinear Equations for a Shallow Unsymmetrical Sandwich Shell of Double Curvature. *In:* Developments in Mechanics, vol. 1, Proc. 7th Midwest. Mech. Conf., Michigan State Univ., Sept. 6–8, 1961. Eds. J. E. Lay and L. E. Malvern, North-Holland Publ. Co. (Amsterdam), pp. 365–380.

GLASER, A. R.: The Vibration of Sandwich Beams. *In:* Developments in Mechanics, vol. 1, Proc. 7th Midwest. Mech. Conf., Michigan State Univ., Sept. 6–8, 1961. Eds. J. E. Lay and L. E. Malvern, North-Holland Publ. Co. (Amsterdam), pp. 228–238.

HU, T. C.; AND SHIELD, R. T.: Uniqueness in the Optimum Design of Structures. J. Appl. Mech., vol. 28, no. 2, June 1961, pp. 284–287.

JOANIDES, J. C.; MELLIN, S. C.; AND LACKMAN, L. M.: Mach 3 Wing Structures—Stiffened Skin Versus Sandwich. SAE Transl., vol. 69, pp. 167–178.

Koch, J. E.: Plane-Strain Bending of Sandwich Plates. *In:* Developments in Mechanics, vol. 1, Proc. 7th Midwest. Mech. Conf., Michigan State Univ., Sept. 6–8, 1961. Eds. J. E. Lay and L. E. Malvern, North-Holland Publ. Co. (Amsterdam), pp. 307–324.

Lu, S. Y.; and Nash, W. A.: Buckling of Thin Cylindrical Shells Stiffened by Soft Elastic Core. *In:* A. Paduart and R. Dutron, eds. Simplified Calculation Methods of Shell Structures, pp. 475–481. Proc. Coll. Brussels, Sept. 4–6, 1961. North-Holland Publ. Co. (Amsterdam), 1962.

Mead, D. J.; and Pretlove, A. J.: On the Vibrations of Cylindrically Curved Elastic Sandwich Plates. Pt. 1: With the Solution for Flat Plates. Univ. Southampton, AASU Rept. 186.

Raville, M. E.; Ueng, E. S.; and Lei, M. M.: Natural Frequencies of Vibration of Fixed-Fixed Sandwich Beams. J. Appl. Mech., vol. 28, no. E3, Sept. 1961, pp. 367–372.

Ricard, G.: Flugzeugstrukturen in Honigwabenbauweise. Luftfahrttechnik, vol. 7, no. 7, July 1961, pp. 192–205.

Seide, P.: Comments on Stability of Flat, Simply-Supported Corrugated-Core Sandwich Plates Under Combined Loads. J. Aerospace Sci., vol. 28, no. 3, Mar. 1961, p. 248.

Smallen, H.; and Roberts, W. F.: Properties of Stainless Steel Sandwich Using Low-Density Honeycomb Cores. Welding J., vol. 40, no. 2, Feb. 1961, pp. 90s–96s.

O'Sullivan, H. P.: Double Block Shear Test for Foil Honeycomb Cores. Aircraft Eng., vol. 33, no. 385, Mar. 1961, pp. 64–66.

Yao, J. C.: Thermal-Stress Analysis of Sandwich-Type Cylindrical Shells by the Cross Method. Aerospace Eng., vol. 20, no. 8, Aug. 1961, pp. 24, 25, 88–93.

Zak, A. R.; and Bollard, R. J. H.: Buckling of Thin, Short Cylindrical Shells Filled With an Elastic Core. *In:* Developments in Mechanics, vol. 1, Proc. 7th Midw. Mech. Conf:, Michigan State Univ., Sept. 6–8, 1961. Eds. J. E. Lay and L. E. Malvern, North-Holland Publ. Co. (Amsterdam), pp. 87–99.

1962

Anon.: Composite Construction for Flight Vehicles. Pt. III: Design Procedures. MIL–HDBK–23, Pt. III. Rev. Oct. 1962. U.S. Government Printing Office (Washington, D.C.).

Brush, D. O.: Some Shell Stability Problems in Missile and Space Vehicle Analysis. *In:* Collected Papers on Instability of Shell Structures—1962—NASA TN D–1510, pp. 35–44.

Brush, D. O.; and Almroth, B. O.: Buckling of Core-Stabilized Cylinders Under Axisymmetric External Loads. J. Aerospace Sci., vol. 29, no. 10, Oct. 1962, pp. 1164–1170.

Chang, C. C.; Ebcioglu, I. K.; and Haight, C. H.: General Stability Analysis of Orthotropic Sandwich Panels for Four Different Boundary Conditions. ZAMM, vol. 42, no. 9, Sept. 1962, pp. 373–389.

Chang, C. C.; and Timmons, M. J.: Compression Tests of Sandwich Panels With Facings at Different Temperatures. Exptl. Mech., vol. 2, no. 8, Aug. 1962, pp. 249–256; also: Proc. SESA, vol. XIX, no. 2, pp. 249–256.

Cheng, S.: On the Theory of Bending of Sandwich Plates. Proc. 4th U.S. Natl. Congr. Appl. Mech., vol. I, pp. 511–518.

Chu, H.-N.: High-Frequency Extensional Vibrations of Sandwich Plates. J. Acoust. Soc. Am., vol. 34, no. 9, pt. 1, Sept. 1962, pp. 1184–1190.

CUNNINGHAM, J. H.; AND JACOBSON, M. J.: Design and Testing of Honeycomb Sandwich Cylinders Under Axial Compression. Engineering Paper no. 1393 Missile and Space Division, Douglas Aircraft Co., Aug. 1962; also in NASA TN D-1510, Dec. 1962.

CUNNINGHAM, J. H.; AND JACOBSON, M. J.: Design and Testing of Honeycomb Sandwich Cylinders Under Axial Compression. *In:* Collected Papers on Instability of Shell structures—1962—NASA TN D-1510, pp. 341–359.

EBCIOGLU, I. K.: Thermo-Elastic Equations for a Sandwich Panel Under Arbitrary Temperature Distribution, Transverse Load, and Edge Compression. Proc. 4th U.S. Natl. Congr. Appl. Mech., vol. I, pp. 537–546.

GALLAGHER, R. H.; RATTINGER, I.; AND KRIVETSKY, A.: Minimum Weight Shells in Bending. Aerospace Eng., vol. 21, no 2, Feb. 1962, pp. 58–59, 64, 68, 72, 74–82.

GIENCKE, E.: Ueber das Knittern von Laminaten aus glasfaserverstarktem Kunststoff. Wiss. Ges. Luftf., Rept. 4/1962, p. 67–76.

GOREE, W. S.; AND NASH, W. A.: Elastic Stability of Circular Cylindrical Shells Stabilized by a Soft Elastic Core. Exptl. Mech., vol. 2, no. 5, May 1962, pp. 142–149; also: Proc. SESA, vol. XIX, no. 1, pp. 142–149.

HARRIS, L. A.; AND BAKER, E. H.: Elastic Stability of Simply-Supported Corrugated-Core Sandwich Cylinders. *In:* Collected Papers on Instability of Shell Structures—1962—NASA TN D-1510, pp. 331–340.

HOWARD, H. B.: The Five-Point Loading Shear Stiffness Test. J. Roy. Aeron. Soc., vol. 66, no. 621, Sept. 1962, p. 591.

JAMES, W. L.: Calculation of Vibration Damping in Sandwich Construction From Damping Properties of the Cores and Facings. FPL Rept. 1888.

JENKINSON, P. M.; AND KUENZI, E. W.: Effect of Core Thickness on Shear Properties of Aluminum Honeycomb Core. FPL Rept. 1886.

KIMEL, W. R.; AND RAVILLE, M. E.: Buckling of Sandwich Panels in Edgewise Bending and Compression. Proc. 4th U.S. Natl. Congr. Appl. Mech., vol. I, pp. 657–666.

KUENZI, E. W.: Buckling of Layered Orthotropic and Sandwich Cylindrical Shells in Axial Compression. *In:* Collected Papers on Instability of Shell Structures—1962—NASA TN D-1510, pp. 323–330.

KUENZI, E. W.; ERICKSEN, W. S.; AND ZAHN, J. J.: Shear Stability of Flat Panels of Sandwich Construction. FPL Rept. 1560 (Rev. May 1962).

KURANISHI, M.: On Several Research Problems of the Instability of Shell Structures. *In:* Collected Papers on Instability of Shell Structures—1962—NASA TN D-1510, pp. 693–719.

MOSELEY, W. M.: Problems Associated With the Design of Large Shell Structure. *In:* Collected Papers on Instability of Shell Structures—1962—NASA TN D-1510, pp. 67–76.

NIEUWENHUIZEN, M. P.: Optimum Sandwich Design. Prepr. 5th European Aeron. Congr. (Venice), Sept. 1962.

PRETLOVE, A. J.: On the Vibrations of Cylindrically Curved Elastic Sandwich Plates: Pt. II: The Solution for Cylindrical Plates. Univ. Southampton, AASU Rept. 187.

SEIDE, P.: The Stability Under Axial Compression and Lateral Pressure of Circular-Cylindrical Shells With a Soft Elastic Core. J. Aerospace Sci., vol. 29, no. 7, July 1962, pp. 851–862. Also see Space Technology Labs. EM 11-3, STL/TR-61-0000-19015, 1961.

STEVENS, G. H.: Compressive and Shear Properties of Two Configurations of Sandwich Cores of Corrugated Foil. FPL Rept. 1889.

STEVENS, G. H.; AND KUENZI, E. W.: Mechanical Properties of Several Honeycomb Cores. FPL Rept. 1887.

SYLVESTER, R. J.: Buckling of Sandwich Cylinders Under Axial Loads. J. Aerospace Sci., vol. 29, no. 7, July 1962, pp. 863–872.

WEINGARTEN, V. I.: Stability Under Torsion of Circular Cylindrical Shells With an Elastic Core. ARS J., vol. 32, no. 4, Apr. 1962, pp. 637–639.

WIEDEMANN, J.: Die beulkritische Last der Sandwichplatte mit orthotropen Kern und orthotropen Deckhäuten unterschiedlicher Stärke bei allseitig gelenkiger Randstützung. Luftfahrttechnik, vol. 8, no. 6, June 1962, pp. 150–158.

YAO, J. C.: Buckling of Axially Compressed Long Cylindrical Shell With Elastic Core. J. Appl. Mech., vol. 29, no. E2, June 1962, pp. 329–334.

YAO, J. C.: Buckling of Sandwich Sphere Under Normal Pressure. J. Aerospace Sci., vol. 29, no. 3, Mar. 1962, pp. 264–268, 305.

YU, Y.-Y.: Damping of Flexural Vibrations of Sandwich Plates. J. Aerospace Sci., vol. 29, no. 7, July 1962, pp. 790–803.

YU, Y.-Y.: Effect of Thickness Deformations on Vibrations of Sandwich Plates. Air Force Office of Scientific Research, TN 2255, Feb. 1962.

YU, Y.-Y.: Extensional Vibrations of Elastic Sandwich Plates. Proc. 4th U.S. Natl. Congr. Appl. Mech., vol. I, pp. 441–447.

YU, Y.-Y.: Nonlinear Flexural Vibrations of Sandwich Plates. J. Acoust. Soc. Am., vol. 34, no. 9, pt. 1, Sept. 1962, pp. 1176–1183.

ZAK, A. R.; AND BOLLARD, R. J. H.: Elastic Buckling of Cylindrical Thin Shells Filled With an Elastic Core. ARS J., vol. 32, no. 4, Apr. 1962, pp. 588–592.

