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Pressure Vessel Design

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FOREWORD

Pressure vessels have long been used for storage, industrial processing and power generation. To design these vessels, Codes were developed which consciously were restricted to giving short and simple formulas. While these formulas had as their basis the results of analytical and experimental investigations, they also reflected the judgement of practical experience. In the past two decades, these Codes and the horizon of the vessel designer had to be expanded because of the need to consider extremes of pressure and temperature as well as severe environmental conditions. These are present, for example, in nuclear power generation, space and ocean exploration and coal gasification. To continue to be able to provide vessels of maximum reliability under these conditions, the designer must have a thorough knowledge not only of the elementary and advanced concepts of strength of materials but also of dynamic analysis, heat transfer, material behavior and computers. It furthermore requires of him a continuing effort to keep abreast of the latest developments in all the areas affecting pressure vessel design.

The papers appearing in this special publication were presented at a two session symposium entitled "Pressure Vessel Design", held at the 1982 PVP Conference and Exhibition at Orlando, Florida. The purpose of the symposium was to present a broad spectrum of present activities in the area of pressure vessel design. To this extent, a large number of experts were invited to discuss their current work. Such topical areas as ASME Code analysis and design,PVRC research, buckling analysis and design criteria, effect of manufacturing tolerances, high temperature and pressure design, and composite vessel construction were covered in the sessions and are detailed in this volume.

In the initial paper, J. R. Farr presents the background and development of design formulas and methods used to calculate the minimum required thickness of pressure vessel components according to the ASME Boiler and Pressure Vessel Code, Section VIII, Divisions 1 and 2. Included is a review of some of the new procedures and design methods which have recently found their way into the Code. Much of this latter material is due to the work done by the Pressure Vessel Research Committee of the Welding Research Council. In his paper, W. L. Greenstreet gives a brief description of programs now in progress in the PVRC Design Division. A sketch of the origin, purpose and organization of PVRC is also presented.

That the role of the design engineer does not consist of design and analysis only is the premise of R. F. Reedy's paper. The main thrust of his argument is that since the engineer is the main link between the owner and fabricator and constructor to assure that the final structure meets the desired criteria, it is necessary for the designer to be involved in material selection, fabrication details, selection of appropriate nondestructive examinations, and complete design control and verification.

The next group of papers involves the buckling of vessels and components. G. D. Galletly examines the provisions of several Codes for preventing buckling in unstiffened cylindrical shells subjected to external pressure or axial load or a combination of these loads. Often, the difference between theoretical and experimental predictions is due to the presence of initial imperfections. In his paper, A. L. Citterly examines the applicability of ASME manufacturing tolerance guidelines to analysis requirements and presents a method of incorporating the tolerances in imperfection analyses. The next paper is by C. R. Steele and G. V. Ranjan, who performed a study to determine the effect of a ring stiffner attached at the knuckle region of a torispherical head on a cylindrical vessel on the critical pressure at which circumferential wrinkling occurs. They conclude that one or more discrete rings are not effective in increasing the critical pressure. In the final paper of this group, the Southwell method for predicting buckling loads is applied by L. H. Sobel to test results on elbows loaded by an in-plane closing bending moment. The major conclusion is that the method is valid for the components examined by him and also for a certain broader class of elastic or plastic nonlinear buckling (geometric collapse) problems.

The effect of bends is introduced into a piping system by using the flexibility factor concept. In their paper, R. Natarajan and S. Mirza present a finite element scheme with a doubly curved shell element which can account for variations in thickness. They show that for in-plane moment loading, a normal thickness variation has a significant effect on the flexibility factors for commonly used 90° bends.

Recently, a great deal of attention has been focused on vessels used in high pressure applications. This generally involves a heavy wall or layered type of construction. H. C. Rauschenplat presents a discussion of multiwall construction, which is created by thermally shrinking individual cylinders over each other. This method of construction has been incorporated into Section VIII of the ASME Code. Fatigue failures in thick-walled pressure vessels can occur at the OD as a result of machined discontinuities or of strengthening by the autofrettage process. In their paper, J. A. Kapp and S. L. Pu present conservative estimates of fatigue life which can be useful in examining designs for failure by fatigue crack initiation. Oftten, several types of construction are proposed for a particular process, and in order to find the optimum one, one needs to perform a comparison study. In the paper by T. R. Tauchert and D. C. Leigh, this is done for four alternative designs (single wall, total layered, prestressed concrete and prestressed cast iron) for a large diameter coal gasification reactor.

Reactor vessels for commercial-size LFMBR plants are quite large and necessarily thin-walled. As such they present the designer with problems in providing a balanced design to accommodate seismic, design basis accident and thermal loads. R. W. Seidensticker et al. present a comprehensive set of scoping calculations in order to guide the vessel designer in subsequent design iterations. With reference to present day nuclear power plants, a problem of concern is the degradation of mechanical properties due to irradiation. The effect of the latter on the crack-arrest toughness of low-copper base plate steels is the topic of a paper by C. W. Marschall et al. Their results indicate a modest temperature shift, in agreement with that ovserved in the Charpy V-notch impact energy curve and with that predicted by NRC Reg. Guide 1.99.