1963

ALMROTH, B. O.; AND BRUSH, D. O.: Postbuckling Behavior of Pressure—or Core-Stabilized Cylinders Under Axial Compression. AIAA J., vol. 1, no. 10, Oct. 1963, pp. 2338–2341.

BRUNN, E. R.: Thermal Deflection of a Circular Sandwich Plate. AIAA J., vol. 1, no. 5, May 1963, pp. 1213–1215.

CLARY, R. R.; AND LEADBETTER, S. A.: An Analytical and Experimental Investigation of the Natural Frequencies of Uniform Rectangular Cross-Section Free-Free Sandwich Beams. NASA TN D–1967.

COHEN, G. A.: Optimum Design of Truss-Core Sandwich Cylinders Under Axial Compression. AIAA J., vol. 1, no. 7, July 1963, pp. 1626–1630.

CRAWFORD, R. F.; AND BURNS, A. B.: Minimum Weight Potentials for Stiffened Plates and Shells. AIAA J., vol. 1, no. 4, Apr. 1963, pp. 879–886.

GIENCKE, E.: Die Hohlplatte, eine schubweiche Platte. ZAMM, vol. 43, no. 7/8, 1963, pp. 311–324.

GRIGOLYUK, E. I.; AND KIRYUKHIN, Y. P.: Linear Theory of Three-Layered Shells With a Soft Core. AIAA J., vol. 1, no. 10, Oct. 1963, pp. 2438–2445.

HESS, T. E.: Analysis of Honeycomb Shells. PIR SM–8156–561, Missile and Space Div., General Electric Co., Dec. 1963.

LEMKE, D. G.: Buckling of a Composite Orthotropic Cylinder Containing an Elastic Foundation. AIAA Summer Meeting (Los Angeles, Calif.), June 17–20, 1963. Preprint 63–233.

MCFARLAND, R. K., JR.: Hexagonal Cell Structures Under Post-Buckling Axial Load. AIAA J., vol. 1, no. 6, June 1963, pp. 1380–1385.

MEAD, D. J.; AND FROUD, G. R.: The Damping of Aluminum Honeycomb Sandwich Beams. Univ. Southampton, AASU, Summary Rept.

NIEUWENHUIZEN, M. P.: Strength of Plastic Sandwich Construction. Preprint Symp. Sandwich Constr. (London), Nov. 1963. Also see The Plastics Inst. Trans. and J., no. 100, Aug. 1964, pp. 272–285.

NORRIS, C. B.; AND ZAHN, J. J.: Design Curves for the Buckling of Sandwich Cylinders of Finite Length Under Uniform External Lateral Pressure. U.S. Forest Service, Res. Note FPL–07.

OPPEL, G. U.; AND MEHTA, P. K.: Stress Analysis of Ring-Stiffened Sandwich Shells Subjected to Uniform External Pressure. Intern. J. Mech. Sci., vol. 5, no. 4, Aug. 1963, pp. 335–352.

SCHMIDT, R.: Sandwich Shells of Arbitrary Shape. ASME Preprint 63–WA–61. Also see J. Appl. Mech., vol. 31, no. 2, June 1964, pp. 239–244.

SWITZKY, H.; AND CARY, J. W.: The Minimum Weight Design of Cylindrical Structures. AIAA J., vol. 1, no. 10, Oct. 1963, pp. 2330–2337.

THOMPSON, W. M., JR.; AND CLARY, R. R.: An Investigation of the Natural Frequencies and Mode Shapes of Double Conical Sandwich Shells. NASA TN D–1940.

VAN DYKE, J. D., JR.: An Investigation of Structural Concepts for a Mach 3+ Supersonic Transport Wing. SAE Preprint 751–A.

ZAHN, J. J.; AND KUENZI, E. W.: Classical Buckling of Cylinders of Sandwich Construction in Axial Compression—Orthotropic Cores. U.S. Forest Service, Res. Note FPL–018.

1964

ALGRA, H.; AND TAAL, L.: Sandwichconstructies Met Hard PVC Kern. Plastica, vol. 17, no. 9, Sept. 1964, pp. 429–438.

ALWAN, A. M.: Large Deflection of Sandwich Plates With Orthotropic Cores. AIAA J., vol. 2, no. 10, Oct. 1964, pp. 1820–1822.

ANDERSON, R. A.: Structures Technology—1964. Astronaut. Aeron., vol. 2, no. 12, Dec. 1964, pp. 14–20.

ANON.: Mechanical Properties of Hexcel Honeycomb Materials. Hexcel Products, Inc., TSB 120.

BAKER, E. H.: Analysis of a Symmetrically Loaded Sandwich Cylinder. AIAA J., vol. 2, no. 1, Jan. 1964, pp. 108–109.

BAKER, E. H.: Stability of Circumferentially Corrugated Sandwich Cylinders Under Combined Loads. AIAA J., vol. 2, no. 12, Dec. 1964, pp. 2142–2149.

BRINK, N. O.: Sandwich Cylinder Construction for Underwater Pressure Vessels. Mater. Res. and Std., vol. 4, no. 4, Apr. 1964, pp. 262–164.

DAVIDSON, O. C.: An Axially Compressed Cylindrical Shell With a Viscoelastic Core. AIAA J., vol. 2, no. 11, Nov. 1964, pp. 2015–2016.

GELLATLY, R. A.: Comment on: Effective Shear Modulus of Honeycomb Cellular Structure. AIAA J., vol. 2, no. 8, Aug. 1964, pp. 1518–1519.

GELLATLY, R. A.; AND GALLAGHER, R. H.: Sandwich Cylinder Instability Under Nonuniform Axial Stress. AIAA J., vol. 2, no. 2, Feb. 1964, pp. 398–400.

GELLATLY, R. A.; AND GALLAGHER, R. H.: Stresses in Sandwich Cylinders. Machine Design, vol. 36, no. 8, Mar. 1964, pp. 197–201.

GRIGOLYUK, E. I.: General Large-Deflection Theory of Elastic Sandwich Shallow Shells. Arch. Mech. Stos., vol. 16, no. 1, pp. 123–133.

HABIP, L. M.: A Review of Recent Russian Work on Sandwich Construction. Intern. J. Mech. Sci., vol. 6, no. 6, Dec. 1964, pp. 483–487.

KUENZI, E. W.; NORRIS, C. B.; AND JENKINSON, P. M.: Buckling Coefficients for Simply Supported and Clamped, Flat, Rectangular Sandwich Panels Under Edgewise Compression. U.S. Forest Service, Res. Note FPL–070.

MEAD, D. J.; AND PRETLOVE, A. J.: On the Vibrations of Cylindrically Curved Elastic Sandwich Plates.

NORRIS, C. B.: Short-Column Compressive Strength of Sandwich Constructions as Affected by Size of Cells of Honeycomb Core Materials. U.S. Forest Service, Res. Note FPL–026.

PENZIEN, J.; AND DIDRIKSSON, T.: Effective Shear Modulus of Honeycomb Cellular Structure. AIAA J., vol. 2, no. 3, Mar. 1964, pp. 531–535.

ZAHN, J. J.; AND CHENG, S.: Edgewise Compressive Buckling of Flat Sandwich Panels: Loaded Ends Simply Supported and Sides Supported by Beams. U.S. Forest Service, Res. Note FPL–019.

1965

COX, H. L.; AND MARTIN, D. W.: Deformation of Sandwich Structure. J. Roy. Aeron. Soc., vol. 69, no. 651, Mar. 1965, pp. 193–197.

DE JONGE, J. B.; AND PLANTEMA, F. J.: Buckling of Rectangular Sandwich Plates Under Biaxial Compression. NLR Rept S. 641. (To be published.)

EBCIOGLU, I. K.: On a Theory of Sandwich Panels in the Reference State. Intern. J. Eng. Sci., vol. 2, no. 6, Mar. 1965, pp. 549–564.

FULTON, R. E.: Effect of Face Sheet Stiffness on Buckling of Curved Plates and Cylindrical Shells of Sandwich Construction in Axial Compression. NASA TN D–2783.

GELLATLY, R. A.; BIJLAARD, P. P.; AND GALLAGHER, R. H.: Thermal Stresses and Instability of Sandwich Cylinders on Rigid Supports. J. of Aircraft, vol. 2, no. 1, Jan.–Feb. 1965, pp. 44–48.

HABIP, L. M.: A Survey of Modern Developments in the Analysis of Sandwich Structure. Appl. Mech. Rev., vol. 18, no. 2, Feb. 1965, pp. 93–98.

HOFF, N. J.: Quelques Nouveux Résultats de Recherches sur le Flambage des Coques Cylindriques. Dept. Aeronautics and Astronautics, Stanford Univ., May 1965.

KUENZI, E. W.: Minimum Weight Structural Sandwich. U.S. Forest Service, Res. Note FPL–086.

PETERSON, J. P.; AND ANDERSON, J. K.: Structural Behavior and Buckling Strength of Honeycomb Sandwich Cylinders Subjected to Bending. NASA TN D–2926.

PLANTEMA, F. J.; AND SEVENHUYSEN, P. J.: Bending of Orthotropic Plates Under Uniform Transverse Loading, and the Analogy With Beams on an Elastic Layer. NLR Rept. MP 234. (To be published.)

WEMPNER, G. A.: Theory for Moderately Large Deflections of Thin Sandwich Shells. J. Appl. Mech., vol. 32, no. 1, Mar. 1965, pp. 76–80.

YAO, J. C.: Bending Due to Ring Loading of a Cylindrical Shell With an Elastic Core. J. Appl. Mech., vol. 32, no. 1, Mar. 1965, pp. 99–103.

FILAMENTARY STRUCTURES

1933

GOODIER, J. N.: Concentration of Stress Around Spherical and Cylindrical Inclusions and Flaws. J. Appl. Mech. (Trans. ASME), vol. 55, 1933, pp. 39–44.

1939

WEIBULL, W.: The Phenomenon of Rupture in Solids. Ing. Vetenskaps Akad., Handl., NR 153, 1939, pp. 1–55.

WESTERGAARD, H. M.: Bearing Pressures and Cracks. J. Appl. Mech. (Trans. ASME), 1939, pp. A49–A53.

1940

GENSAMER, M.; PEARSALL, E. B.; AND SMITH G. V.: The Mechanical Properties of the Isothermal Decomposition Products of Austenite. Trans. ASME, vol. 28, 1940, p. 380.

1947

DEWEY, J. M.: The Elastic Constants of Materials Loaded With Non-Rigid Fillers. J. App. Phys., vol. 18, 1947, p. 578.

SADOWSKY, M. A.; AND STERNBERG, E.: Stress Concentration Around an Ellipsoidal Cavity in an Infinite Body Under Arbitrary Plane Stress Perpendicular to the Axis of Revolution of Cavity. Trans. ASME, vol. 69, 1947, pp. 191–201.

SANDFORD, E. J.; AND TRENT, E. M.: The Physical Metallurgy of Sintered Carbides. Symp. Powder Metallurgy, Spec. Rept. No. 38, The Iron & Steel Institute, 1947, p. 84.

1948

ENGLE, E. W.: Cemented Carbides. Powder Metallurgy, J. Wulff, ed., 1942, p. 436.

Epstein, B.: J. App. Phys., vol. 19, 1948, p. 140.

OROWAN, E.: Fracture and Strength of Solids. Rept. on Progress in Physics, vol. 12 (1948–49), p. 185.

1949

DIETZ, A. G. H., ED.: Engineering Laminates. John Wiley & Sons, Inc., 1949.

SADOWSKY, M. A.; AND STERNBERG, E.: Stress Concentrations Around a Triaxial Elliptical Cavity. Trans. ASME, vol. 71, 1949, pp. 149–157.

WERREN, F.; AND NORRIS, C. B.: Directional Properties of Glass-Fabric-Base Plastic Laminate Panels of Sizes That Do Not Buckle. Forest Prods. Lab. Rept. 1803, 1949.

1950

MACKENZIE, J. K.: The Elastic Constants of a Solid Containing Spherical Holes. Proc. Phys. Soc. (London), vol. 63, no. 2, 1950.

1951

EDWARDS, R. H.: Stress Concentrations Around Spheroidal Inclusions and Cavities. J. Appl. Mech., 1951, pp. 19–30.

ESHELBY, J. D.: The Force on an Elastic Singularity. Phil. Trans. Roy. Soc. (London), (A) 244, 1951, p. 87.

1952

GURLAND, J.; AND NORTON, J. T.: Role of the Binder Phase in Cemented Tungsten Carbide-Cobalt Alloys. Trans. ASME, vol. 194, 1952, p. 1051.

1953

DUCKWORTH, W.: Discussion of Paper by E. Ryshkewitch, Compression Strength of Porous Sintered Alumina and Zirconia. J. Am. Ceram. Soc., vol. 36, no. 65, 1953.

HOLLINGER, D. L.; PLANT, H. T.; MULVEY, B. F.; AND JORDAN, T. J.: High Strength Glass Fibers Development Program, Task 2. General Electric Corp., Evendale, Ohio, Final Rept., Contract NOw 61 0641-c (FBM), May 1953.

SHAW, R. B.; SHEPPARD, L. A.; STARR, C. D.; AND DORN, J. E.: The Effect of Dispersions on the Tensile Properties of Aluminum-Copper Alloys. Trans. ASME, vol. 45, 1953, p. 249.

1955

BROWN, W. F., JR.: Solid Mixture Permittivities. J. Chem. Phys., vol. 23, 1955, p. 1514.

COBLE, R. L.; AND KINGERY, W. D.: Effect of Porosity on Thermal Stress Fracture. J. Am. Ceram. Soc., vol. 38, no. 33, 1955.

GURLAND, J.; AND BARDIL, P.: Relation of Strength, Composition, and Grain Size of Sintered WC–Co Alloys. Trans. AIME., J. Metals, 1955, pp. 311–315.

HASHIN, Z.: The Moduli of an Elastic Solid Reinforced by Rigid Particles. Bull. Res. Council Israel, vol. 5C, 1955, p. 46.

1956

DIETZ, A. G. H.: Design Theory of Reinforced Plastics. *In:* Fiberglass Reinforced Plastics, R. H. Sonneborn, ed., Reinhold Publ. Corp., 1956.

EKVALL, J. C.: Elastic Properties of Orthotropic Monofilament Laminates. ASME Paper 61–AV–56, 1956.

GURLAND, J.; AND NORTON, J. T.: Plansee Proceedings, 1955. Pergamon Press (London), 1956, p. 99.

OLDROYD, J. G.: The Effect of Small Viscous Inclusions on the Mechanical Properties of an Elastic Solid. *In:* Deformation and Flow of Solids, Grammel, ed., Springer-Verlag (Berlin), 1956, p. 304.

1957

CUTLER, I. B.: Strength Properties of Sintered Alumina in Relation to Porosity and Grain Size. J. Am. Ceram. Soc., vol. 40, no. 20, 1957.

ESHELBY, J. D.: The Determination of the Elastic Field of an Ellipsoidal Inclusion and Related Problems. Proc. Roy. Soc. (London), (A) 241, 1957, p. 376.

1958

BRENNER, S. S.: Growth and Properties of Whiskers. Science, vol. 128, Sept. 12, 1958.

COLEMAN, B. D.: J. Mech. Phys. Solids, vol. 7, 1958, p. 60.

DOREMUS, R. H.; ROBERTS, B. W.; AND TURNBULL, D.: Growth and Perfection of Crystals. John Wiley & Sons, Inc., 1958.

GURLAND, J.: Trans. Am. Soc. Metals, vol. 50, 1958, p. 1064.

LAZAN, B. J.; AND ANDERSON, V. W.: Damping and Fatigue Properties of Plastic Materials. Univ. of Minnesota, 1958.

NORRIS, C. B.; AND HELLER, J. T.: Fiber-Resin Bond in Reinforced Plastics. WADS Tech. Rept. 58–356, Oct. 1958.

STEVENS, G. H.; AND BOLLER, K. H.: Effect of Type of Reinforcement on Fatigue Properties of Plastic Laminates. Forest Products Lab., WADC 59–27, 1958.

YOUNG, R. E.: History and Potential of Filament Winding. 13th Ann. Technical and Management Conf., Reinforced Plastics Div., Society of the Plastics Industry (Chicago), Feb. 4, 1958.

1959

ANON.: Composite Construction for Flight Vehicles. Pts I and II, MIL–HDBK–23, Government Printing Office, Washington, D.C., 1959.

HASHIN, Z.: The Moduli of an Elastic Solid, Containing Spherical Particles of Another Elastic Material. Proc. IUTAM Symp. on Non-Homogeneity in Elasticity and Plasticity (Warsaw, Poland), Pergamon Press, Inc., 1959, p. 463.

HOFFMAN, G. A.: Future Possibilities in Fibered Materials. Proc. 6th Sagamore Conf. on Composite Materials and Composite Structures (Racquette Lake, N.Y.), 1959.

KITZMILLER, A. H., JR.; DE HAVEN, C. C.; AND YOUNG, R. E.: Design Consideration for Spiralloy Glass-Reinforced Filament Wound Structures as Rocket Inert Parts. Am. Rocket Soc., Prepr. 983–59, 1959.

KNUDSEN, F. P.: Dependence of Mechanical Strength of Brittle Polycrystalline Specimens on Porosity and Grain Size. J. Am. Ceram. Soc., vol. 42, 1959, p. 376.

KRIVOGLAZ, M. A.; AND CHEREVKO, A. S.: On the Elastic Moduli of a Two-Phase Solid. Fis. Metal, Metalloved., vol. 8, 1959, p. 161.

MCA–MIT Progress Report. Resin Shrinkage Pressures During Cure. Jan. 15, 1959.

MCA–MIT Progress Report. Resin-Glass Bond Study. Feb. 15, 1959.

MCGARRY, F. J.: Resin-Glass Bond Characteristics. ASTM Bull. 235, Jan. 1959, p. 63.

MCGARRY, F. J.; AND DESAI, M. B.: Failure Mechanisms in Glass-Fiber Reinforced Plastics. ASTM Bull. 239, June 1959.

MCGUIRE, F. G.: Fiberglass Wrapping Lightens Atlas. Missiles Rockets, vol. 7, no. 18, Dec. 28, 1959.

OTTO, W. H.: Properties of Glass Fibers at Elevated Temperatures. O.C.F. Res. Centr.—Final Report on Contract NOas–58–841–C, to the Bur. of Aeron.—U.S. Navy, 1959.

SCHUERCH, H. U.: Preliminary Study of Application of Advanced Structural Materials to Space Vehicles and Boost Systems. General Electric—TEMPO, RM59TMP–9, Jan. 31, 1959.

WHITEHURST, H. B.; MICHENER, J. W.; AND LOCKWOOD, P. A.: Investigation of Glass-Metal Composites. Proc. 6th Sagamore Conf. Comp. Mats. and Comp. Strucs. (Racquette Lake, N.Y.), 1959.

1960

BRENNER, S. S.: Considerations of Whisker Reinforced Metals. First Quarterly Progress Report, G.E. Space Sciences Lab. Contract Now–60–0465–d, 1960.

CUMMINGS, H. M.: Some Quantitative Aspects of Fatigue of Materials. Curtiss-Wright Corp., WADD Rept. 60–42, 1960.

HOFFMAN, G. A.: The Effect of Filamentary Materials on Pressure Vessel Design. Intern. Astronaut. Congr. (Stockholm, Sweden), Aug. 1960.

MATLACH, J. D.; MCABEE, E.; AND CHMURA, M.: Effect of Rapid Stressing on the Tensile Properties of Glass Reinforced Plastics. 2d Intern. Reinforced Plastics Conf. (London), 1960.

MCDANELS, D. L.; JECH, R. W.; AND WEETON, J. W.: Metal Progr., vol. 78, 1960, p. 118.

NICHIMATSU, C.; AND GURLAND, J.: Trans. Am. Soc. Metals, vol. 52, 1960, p. 469.

PARRATT, N. J.: Defects in Glass Fibers and Their Effect on Strength of Plastic Mouldings. Rubber Plastics Age, vol. 41, no. 3, 1960.

PAUL, B.: Prediction of Elastic Constants of Multi-Phase Materials. Am. Inst. Mech. Engrs., Trans. Met. Soc., vol. 218, 1960, pp. 1017–1022.

273-140 O - 67 - 11

RILEY, M. W.; et al.: Filament Wound Reinforced Plastics: State of the Art. Reprint from Materials in Design Engineering, Aug. 9, 1960.

SCHUERCH, H. U.: Space Structure Design With Composite Materials. Astro Research Corp., ARS Preprint 1096-60, Apr. 1960.

THOMAS, W. F.: Investigation of the Factors Likely to Affect the Strength and Properties of Glass Fibers. Phys. Chem. Glasses, vol. 1, no. 1, 1960.

THORKILDSEN, R. F.: Design Data for Circular Reinforced Plastic Plates. Proc. 15th Ann. Conf., Ref. Plas. Div., SPI, 1960.

1961

ANON.: Filament Winding Goes Commercial. Mod. Plastics, vol. 39, no. 2, Oct. 1961, p. 94.

BELL, J. E.: The Effect of Glass Fiber Geometry on Composite Material Strength. ARS J., vol. 31, 1961, pp. 1260-1264.

BERSTEIN, H.; AND KIES, J. A.: The Fiberglass Motor Case in the Polaris Program. SAMPE Mtg. Rept., Mar. 1961, p. 273.

BIKERMAN, J. J.: The Science of Adhesive Joints. Academic Press (New York), 1961.

BRADLEY, H. B.; AND STERMAN, S.: A New Interpretation of the Glass-Coupling Agent Surface Through Use of Electron Microscopy. Proc. 16th Ann. Conf., Ref. Plas. Div., SPI (Chicago), 1961.

BUHL, J. E.; AND PULOS, J. G.: Reinforced Plastics for Hydrospace Vehicles. SAMPE Mtg. Rept., Mar. 1961, p. 325.

CHARLES, R. J.; AND HILLIG, W. B.: The Kinetics of Glass Failure by Stress Corrosion. G.E. Res. Lab. Rept. 61-RL-2790M, 1961. Also see Charles, R. J., J. App. Phys., vol. 29, no. 11, 1958, pp. 1554-1560.

DALLAS, R. N.: Filament Winding a Lockheed Star Fighter Radome. SAMPE Mtg. Rept., Mar. 1961, p. 225.

DEAN, W. J.: The Effect of Some Primary Variables on the Strength of Glass Reinforced Plastics for Use on Space Vehicles. Proc. 16th Ann. Conf., Ref. Plas. Div., SPI, 1961.

EAKINS, W. J.: A Study of the Degree of Bonding Between Coupling Agents and Silica, Type E and High Modulus Glass. SAMPE Preprint, Mar. 1961, p. 1.

ESHELBY, J. D.: Elastic Inclusions and Inhomogeneities. Progress in Solid Mechanics, II, Sneddon and Hill, eds., North-Holland (Amsterdam), 1961.

HASHIN, Z.; AND SHTRIKMAN, S.: On Some Variational Principles in Elasticity and Their Application to the Theory of Two-Phase Materials. Univ. of Penn., Contract NONR 551(42), TR no. 1, 1961.

HEDGEPETH, J. M.: Stress Concentrations in Filamentary Structures. NASA TN D-882, 1961.

LAYTON, P.: Review of Various Glass Filaments. SAMPE Preprint Mar. 1961, p. 197.

MCABEE, E.; AND CHMURA, M.: Effects of High Rates Compared With Static Rates of Loading on the Mechanical Properties of Glass Reinforced Plastics. Proc. 16th Ann. Conf., Ref. Plas. Div., SPI, 1961.

OTTO, W. H.: Compaction Effects in Glass Fibers. J. Am. Ceram. Soc., vol. 44, no. 2, 1961.

OTTO, W. H.: Properties of Glass Fibers at Elevated Temperatures. SAMPE Mtg. Rept., Mar. 1961, p. 32.

PETERSON, G. P.: Optimum Filament-Wound Plastic Composites. SAMPE Mtg. Rept., Mar. 1961, p. 316.

Proc. Filament Winding Conf., SAMPE, Mar. 28-30, 1961.

REISSNER, E.; AND STAVSKY, Y.: Bending and Stretching of Certain Types of Heterogeneous Aeolotropic Elastic Plates. J. App. Mech., vol. 28, 1961.

RUMBLES, W. E.: Properties of Fine Wire for Use in Filament-Wound Vehicles. SAMPE Mtg. Rept., Mar. 1961, p. 122.

SADOWSKY, M. A.: Transfer of Force by High Strength Flakes in a Composite Material. Watervliet Arsenal TRWVT RR 6015–R, June 1961.

SCHUERCH, H. U.; AND KYSER, A. C.: Isotensoid Torus Design—Final Report. Astro Research Corp., Contract NAS 1–889, Feb. 16, 1961.

1962

BOLLER, K. H.: Fatigue Properties of Plastic Laminates Reinforced With Unwoven Glass Fibers. Forest Products Lab., ASD TDR 62–464, 1962.

BRENNER, S. S.: Mechanical Behavior of Sapphire Whiskers at Elevated Temperatures. J. App. Phys., vol. 33, no. 1, 1962, pp. 33–39.

BREWER, G. D.: Liquid vs. Solid Fuels. Ind. Res., vol. 4, no. 4, Apr. 1962, p. 41.

EDELSON, B. I.; AND BALDWIN, W. M., JR.: Trans. Am. Soc. Metals, vol. 55, 1962, p. 230.

FRIED, N.; AND WINANS, R. R.: Research and Development Report on Reinforced Plastics for Deep Submergence and Other High Strength Applications—Compressive Strength. Lab Project 6189, Progress Report No. 3, Materials Development Branch, Materials Laboratory, New York Naval Shipyard, Mar. 8, 1962.

GUCER, D. E.; AND GURLAND, J.: J. Mech. Phys. Solids, vol. 10, 1962, p. 365.

HASHIN, Z.: The Elastic Moduli of Heterogeneous Materials. J. Appl. Mech., vol. 29. Trans. ASME, vol. 84, 1962, pp. 143–150.

HASHIN, Z.; AND SHTRIKMAN, S.: On Some Variational Principles in Anisotropic and Non-Homogeneous Elasticity. J. Mech. Phys. Solids, vol. 10, 1962, p. 334.

HILL, R.: Report on Theories of the Elastic Properties of Reinforced Solids. British Iron & Steel Res. Assoc. Rept. P/19/62, 1962.

JUDGE, J. F.: Lamtex Constructing Biggest Reinforced Case. Missiles Rockets, vol. 10, no. 22, Jan. 15, 1962.

KIES, J. A.: Maximum Strains in Resin of Fiberglass Composites. Polaris Res. and Develop Conf. (Palo Alto, Calif.), 1962.

PRAGER, S.: Viscosity of Concentrated Suspensions of Rigid Particles. Trans. Soc. Rheol., vol. VI, 1962, p. 393.

READ, W. S.: Equilibrium Shapes for Pressurized Fiberglass Domes. ASME Paper 62–AV–4, 1962.

REGESTER, R.; ET AL.: Development of Composite Structural Materials for High Temperature Applications. G.E. Space Sciences Lab. 8th Quart. Rept., Contract NOw–60–0465d, 1962.

ROSEN, B. W.; ET AL.: Hollow Glass Fiber Reinforced Plastics. OTS Rept. AD–400237, U.S. Dept. Commerce (Washington, D.C.), 1962.

ROSEN, B. W.; ET AL.: Hollow Glass Fiber Reinforced Plastics. General Electric Corp., Philadelphia, Final Rept., U.S. Navy Bureau of Naval Weapons, Contract NOw 610613–d, Nov. 1962.

SCHMIDT, D. L.; AND JONES, W. C.: Carbon-Base Fiber Reinforced Plastics. ASD TR No. ASD–TDR–62–635, Aero, Sys. Div., WPAFB, Ohio, 1962.

SUTTON, W. H.: Development of Composite Structural Materials for Space Vehicle Applications. Am. Rocket Soc. J., 1962.

TSAI, S. W.: Composite Stiffness of Fiber-Reinforced Media. Aeronutronic Publ. U–1699, 1962.

TSAI, S. W.: A Variational Formulation of Two-Dimensional Heterogeneous Media. Aeronutronic Publ. U–1698, 1962.

WADD: Growth and Mechanical Properties of Filamentary Silicon Carbide Crystals. TR no. WADD TR 61-252, 1962.

WATERS, F., JR.: Winding Mechanics for Filament Wound Pressure Vessels. Allison TDR Bx 0240-007, July 1962.

ZISMAN, W. A.: Influence of Constitution of Liquids and Solids on Their Adhesion. Navy Tech. Forum, U.S. Government Printing Office (Washington, D.C.), 1962.

1963

ANDERSON, J. A.; AND McCARTHY, J. A.: Prepreg Reinforced Plastics in Fatigue Applications. Minnesota Mining & Mfg. Co., 1963.

ANON.: Filament Winding Advances. Mod. Plastics, vol. 40, no. 6, Feb. 1963, pp. 95-97.

ANON.: Filament Winding Box Shapes. Iron Age, vol. 191, Apr. 11, 1963, pp. 76-77.

ANON.: Fiberglass Invades Tank Car Market. Iron Age, vol. 191, Apr. 18, 1963, p. 79.

CLARK, W. J.; ET AL.: Evaluation of High-Strength-and-Modulus Brittle Materials in Filament and Composite Forms. ASTM, STP, 327, 1963.

CLARK, H. A.; AND PLUEDDEMANN, E. P.: Bonding of Silane Coupling Agents With Resin and With Glass in Glass-Reinforced Plastics. Proc. 18th Ann. Conf., Ref. Plas. Div., SPI (Chicago), 1963.

CRATCHLEY, D.: Factors Affecting the UTS of a Metal/Metal-Fibre Reinforced System. Powder Met., no. 11, 1963.

DOW, N. F.: Study of Stresses Near a Discontinuity in a Filament Reinforced Composite Metal. General Electric Corp., Philadelphia, TIS R635D61, Aug. 1963.

EAKINS, W. J.: A Study of Interlaminar Shear Failure in Single Filament Wound Rings Made at the Bushing. Proc. 18th Ann. Conf., Ref. Plas. Div., SPI, 1963.

Editors, Materials in Design Engineering: The Promise of Composites. MDE Spec. Rept. no. 210, 1963, pp. 79-126.

FRIED, N.: The Compressive Strength of Parallel Filament Reinforced Plastics—The Role of the Resin. Proc. 18th Ann. Conf., Ref. Plas. Div., SPI, 1963.

FRIED, N.: Survey of Methods of Test for Parallel Filament Reinforced Plastics. ASTM Spec. Tech. Publ. 327, 1963, pp. 13-39.

GUCER, D. E.; AND GURLAND, J.: Jernkontorets Annaler, vol. 147, 1963, p. 111.

GURLAND, J.: Trans. Met. Soc. AIME, vol. 227, 1963, p. 1146.

GURLAND, J.; AND PLATEAU, J.: Trans. Am. Soc. Metals, vol. 56, 1963, p. 442.

HANNOOSH, M. M.; AND MULDOON, R. A.: Trends in Radome Design Parameters. Proc. 18th SPI Conf. (Chicago, Ill.), 1963.

HARRINGTON, R. A.; ET AL.: A Research Program To Obtain Design Information on Rocket Cases From Analytical and Experimental Studies on Cylinders Subjected to Combined Loadings. B. F. Goodrich Research Center (Brecksville, Ohio), 1963-64.

HARTUNG, R. F.: Planar-Wound Filamentary Pressure Vessels. AIAA J., vol. 1, no. 12, Dec. 1963, pp. 2842-2844.

HASHIN, Z.; AND SHTRIKMAN, S. A.: A Variational Approach to the Theory of the Elastic Behavior of Multiphase Materials. J. Mech. Phys. Solids, vol. 11, 1963, p. 127.

HILL, R.: Elastic Properties of Reinforced Solids: Some Theoretical Principles. J. Mech. Phys. Solids, vol. 11, no. 5, 1963, pp. 357-372.

HOFEDITZ, J. T.: Structural Design Considerations for Fiber Glass Pressure Vessels. Proc. 18th SPI Conf. (Chicago, Ill.), 1963.

IRVING, R. R.: Filament Winding Goes After Commercial Markets. Iron Age, vol. 192, Nov. 14, 1963, pp. 159-161.

KELLY, A.; AND NICKOLSON, R. B.: Progress in Materials Science, vol. 10, 1963, p. 149.

LEKHNITSKIY, S. G.: Theory of Elasticity of an Anisotropic Body. Holden-Day, Inc. (San Francisco, Calif.), 1963.

LINDSAY, E. M.; AND HOOD, J. C.: Glass Reinforcements for Filament Wound Composites. WPAFB, Ohio, Mfg. Technical Div., 1963.

LOVE, G. G.: Methods of Stress Analysis of Filament Wound Pressure Vessels. Allison EDR 2784, May 1963.

McDANELS, D. L.; JECH, R. W.; AND WEETON, J. W.: NASA TN D-1881, 1963.

McGARRY, F. J.; AND MARSHALL, D. W.: Research on Wire-Wound Composites Materials. *In:* Standards for Filament-Wound Reinforced Plastics. ASTM Spec. Tech. Publ. no. 327, 1963.

OUTWATER, J. O.: Filament Wound Internal Pressure Vessels. Mod. Plastics, vol. 40, Mar. 1963, pp. 135-139.

OUTWATER, J. O.: The Effect of Repeated Loading on Filament Wound Internal Pressure Vessels. Vermont Univ. (Burlington), Contract Nonr 32-19(01)(X), Sept. 1963.

OUTWATER, J. O.; AND SEIBERT, W. J.: On the Mechanics of the Failure of Filament Wound Vessels. Proc. 18th Ann. Conf., Ref. Plas. Div., SPI, 1963.

RITTENHOUSE, J. B.; ET AL.: Large Solid Propellant Boosters for Spacecraft. SAMPE Symp. sec. 19, June 1963.

ROMUALDI, J. P.; AND BATSON, G. B.: Mechanics of Crack Arrest in Concrete. Proc. Am. Soc. Civil Engrs., vol. 89, EM 3 (J. Eng. Mech. Div.), 1963, pp. 147-168.

RONAY, M.: Non-Homogeneous Straining and Fracture Mechanism in a Filled Elastomer. Columbia Univ., Dept. Civil Eng. and Eng. Mech. TR 19, Contract Nonr 266(78), 1963.

ROSEN, B. W.; DOW, N. F.; AND HASHIN, Z.: Mechanical Properties of Fibrous Composites. NASA CR-31, Aug. 1963.

ROSKOS, T. G.; Pflederer, F. R.; AND PROSEN, S. P.: The Compressive Testing of Filament Wound Ring Specimens. Proc. 18th Ann. Conf., Ref. Plas. Div., SPI, 1963.

SCHUERCH, H.: Analytical Design for Optimum Filamentary Pressure Vessels. AIAA Preprint Paper 2914-63, Apr. 1963.

SHIBLEY, A. M.: Filament Winding in Military Applications: A Discussion of Problems Associated With Filament Wound Motor Cases. Picatinny Arsenal (Dover, N.J.), Sept. 1963.

SUTTON, W. H.; AND CHORNE, J.: Development of High-Strength, Heat Resistant Alloys by Whisker Reinforcement. Met. Eng. Quart., vol. 3, no. 1, 1963.

THROCKMORTON, P. E.; HICKMAN, H. M.; AND BROWNE, M. F.: Origin of Stress Failure in Glass Reinforced Plastics. Mod. Plastics, vol. 41, no. 3, 1963, p. 140, 142, 145, 148, 150, 189-190, 192, 198.

TOLLEY, C. P.; ET AL.: Boron Reinforcements for Structural Composites. Proc. Conf. Structural Plastic Adhesives and Filament Wound Composites, ASD Tech. Rept. no. ASD-TDR-63-396, Aeron. Sys. Div., WPAFB, Ohio, 1963.

TSISKRELY, G. D.: On Tensile Strength of Reinforced Concretes. Beton i Zhelezobeton, no. 3, 1963, pp. 124-127.

WILSON, F.: Filament Wound Pressure Vessels. Telecomputing Corp. (San Diego), Final Rept., Dec. 1963.

YOUNG, R. E.: Filament Wound Products—Design and Fabrication. SAE Paper 684B, Mtg. Apr. 8-11, 1963.

1964

Aerojet-General, Sacramento, Calif.: Investigation of Filament Winding Patterns. Bimonthly Rept., Contract NOw 63-027-C(FBM), Mar. 1964.

ANON.: Glass-Fiber-Reinforced Plastic Structures. NASA SP-5018, June 2, 1964.

ANON.: Textile Fiber Materials for Industry. Owens-Corning Fiberglas Corp., Feb. 1964.

ANSON, M.: An Investigation Into a Hypothetical Deformation and Failure Mechanism for Concrete. Mag. Concrete Res., vol. 16, no. 47, 1964, pp. 73–82.

ARRIDGE, R. G. C.; BAKER, A. A.; AND CRATCHLEY, D.: Metal Coated Fibres and Fibre Reinforced Metals. J. Sci. Instr., vol. 41, 1964, pp. 259–261.

BERSHTEIN, V. A.; AND GLIKMAN, L. A.: On a Rapid Method of Determining the Fatigue Strength of Fiber Glass Reinforced Plastics. Industr. Lab., vol. 30, no. 2, 1964, pp. 274–277. (Transl. Zavodskaya Lab., vol. 30, no. 2, 1964, pp. 215–218, by Instr. Soc. Am., Pittsburgh, Pa.)

BOUE, C. A.: A Microscopic Study of Mode of Fracture in Filament-Wound Glass-Resin Composites. Proc. 19th Ann. Tech. and Manag. Conf., Ref. Plas. Div., SPI, 1964.

BROMS, B. B.: Stress Distribution, Crack Patterns, and Failure Mechanisms of Reinforced Concrete Members. J. Am. Concrete Inst., vol. 61, no. 12, 1964, pp. 1535–1557.

BURKLEY, R. A.; BOLLER, T. J.; AND BUTCHER, I. R.: Study of the Effects of Mechanical Damage on the Performance of Filament Wound Motor Cases. Goodyear Aerospace (Akron, Ohio), Contract NOw 63-0449-c(FBM), June 1964.

CHAZOTTES, M. L.: Filament Winding Is Coming of Age. Can. Chem. Process., vol. 48, no. 4, Apr. 1964, pp. 90, 93–94.

CHESTER, B. F.: Determination of Effects of Materials and Process Variables on Filament Wound Structures. Aerojet-General Corp. (Azusa, Calif.), Contract N140(131)-75687B(X), July 1964.

COONEY, J. J.: Development of Improved Resin Systems for Filament Wound Structures. Aerojet Corp. (Sacramento, Calif.), Contract NOw 630627-C(FBM) Aug. 1964.

CORNISH, R. H.; ABBOTT, B. W.; AND COLE, C. K.: An Investigation of Material Parameters Influencing Creep and Fatigue Life in Filament Wound Laminates. Contract NoBs 86461, May 1964.

CORNISH, R. H.; NELSON, H. R.; AND DALLY, J. W: Compressive Fatigue and Stress Rupture Performance of Fiber Reinforced Plastics. Proc. 19th Ann. Tech. and Manag. Conf., Ref. Plas. Div., SPI, 1964.

CRATCHLEY, D.; AND BAKER, A. A.: Tensile Strength of a Silica Fibre Reinforced Aluminum Alloy. Metallurgia, 1964.

DARMS, F. J.; AND LITVAK, S.: Optimum Design for Filament-Wound Rocket-Motor Cases. Proc. 19th Ann. Tech. and Manag. Conf., Ref. Plas. Div., SPI, 1964.

DARMS, F. J.; MOHLO, R.; AND CHESTER, B. E.: Improved Filament Wound Construction for Cylindrical Pressure Vessels. Vol. I: Structural Analysis and Materials and Processes. Aerojet-General Corp. (Azusa, Calif.), Contract AF33(616)8442, Mar. 1964.

DARMS, F. J.; MOHLO, R.; AND CHESTER, B. E.: Improved Filament Wound Construction for Cylindrical Pressure Vessels. Vol. II: Design Procedures. Aerojet-General Corp. (Azusa, Calif.), Contract AF33(616)8442, Mar. 1964.

DAVIS, J. W.: Standards for Filament Wound Plastics. SPE J., vol. 20, no. 7, July 1964, pp. 601–604.

Downey, T. F.: Development and Evaluation of Improved Resins for Filament Wound Plastics. U.S. Polymeric Chemicals, Inc. (Santa Ana, Calif.), Quart. Rept., Dec. 1964.

Elkin, Robert A.: Interlaminar Shear of Filament Wound Reinforced Plastics. Telecomputing Corp. (San Diego, Calif.), Contract N60921-7094, Oct. 1964.

Epstein, G.: Filament Winding Puts Strength Where Needed. Mater. Design Eng., vol. 59, Feb. 1964, p. 90.

Fracture of Engineering Materials. Papers presented at a conference sponsored by the Eastern New York Chapter of the American Society of Metals, Aug. 1959, Am. Soc. for Metals, 1964, pp. 66-69.

Fried, N.; and Kaminetsky, J.: The Influence of Material Variables on the Compressive Properties of Parallel Filament Reinforced Plastics. Proc. 19th Ann. Tech. and Manag. Conf., Ref. Plas. Div., SPI, 1964.

Gangulee, A.; and Gurland, J.: Brown Univ. Tech. Rept. no. 8, Contract AT(301)2394, 1964.

Hanley, D. P.: Rocket Motor Case Winding Pattern Studies. Hercules Powder Co., Cumberland, Md., 1964.

Harrington, R. A.: Design Information From Analytical and Experimental Studies on Filament Wound Structures Subjected to Combined Loading. B. F. Goodrich Co. (Rialto, Calif.), (ABL03828), Feb. 1964.

Hashin, Z.; and Rosen, B. W.: The Elastic Moduli of Fiber-Reinforced Materials. Trans. ASME, vol. 31, no. 2, 1964, pp. 223-232.

Hashin, Z.: Theory of Mechanical Behavior of Heterogeneous Media. Appl. Mech. Rev., vol. 17, no. 1, 1964, pp. 1-9.

Hashin, Z.; and Rosen, B. W.: The Elastic Moduli of Fibre Reinforced Materials. J. Appl. Mech., June 1964.

Hill, R.: Theory of Mechanical Properties of Fibre-Strengthened Materials: Pt. 1, Elastic Behavior. J. Mech. Phys. Solids, vol. 12, no. 4, 1964, pp. 199-212.

Keegin, S. W.; and Siefert, R. F.: Filament Winding Goes Commercial. Machine Design, vol. 36, no. 11, May 7, 1964, pp. 130-133.

Kinna, M. A.; and Warfield, R. W.: Tne Winding and Curing of Large Reinforced Panal Structures, Naval Ordance Lab., 1964.

McGarry, F. J.: Resin Cracking in Composites. The Chem. Engr., Inst. of Chem. Engrs. (London), 1964.

Myers, N. C.; Lee, G. D.; Wright, F. C.; and Daines, J. V.: Investigation of Structural Problems With Filament Wound Deep Submersibles. Thompson Fibre Glass Co. Contract NObs-88351 RDD 2994, Jan. 1964.

Oleesky, S. S.; and Mohr, J. G.: Handbook of Reinforced Plastics. Reinhold Pub. Corp. (New York), 1964.

Patrik, R. L.; Ripling, E. J.; and Mostovoy, S.: Fracture Mechanics Applied to Heterogeneous Systems. Proc. 19th Ann. Tech. and Manag. Conf., Ref. Plas. Div., SPI, 1964.

Pelloux, R. M. V.: Fractographic Analysis of the Influence of Constituent Particles on Fatigue Crack Propagation in Aluminum Alloys. Trans. Am. Soc. Metals, vol. 57, no. 2, 1964, pp. 511-518.

Proceedings of the World Conference on Shell Structures. Building Res. Inst. Publ., 1964.

Rabinovich, A. L.: Equations for the Two-Dimensional Stressed State in Oriented Glass Fiber Plastics. Soviet Phys. Dokl., vol. 8, no. 12, 1964, pp. 1233-1236.

Ronay, M.: On Strain Incompatibility and Grain Boundary Damage in Fatigue. Columbia Univ., Dept. Civil Eng. and Eng. Mech., TR 9, Contract Nonr 266(91), 1964.

Rosato, D. V.: Reinforced Plastics. SPE J., vol. 20, no. 5, May 1964, pp. 414-415.

ROSATO, D. V.: Why Use Metal Wires in Filament Winding? Iron Age, vol. 193, Mar. 26, 1964, pp. 102–103.

ROSEN, B. W.: A Note on the Failure Model of Filament Reinforced Materials Including the Influence of Constituent Geometry and Properties. Proc. 19th Ann. Tech. and Manag. Conf., Ref. Plas. Div., SPI, Feb. 1964.

ROSEN, B. W.; ET AL.: Hollow Glass Fiber Reinforced Laminates. Space Sciences Lab., General Electric Co., Aug. 15, 1964.

ROSEN, B. W.: Tensile Failure of Fibrous Composites. Prepr. no. 64–73, Presented at AIAA Aerospace Sci. Mtg. (New York), Jan. 1964.

ROSEN, B. W.; DOW, N. F.; AND HASHIN, Z.: Mechanical Properties of Fibrous Composites. NASA CR-31, General Electric Co., Contract No. NAsw 470, 1964.

SECOR, K. E.; AND MONISMITH, C. L.: Analysis and Interrelation of Stress-Strain-Time Data for Asphalt Concrete. Trans. Soc. Rheol., vol. 3, 1964, p. 19.

SCHMIDT, G. A.: Whisker Matrix Bonding Studies. G.E. Space Sciences Lab. Progr. Repts. on U.S.N. BuWeps Contract NOw–64–0330–d, 1964–65.

SHAFFER, B. W.: Material Properties of Reinforced Plastics. Soc. Plas. Engrs., Trans., vol. 4, 1964, pp. 67–77.

SHAFFER, B. W.: Stress-Strain Relations of Reinforced Plastics Parallel and Normal to Their Internal Filaments. AIAA J., vol. 2, no. 2, 1964, pp. 348–352.

TSAI, S. W.: Structural Behavior of Composite Materials. Philco Corp. Res. Lab., Publ. no. U–2428, 1964.

WEST, D. C.: Resin-Fiber Load Transfer in Fiber-Reinforced Plastics. Proc. 19th Ann. Tech. and Manag. Conf., Ref. Plas. Div., SPI, 1964.

YARSLEY, V. E.; AND FAVELL, W.: Reinforced Plastics in the United Kingdom and Continental Europe in the Year 2000. Proc. 19th Ann. Mtg. SPI (Chicago, Ill.), 1964, Sec. 16C.

1965

ANON.: Doing the Tough Jobs in Space. Mod. Plastics, vol. 43, no. 2, Oct. 1965, p. 90.

ANON.: Filament Strengthens Epoxy Pipe Oil and Gas. Oil Gas J., vol. 63, no. 110, May 31, 1965.

ANON.: Filament Winding High Strength Products: Review of an Important Reinforced Plastic Fabricating Technique. Plastics World, vol. 23, pp. 26–29, Feb. 1965.

ANON.: Filament Winding Made Easier—Plastics Technology. Vol. VII, no. 10, Oct. 1965, p. 9.

ANON.: Filament Wound Reinforced Plastic Vessels Made by New Process Withstand High Pressures. Mod. Plastics, vol. 42, May 1965, p. 142.

ANON.: Flat RP/PP Sheets Make Tall Round Silo. Mod. Plastics, vol. 43, no. 4, Dec. 1965, p. 100.

ANON.: For Fast Redesign: Filament Wound Reinforced Plastic. Mod. Plastics, vol. 42, Apr. 1965, pp. 10–11.

ANON.: High Strength Low Alloy Steels. Mat. Design Eng., vol. 61, no. 2, Feb. 1965, p. 113.

ANON.: Improved Skis Pioneer New Laminate Construction. Mod. Plastics, vol. 43, no. 4, Dec. 1965, p. 102.

ANON.: Low Carbon Steel Is Strongest Ever Made. Mat. Design Eng., vol. 61, no. 2, Mar. 1965, p. 7.

ANON.: New Chemicals and Materials for Fibers and Filaments. Ind. Eng. Chem., vol. 57, Sept. 1965, p. 94.

ANON.: Proc. 6th Ann. Tech. Symp., Filament Structures Technology, ASME and Univ. of New Mexico, 1965.

BASCOM, W. D.: Some Surface Chemical Aspects of Glass-Resin Composites, pt. 2: The Origin and Removal of Microvoids in Filament Wound Composites. Naval Res. Lab., NRL6268, 1965.

BELL, R. L.; AND YOUNG, E. C.: Filament Wound Fiberglass Tanks Cut Costs. Chem. Eng. Progr., vol. 61, Apr. 1965, pp. 57–64.

BOLLER, T. J.; BUTCHER, I. R.; AND BURKLY, R. A.: Damage and Repair Aspects of Filament-Wound Motor Cases. Proc. 20th Ann. Tech. Conf., SPI, Feb. 1965, paper 12 E.

BROMS, B. B.: Stress Distribution in Reinforced Concrete Members With Tension Cracks. J. Am. Concrete Inst., vol. 62, no. 9, 1965, pp. 1095–1108.

BROUTMAN, L. J.: Failure Mechanisms for Filament Reinforced Plastics. Mod. Plastics, vol. 42, Apr. 1965, pp. 143–145.

BURGMAN, J. A.: Hollow Glass Fibers. Proc. 20th Ann. Tech. Conf., Ref. Plas. Div., SPI, 1965.

CHIAS, T. T.: Design for Commercial Filament Winding. SPE, New Frontiers for Reinforced Plastics, Tech. Papers for Mtg., Oct. 4–5, 1965, p. 7–20.

DIETZ, A. G. H.: Composite Materials. Presented at 68th ASTM Ann. Mtg. (West Lafayette, Ind.), June 16, 1965.

DRUCKER, D. C.: 2d Intern. Symp. High Strength Materials (Univ. Calif., 1964), High Strength Materials, V. F. Zackay, ed., John Wiley & Sons, Inc., 1965, p. 795.

Editors, Materials in Design Engineering: Filament Winding: Its Promise for Industrial and Consumer Products. Mat. Design Eng., vol. 61, no. 4, Apr. 1965, pp. 117–132.

ELPATÉVSKII, A. N.; AND VASILÉV, V. V.: Investigation of the Stress State of a Glass-Wound Cylindrical Shell. Inzh. Zh., vol. 5, no. 1, 1965, pp. 129–142.

ESHBAUGH, R. W.: Filament Winding System, Design and Operational Characteristics. Naval Ordnance Lab., White Oak, Md , NoL TR 64 221, Feb. 1965.

FRIED, N.; AND SILVERGLEIT, M.: Compressive Strength of Filament Wound Plastic Materials. SPE 21st Ann. Tech. Conf., Mar. 1965, paper XI–1.

GILLIS, T. W.: Randomly Distributed Rough Particles in Concentrated Suspensions. Trans. Soc. Rheol., vol. 9, pt. 1, 1965, p. 447.

HABIP, L. M.: A Review of Recent Work on Filamentary Structures. Intern. J. Mech. Sci., vol. 7, no. 4, 1965, pp. 297–299.

HALL, A. M.: Getting the Most Out of Ultra High Strength Steels. Mat. Design Eng., vol. 61, no. 2, Mar. 1965, p. 104.

H DE L'ESTOILE: Properties of Composite Materials, Reinforced by Trichite Filaments (Whiskers). May 1965, Refs. Trans. English from Doc-Air-Espace (France), no. 90, 1965, pp. 19–24, NASA N65–27680.

HENNESSEY, J. M.; WHITNEY, J. M.; AND RILEY, M. B.: Experimental Methods of Measuring Shear Modulus of Fiber Reinforced Composite Materials. Air Force Materials Lab., WPAFB, ML–TDR–64–42, 1965.

HERZOG, J. A.: Potential of Composite Materials to 2000° F. A66–17110, Soc. Autom. Engr., Combined Powerplant and Transportation Mtg. (Cleveland, Ohio), 1965.

HOFFMAN, J. W.: Composite Materials for Launch Shell Structure. AIAA 2d Ann. Mtg. (San Francisco, Calif.), Preprint Paper 65–286, July 1965.

INNES, G.: Connections for Strip Wound Pressure Vessels. Engineer, vol. 219, May 7, 1965, pp. 807–808.

KELLY, A.; AND DAVIES, G. J.: Met. Rev., vol. 10, 1965, p. 1.

KUHN, W.: Das Vickelverfahren fuer hochbeanspructe glasfaserverstaerkte Kunststoffrohre adn Drucklehaelter. Kunststaffe, vol. 55, no. 5, May 1965, pp. 375–378.

LEVENETZ, B.: Large-Diameter-Fiber Reinforced Plastic Composite Structures. Proc. 20th Ann. Conf., Ref. Plas. Div., SPI, 1965.

LIU, C. T.; AND GURLAND, J.: Trans. Am. Soc. Metals, vol. 58, 1965, p. 66.

MAJERUS, J. N.; AND FERRIERA, S. K.: Fracture and Reliability of Filament Wound Chambers. Proc. Am. Soc. Civil Engrs., vol. 91, EM 1 (J. Eng. Mech. Div., pt. 1), 1965, pp. 107–136.

MARTENSON, J. A.: Do Filament Wound Reinforced Plastics Really Meet Requirements? Proc. 22d Ann. Tech. Conf., SPI (Coronado, Calif.), Mar. 25–26, 1965, pp. 19–1—19–4.

MARTENSON, J. A.: Materials for Filament Wound High Temperature Structural Uses. Mod. Plastics, vol. 42, May 1965, pp. 167–168.

MCGARRY, F. J.: Composite Structural Materials. MIT-Dept. of Civil Engineering, 1965.

MIKULASYK, M. M.; AND STEIN, M.: Bucklings of a Cylindrical Shell Loaded by a Pretensioned Filament Winding. AIAA J., Mar. 1965, pp. 560–561.

MILAWSKI, J. V.; AND SHYNE, J. J.: Whiskers Make Reinforced Plastics Better Than Metals. Proc. 20th Ann. Conf., Ref. Plas. Div., SPI, 1965.

PARADY, V.: How To Reinforce Holes in Filament Structures. Mat. Design Eng., vol. 61, no. 2, Feb. 1965, p. 108.

PARRATT, N. J.: Reinforcement of Plastics and Metals by Silicon Nitride Whiskers. Am. Inst. Chem. Eng., 58th Ann. Mtg. (Philadelphia, Pa.), Symp. Whisker Tech., Prepr. 18D, 1965.

PETERSON, G. P.: Engineering Properties of High Modulus Reinforced Plastics. Rept. no. ASD–TDR–62–65, Aeron. Sys. Div., WPAFB, Ohio, 1962. Plastics World, Mar. 1965.

ROSATO, D. V.; AND GROVE, G. S., JR.: Filament Winding. John Wiley & Sons, Inc., 1965.

ROSEN, B. W.: Fiber Composite Materials. Am. Soc. Metals, 1965, p. 37.

SCHUERCH, H. U.: A Contribution to the Micromechanics of Composite Materials-Stresses and Failure Mechanisms Induced by Inclusions. NASA CR–582, Sept. 1966.

SIEGRIST, F. L.: Reinforced Plastics in Engineering Applications. Metal Progr., vol. 87, May 1965, pp. 93–99.

STEDFIELD, R. L.; AND HOOVER, C. T.: The Filament Wound Pressure Vessel. Allison V5, 1965, p. 2.

TSAI, J.; FELDMAN, A.; AND STANG, D. A.: The Buckling Strength of Filament Wound Cylinders under Axial Compression. NASA CR–266, July 1965.

WATSON, H.: Plastics in Building. Can. Plastics, vol. 23, Oct. 1965, pp. 28–33.

WEST, P.: Whisker Composites: Where Do They Stand Today? Mat. Design Eng., 1965, pp. 112–116.

YOUNG, J.: Continuous Filaments. Strengthen FRP. Design Eng., vol. II, no. 8, Aug. 1965, p. 30–31.

YOUNG, R. E.: Filament Wound Structures. ASTM Spec. Tech. Publ. 375, 1965, pp. 63–72.

1966

ANON.: Abstract from Mod. Plastics, Jan. 1966, p. 60.

ANON.: Building Construction—Plastics Most Explosive Growth Market, Poses Many Challenges. Mod. Plastics, vol. 43, no. 9, May 1966, p. 180.

ANON.: Epoxy Tape Adhesive Provides High Strength With Low Heat Cure. Mat. Design Eng., vol. 63, no. 2, Feb. 1966, p. 19. Reinforced Thermoplastics.

ANON.: Mod. Plastics, vol. 43, no. 6, Feb. 1966, p. 97.

BACON, R.; PALLOZZI, A. A.; AND SLOSARIK, S. E.: Carbon Filament Reinforced Composites. Proc. 21st Ann. Tech. Conf., SPI, Feb. 1966, Paper 8-E, p. 6.

BOHLEN, J. C.; AND KIRKPATRICK, H. B.: Use of Polyester Resin in Filament Wound Dwelling Units. Proc. 21st Ann. Tech. Conf., SPI, Feb. 1966, Paper 2-E, p. 8.

CRYOR, R. E.: Asbestos Reinforced Plastics Resist Heat and Chemical. Mat. Design Eng., Apr. 1966, pp. 94-96.

DUNBAR, G. R.: Filament Winding. ASME Paper 66, MD-86 Mtg. May 9-12, 1966, p. 5.

FLETCHER, J. A.; ET AL.: Can RTP Meet Your Produce Requirements? Mod. Plastics, vol. 43, no. 7, Mar. 1966, p. 102.

GALLI, J. R.; AND POLLOCK, M. R.: Development of Reinforced Plastic Parts for Aircraft. Mod. Plastics, vol. 43, no. 5, Jan. 1966, p. 138.

GARRITANO, R. A.: Techniques for Filament Winding Parts. Machine Design, Apr. 14, 1966, pp. 167-172.

GRIMES, D. L.: Composite Structures—They Start Where Metals Leave Off. Mod. Plastics, vol. 43, no. 9, May 1966, p. 221.

GURLAND, J.: Some Aspects of the Fracture of Metallic Composites. U.S. Atomic Energy Commission Contract no. A7(30-1)-2394, TR no. 18, 1966.

HAUCK, J. E.: Forecast '66 Plastics and Non-Metallics. Mat. Design Eng., vol. 63, no. 1, Jan. 1966, p. 82.

HAUCK, J. E.: New Inorganic Filaments Are Stiffer and Lighter Than Metals. Mat. Design Eng., vol. 63, no. 2, Feb. 1966, p. 82.

HOLLIDAY, L., ED.: Composite Materials. Elsevier Publ. Co. (London), 1966.

KORMAN, S.: Some New Metal and Metal-Ceramic Composites. NASA SP-5060, 1966.

MILEWSKI, J. V.: R and D Challenge of Whisker Composites. Res./Develop., Mar. 1966, pp. 31-34.

MILLER, R. J.; NEBESAR, R. J.; AND SCHNEIDER, M. H., JR.: Design and Test of Ring Stiffened Fiberglass Cylinders Under External Pressure. Proc. 21st Ann. Tech. Conf., SPI, Feb. 1966, Paper 17-D, p. 16.

SUTTON, W. H.: Whisker Composite Materials—A Prospectus for the Aerospace Designer. Astronaut. Aeron., Aug. 1966, pp. 46-51.

TSAI, S. W.; ADAMS, D. F.; AND DONER, D. R.: Analysis of Composite Structures. NASA CR-620, Nov. 1966.

STRUCTURAL DESIGN SYNTHESIS AND OPTIMIZATION

1948

GOMZA, A.; AND SEIDE, P.: Minimum Weight Design of Simply Supported Transversely Stiffened Plates Under Compression. NASA TN 1710, 1948.

1954

CATCHPOLE, E. J.: The Optimum Design of Compressive Surfaces Having Unflanged Integral Stiffeners. J. Roy. Aeron. Soc., vol. 58, Nov. 1954, pp. 765-768.

DOW, N. F.; LIBOVE, C.; AND HUBKA, R. E.: Formulas for the Elastic Constants of Plates With Integral Waffle-Like Stiffening. NACA Rept. 1195, 1954.

1955

KLEIN, B.: Direct Use of Extremal Principles in Solving Certain Optimization Problems Involving Inequalities. J. Operations Res. Soc. Am., vol. 3, no. 2, May 1955, pp. 168-175, 548.

1956

GERARD, G.: Minimum Weight Analysis of Compression Structures. New York Univ. Press, 1956.

LEKHNITSKII, S. G.: Anisotropic Plates. *In* Contributions to the Metallurgy of Steel, no. 50, Am. Iron & Steel Inst., June 1956.

1958

COX, H. L.: Structures of Minimum Weight: The Basic Theory of Design Applied to the Beam Under Pure Bending. Aeron. Res. Council Rept. 19785, Jan. 1958.

COX, H. L.: The Theory of Design. Aeron. Res. Council Rept. 19791, Jan. 1958.

DORFMAN, R.; SAMUELSON, P. A.; AND SOLOW, R. M.: Linear Programming and Economic Analysis. McGraw-Hill Book Co., Inc., 1958.

GERARD, G.: An Evaluation of Structural Sheet Materials in Missile Applications. Jet Propulsion, vol. 28, no. 8, Aug. 1958, pp. 511–520.

HILTON, H. H.; AND FEIGEN, M.: Minimum Weight Analysis Based on Structural Reliability. Proc. 3d Symp. High Speed Aerodynamics and Structures, USAF, ARDC, vol. II, Mar. 1958, pp. 11–173.

MICKS, W. R.: Bibliography of Literature on Optimum Design of Structures and Related Topics. The Rand Corp., RM-2304, ASTIA AD-215771, Dec. 15, 1958.

PEARSON, C.: Structural Design by High-Speed Computing Machines. Proc. 1st Natl. Conf. Electron. Comp., Struct. Div., ASCE (Kansas City, Mo.), 1958, pp. 417–436.

SCHMIDT, L. C.: Fully Stressed Design of Elastic Redundant Trusses Under Alternative Load Systems. Australian J. Appl. Sci., vol. 9, Dec. 1958, pp. 337–348.

1959

GERARD, G.: Structural Interplay: Design and Materials. Aero/Space Eng., Aug. 1959, pp. 37–42.

1960

GERARD, G.: Minimum Weight Analysis of Orthotropic Plates Under Compressive Loading. J. Aerospace Sci., vol. 27, 1960, pp. 21–26, 64.

HILTON, H. H.; AND FEIGEN, M.: Minimum Weight Analysis Based on Structural Reliability. J. Aerospace Sci., vol. 27, 1960, pp. 641–652.

SCHMIT, L. A.: Structural Design by Systematic Synthesis. Proc. 2d Natl. Conf. Electron. Comp., Struct. Div., Am. Soc. Civil Engrs., 1960, pp. 105–132.

SHANLEY, F. R.: Weight-Strength Analysis of Aircraft Structures. 2d ed., Dover Publications, 1960.

WILLIAMS, M. L.; GERARD, G.; AND HOFFMAN, G. A.: Selected Areas of Structural Research in Rocket Vehicles. XI Intern. Aeron. Congr. (Stockholm), vol. 1, 1960, pp. 146–166.

1961

GERARD, G.: Minimum Weight Design of Ring Stiffened Cylinders under External Pressure. J. Ship Res., vol. 5, no. 2, Sept. 1961, pp. 44–49.

SCHMIT, L. A.; AND KICHER, T. P.: Structural Synthesis of Symmetric Waffle Plates With Integral Orthogonal Stiffeners. EDC Rept. 2-61-1, Case Inst. of Tech., Oct. 1961.

1962

BERT, C. W.; AND HYLER, W. S.: Design Considerations in Selecting Materials for Large Solid-Propellant Rocket-Motor Cases. Defense Metals Information Center Rept. 180, Dec. 1962.

FEDER, D. P.: Automatic Lens Design With a High-Speed Computer. J. Opt. Soc. Am., vol. 52, no. 2, 1962, pp. 177–183.

HOFFMAN, G. A.: Minimum-Weight Proportions of Pressure-Vessel Heads. J. Appl. Mech., vol. 29, no. 4, Dec. 1962, pp. 662–668.

JOHNSTON, G. S.: Weight of Ellipsoid of Revolution With Optimum Tapered Thickness. J. Aerospace Sci., vol. 29, no. 10, Oct. 1962, pp. 1269–1270.

KALABA, R.: Design of Minimum Weight Structures for Given Reliability and Cost. J. Aerospace Sci., vol. 29, 1962, pp. 355–356.

LEITMANN, G., ED.: Optimization Techniques. Academic Press, 1962.

SCHMIT, L. A.; AND KICHER, T. P.: Structural Synthesis of Symmetric Waffle Plate. NASA TN D-1691, Dec. 1962.

SCHMIT, L. A.; AND KICHER, T. P.: Synthesis of Material and Configuration Selection. J. Struct. Div., Proc. Am. Soc. Civil Engrs., vol. 88, no. ST3, June 1962, pp. 79–102.

SCHMIT, L. A.; AND MALLETT, R. H.: Design Synthesis in a Multi-Dimensional Space With Automated Material Selection. Eng. Design Center Rept. 2–62–2, Case Inst. of Tech., Aug. 1962.

1963

BERT, C. W.: Ellipsoidal Closures for Minimum Weight Pressure Vessels. Can. Aeron. Space J., vol. 9, May 1963, pp. 133–136.

COHEN, G. A.: Optimum Design of Truss-Core Sandwich Cylinders Under Axial Compression. AIAA J., vol. 1, no. 7, July 1963, pp. 1626–1630.

PIPKIN, A. C.; AND RIVLIN, R. S.: Minimum-Weight Design for Pressure Vessels Reinforced With Inextensible Fibers. J. Appl. Mech., vol. 30, no. 1, Mar. 1963, pp. 103–108.

SCHMIT, L. A.; AND MORROW, W. M.: Methods of Waffle Plate Synthesis. EDC Rept. 2–63–3, Case Inst. of Tech., June 1963.

SCHMIT, L. A.; AND MORROW, W. M.: Structural Synthesis With Buckling Constraints. J. Struct. Div., Proc. Am. Soc. Civil Engrs., vol. 89, no. ST2, Apr. 1963, pp. 107–126.

SCHUERCH, H.: Analytical Design for Optimum Filamentary Pressure Vessels. AIAA Preprint 2914–63, Apr. 1963.

SLINEY, J. L.; CORRIGAN, D. A.; AND SCHMID, F.: Preliminary Report on the Biaxial Tensile Behavior of Anisotropic Sheet Materials. U.S. Army Material Research Agency, AMRA TR 63–11, Aug. 1963.

1964

BEST, G. C.: Completely Automatic Weight-Minimization Method for High Speed Digital Computers. J. Aircraft, vol. 1, 1964, pp. 129–133.

BREWER, W. N.; AND JEPPESON, N. L.: Methods of Evaluation of Inflatable Structures for Space Applications. Proc. AIAA/ASME 5th Ann. Structures and Materials Conf., Apr. 1964, pp. 344–360.

Editors, Materials in Design Engineering: Selecting Materials for Minimum Weight. Mat. Design Eng., Mar. 1964.

MARTIN, W. J.; MATSUDA, T.; AND KALUZA, E. F.: Study of Titanium Alloy Tankage at Cryogenic Temperatures. Douglas Aircraft Co. Rept. no. SM–43116, May 1964. (U.S. Dept. Com., OTS, AD–604053).

MILLS, E. J.; ATTERBURY, T. J.; CASSIDY, L. M.; EIBER, R. J.; DUFFY, A. R.; IMGRAM, A. E.; AND MASUBUCHI, K.: Design, Performance, Fabrication and Material Considerations for High-Pressure Vessels. U.S. Army Missile Command, Redstone Scientific Information Center, RSIC–173, Mar. 1964.

MORRIS, E. E.: Glass Cases for the Biggest Solids. Astronaut. Aeron., vol. 2, no. 7, July 1964, pp. 28–38.

MOSS, H.: A Generalized Approach to the Selection of Vehicle Design Parameters. Soc. Automotive Engrs. Paper 782C, 1964.

SCHMIT, L. A.: Comment on Completely Automatic Weight Minimization Method for High-Speed Digital Computers. AIAA J., vol. 1, no. 6, 1964, pp. 375–377.

SCHMIT, L. A.; AND FOX, R. L.: An Integrated Approach to Structural Synthesis and Analysis. Proc. AIAA/ASME 5th Ann. Structures and Materials Conf., Apr. 1964, pp. 294–315.

1965

CRAWFORD, R. F.; AND SCHWARTZ, D. B.: General Instability and Optimum Design of Grid-Stiffened Spherical Domes. AIAA J., vol. 3, no. 3, Mar. 1965, pp. 511–515.

GERARD, G.: Aerospace Pressure Vessel Design Concepts. NASA CR–287, Aug. 1965.

JOHNSTON, G. S.; AND LANTZ, R. B.: Efficiency of Waffle Grid Panels Under Compressive Loading. Aircraft J., vol. 2, Sept.–Oct. 1965, pp. 437–438.

KICIMAN, M. O.: Optimum Cost and Weight Design of Structural Members for Air Vehicles Subjected to Elevated Temperatures. Ph. D. thesis, Univ. of California at Los Angeles, 1965.

RANZANI, R.: Behavior of Fully Stressed Design of Structures and Its Relationship to Minimum-Weight Design. AIAA J., vol. 3, no. 12, Dec. 1965, pp. 2262–2268.

RANZANI, R.: The Iterative Smoothing Method and Its Application to Minimum-Cost Design of Highway Bridge Girders. Ph. D. thesis, Case Inst. of Tech., 1965.

1966 (Partial Listing)

ANON.: Structural Systems and Program Decisions. Vol. 1, NASA SP–6008, 1966.

ARMSTRONG, H. H.; Burns, A. B.; AND CRAWFORD, R. G.: Structural Weight and Cost Comparison of Large Space Structures Employing Beryllium-Aluminum Alloys. Proc. AIAA/ASME 7th Ann. Structures and Materials Conf. (Cocoa Beach, Fla.), Apr. 1966, pp. 388–397.

BURNS, A. B.; AND ALMROTH, B. O.: Structural Optimization of Axially Compressed, Ring-Stringer Stiffened Cylinders. AIAA J. Spacecraft Rockets, vol. 3, 1966, pp. 19–25.

GERARD, G.: Materials Evaluation and Design. Astronaut. Aeron., Mar. 1966, pp. 58–62.

GHISTA, D. N.: Structural Optimization With Probability of Failure Constraint. NASA TN D–3777, Dec. 1966.

HARRIS, L. A.; MITCHELL, J. C.; AND MORGAN, G. W.: Computer-Aided Design for Civil Engineering Structures. Symp. Technology Status and Trends (Huntsville, Ala.), Apr. 21–23, 1965, NASA SP–5030, 1966, pp. 147–160.

KICIMAN, M. O.: A Random Sampling Procedure With Applications to Structural Synthesis Problems. Preprint AIAA Paper 66–531, 4th Aerospace Sci. Mtg. (Los Angeles), 1966.

Glossary

Anisotropic—Having properties that differ in several directions; possessing several natural axes, i.e., preferred directions with respect to a particular property or properties.

Anticlastic—Having opposite curvatures; having the center of principal radii at different sides of the observed tangent plane.

Cementation (diffusion)—Process for application of a coating metal by means of diffusion at elevated temperatures. The base metal is heated with a powdered coating metal to a temperature high enough to permit diffusion of fine particles.

Cermet—Composite having ceramic grains embedded in a metal matrix. Structural constituent accounts for as much as 30 percent of total value.

Chemically rigidified structure—Structure given rigidity through chemical reaction.

Cladding—Process by which a dense homogeneous layer of metal is bonded firmly and permanently to the base metal on one or two sides.

Composite (composite material)—Mixture or combination of two or more macroconstituents that differ in form and/or material composition and that are essentially insoluble in one another.

Composite structures—Material systems, designed and produced for a given application, that are the finished structures or products themselves.

Criterion (merit) function—Mathematical equation expressing all considerations associated with selecting the prime solution of a design problem; the function to be optimized.

Design—Process of evolving a configuration to perform specific functional requirements.

Design process—Sequence of operation by which the design configuration is evolved.

Dispersion-hardened-alloy composite—Composite having hard particles (usually of submicron size) of a structural constituent dispersed in a matrix of softer metal. Structural constituent is usually less than 3 percent by volume.

Elastic recovery structures—Structures utilizing the basic elastic properties of the materials to deploy and provide structural rigidity.

Fiber (filament) composite—Composite of fibers in continuous or discrete filament form embedded in a continuous matrix.

Filament-wound structure—Structure formed by draining continuous fiber (filament) through a resin bath and winding continuously onto a form, or mandrel, corresponding in shape to the inner structure of the fabricated part.

Filled (skeletal) composite—A continuous three-dimensional constituent having a random network of open pores or passages, cells, or an ordered honeycomb filled with another constituent.

Flake composite—Composition of flat particles or flakes, usually of an isotropic material, held together by an interface binder or embedded in a continuous matrix.

Fully stressed design—Structural members or elements are proportioned by equalizing the stress in any member (or element) in at least one loading condition.

Functional constraint—Mathematical description of the archetype of the proposed design.

Geodesic dome—Type of roof employing the triangle and pentagon used in subdivision with bars of equal length. Vertexes of the curved figures forming the structures mark the arcs of great circles (geodesics).

Inflatable structure—Fabric or film envelope maintaining structural integrity by internal pressurization.

Interface—Surface forming the common boundary of the constituents.

Isostatics—Plot of principal stress lines.

Isotensoid—Filamentary structure (pressure vessel) in which the filaments are oriented so that there is equal and uniform tension in each fiber.

Isotropic—Having no preferred direction with respect to a particular property or properties; having no natural axes.

Laminar composite—Composition of layers of single constituents bonded as superimposed layers.

Lattice dome—Shell system constructed from thin laths in which a flat square grid is deformed into a spatially curved surface.

Matrix—Body constituent of a composite; the "enclosure" material.

Membranes—Very thin stress skins capable of resisting only tension.

Micromechanics—Analysis of composite behavior based on the individual constituents and their interactions.

Minimum-weight design—Arrangement of structural elements in which all the design requirements such as stresses, deflections, and geometric constraints are satisfied while the total weight of the entire structure is minimized.

Monolithic material—A simple material.

Optimization—Technique for finding the best combination of design parameter values that satisfy the design requirements.

Orthotropic material—Material possessing three natural axes at right angles.

Particulate composite—Composition of minute particles, usually of uniform shape, embedded in a continuous matrix.

Regional constraint—Sets the allowable limits on design parameters or derived groups of parameters that represent the more complex attributes of the proposed object.

Roving—Untwisted grouping of filaments.

Sandwich—Construction comprising a combination of alternating dissimilar, simple or composite materials, assembled and intimately fixed in relation to each other so as to use the properties of each to the specific structural advantage of the whole assembly.

SAP—Sintered aluminum powder.

Self-lubricating alloy—Dispersion of dry lubricant powder in a metal matrix.

Semimonocoque—Skin structure stiffened by a number of reinforcing elements.

Structural constituent—Constituent determining the internal structure of the composite.

Synthesis—Fitting together design elements or separate concepts to produce an integrated whole.

Unfurlable structure—Body that is mechanically deployed by hinges, sliding sections, and telescoping members.

Viscoelastic material—Material having behavior characteristic of fluids while maintaining some of the rigidity of solids.

Symbols

Notations

- A area, in.^{2}; subspace of acceptable design points
- b spacing; width, in.
- C coefficient; buckling correction factor; optimum design parameter
- c specific heat, Btu/lb/°F
- D structural design index, psi
- d stringer spacing, in.; diameter, in. or ft
- E modulus of elasticity, psi
- e emissivity
- F force, lb; merit function to be optimized
- G shear modulus, psi; combined constraint boundary between A and U
- g_i ith constraint function
- h depth, in.
- I moment of inertia, in.^{4}
- J torsional constant, in.^{4}
- K coefficient; shape factor
- k thermal conductivity, Btu/hr/ft^2/°F; coefficient
- k/c_p diffusivity, ft^2/hr
- L length, in. or ft
- l spacing, in. or ft
- M material efficiency factor, pci/psi; moment, in.-lb or ft-lb
- m number of half waves in cylinder buckle pattern in longitudinal direction
- N edge or end loading, lb/in.
- n number of variables; number of full waves in cylinder buckle pattern in circumferential direction
- P load, lb; probability symbol
- p pressure, psi
- R radius, in. or ft
- S structural strength, psi; structural efficiency factor, nondimensional; design space
- t thickness, in.
- U criterion (merit) function; space of unacceptable design points

273-140 O - 67 - 12

V volume, in.^{3} or ft^3
W weight, lb; weight efficiency factor, pci/psi
w width, in. or ft
X variable; design parameter
x coordinate
y coordinate
α coefficient of thermal expansion, in./in.-°F
η plasticity reduction factor, nondimensional
μ Poisson's ratio, nondimensional
ξ variable; coordinates
ρ density, pci
σ stress, psi
σ_{cy} compressive yield strength, psi
ϕ regional constraint
ψ functional constraint

Subscripts

c core
f face sheet
r rib
s stringer
w depth
x x-direction
y y-direction

Special Notations

A_{11} Extensional stiffness in longitudinal direction, lb/in.
A_{22} Extensional stiffness in circumferential direction, lb/in.
A_{33} Shear stiffness, lb/in.
D_{11} Flexural stiffness in longitudinal direction, in.-lb
D_{22} Flexural stiffness in circumferential direction, in.-lb
D_{33} Torsional stiffness, in.-lb
$\bar{Z}_r$ Distance from centroid of ring to middle surface of shell; positive if ring lies on external surface of shell, in.
$\bar{Z}_s$ Distance from centroid of stringer to middle surface of shell; positive if stiffener lies on external surface of shell, in.
ψ_r Indicates whether rings are external or internal to the skin surface; -1 if internal, $+1$ if external
ψ_s Indicates whether stringers are external or internal to the skin surface; -1 if internal, $+1$ if external
ϵ Reads "contained in"

Index

U.S. GOVERNMENT PRINTING OFFICE : 1967 O - 273-140

"The aeronautical and space activities of the United States shall be conducted so as to contribute . . . to the expansion of human knowledge of phenomena in the atmosphere and space. The Administration shall provide for the widest practicable and appropriate dissemination of information concerning its activities and the results thereof."

—NATIONAL AERONAUTICS AND SPACE ACT OF 1958

NASA TECHNOLOGY UTILIZATION PUBLICATIONS

These describe science or technology derived from NASA's activities that may be of particular interest in commercial and other non-aerospace applications. Publications include:

TECH BRIEFS: Single-page descriptions of individual innovations, devices, methods, or concepts.

TECHNOLOGY SURVEYS: Selected surveys of NASA contributions to entire areas of technology.

OTHER TU PUBLICATIONS: These include handbooks, reports, notes, conference proceedings, special studies, and selected bibliographies.

Details on the availability of these publications may be obtained from:

National Aeronautics and Space Administration
Code UT
Washington, D.C. 20546

Technology Utilization publications are part of NASA's formal series of scientific and technical publications. Others include Technical Reports, Technical Notes, Technical Memorandums, Contractor Reports, Technical Translations, and Special Publications.

Details on their availability may be obtained from:

National Aeronautics and Space Administration
Code US
Washington, D.C. 20546

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Washington, D.C. 20546