The last group of papers involves the application of structural plastics in pressure vessel construction. The lead-off paper is by G. E. O. Widera and D. L. Logan, who present a uniformly valid set of first approximation shell equations. These equations are applicable to cylindrically shaped vessels having a construction which is anisotropic and nonhomogeneous in the thickness direction. Typical example calculations are given for laminated cylinders. Next, S. G. Ladkamy discusses analysis and design considerations for large, high performance, prestressed aluminum vessels which are circumferentially reinforced with fiberglass epoxy or pultruded polyester. Included in his analysis are topics such as the prestressing pressure, the stability of the stiffened aluminum liner, the burst pressure and a controlled failure mode in the circumferential direction to improve the probability of a leak before burst condition. This paper is followed by one by U. Yuceoglu and D. Updike who discuss the elasto-static interaction problem of two closely spaced initial delamination flaws in a multi-layer pressurized cylindrical shell. They show that there exists an interaction length for the adhesive ligament beyond which no interaction occurs between the two flaws. Finally, A. V. Singh et al. report on the development of an annular finite element for the elastic analysis of sandwich shells. Their analysis involves the use of an improved shell theory which takes into account the effects of rotatory inertia and transverse shear deformation. They show that their element is a simple but powerful one from the point of view of computer time and accuracy of results.

It is impossible in a two session symposium to give a complete survey of all present day activities in pressure vessel design. Therefore, the reader of this special publication who wishes further information on this topic may want to consult the following books:

"Pressure Vessel and Piping Design Technology, 1982 – A Decade of Progress." Eds. S. Zamrik and D. Dietrich, ASME, New York, 1982.

"Pressure Vessels and Piping: Design and Analysis – A Decade of Progress: Volume One– Analysis," Eds. G. J. Bohm, L. C. Hsu, R. L. Cloud, D. H. Pai, R. F. Reedy, ASME, NEW York, 780 pp., 1972.

"Pressure Vessels and Piping: Design and Analysis – A Decade of Progress: Volume Two – Components and Structural Dynamics," Eds. G. J. Bohm, L. C. Hsu, R. L. Cloud, D. H. Pai, R. F. Reedy, ASME, New York, 812 pp., 1972.

as well as these other publications:

"Advances in Design Analysis Methodology for Pressure Vessels and Piping," Eds. C. E. Pugh and B. C. Wei, PVP-56, ASME, New York, 1982.

"Aspects of Fracture Mechanics in Pressure Vessels and Piping," Eds. S. G. Sampath and S. S. Palusamy, PVP-58, ASME, New York, 1982.

"Flow Induced Vibration Design Guidelines," Ed. P. Y. Chen, PVP-52, ASME, New York, 143 pp., 1981.

"Inelastic Behavior of Pressure Vessel and Piping Components," Eds. T. Y. Chang and E. Krempl, PVP-PB-028, ASME, New York, 175 pp., 1981.

"Stress Indices and Stress Intensification Factors of Pressure Vessel and Piping Components," Eds; R. W. Schneider and E. C. Rodabaugh, PVP-50, ASME, New York, 164 pp., 1981.

"Design, Inspection and Operation of High Pressure Vessels and Piping Systems," Ed. J. R. Sims, PVP-48, ASME, New York, 164 pp., 1981.

"Simplified Methods in Pressure Vessel Analysis," Ed. R. S. Barsoum, PVP-PB-029, ASME, New York, 127 pp., 1978.

"Pressure Vessels and Piping – Computer Program Evaluation and Qualification," Ed. D. E. Dietrich, PVP-PB-024, ASME, New York, 122 pp., 1977.

"Composites in Pressure Vessels and Piping," Eds. S. V. Kulkarini and C. H. Zweben, PVP-PB-021, ASME, New York, 184 pp., 1977.

In conclusion, the editor wishes to thank each of the authors for their valuable contributions. The successful completion of this special publication is entirely due to their effort and to that of the reviewers, who unselfishly provided critiques and advice. With regards to the latter, special thanks are extended to J. Hagstrom and O. Byazid.

G. E. O. Widera

CONTENTS

DESIGN OF PRESSURE VESSELS ACCORDING TO THE ASME BOILER AND PRESSURE VESSEL CODE, SECTION VIII, DIVISIONS 1 and 2

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ABSTRACT

A discussion is given of the background and development of design formulas and methods used to calculate minimum required thickness of pressure vessel components, according to the ASME Boiler and Pressure Vessel Code, Section VIII, Divisions 1 and 2[1,2]. Comparisons are made of the similarities and differences in theories, criteria, allowable stresses, and design formulas between Division 1 and Division 2[1]. Also, a review is given of some of the new procedures and design methods which have recently been added to the Code.

INTRODUCTION

The original ASME Boiler Construction Code was issued as the 1914 Edition on February 13, 1915. Although several other sections were issued in the following years, it was not until 1925 that Section VIII, Unfired Pressure Vessels, was issued. This section continued as the construction code for pressure vessels until 1968. In 1968, the title was changed to Section VIII, Division 1, Rules for Construction of Pressure Vessels, and Section VIII, Division 2, Alternative Rules for Construction of Pressure Vessels, was added. Each division was issued as a separate volume because it is administratively handled as though it were an individual section. All of the rules of Division 1, as a whole, or all of the rules of Division 2, as a whole, shall be used individually without intermixing of the rules of the two divisions.

DESIGN PHILOSOPHY AND STRENGTH THEORIES

The original design philosophy of the ASME Code from the issuance of the 1914 Edition of the Boiler Construction Code and the 1925 Edition of the Unfired Pressure Vessel Code was one where the primary membrane stress, or maximum direct stress as it is called, based on the maximum stress theory, did not exceed an allowable tensile stress value. The analysis was limited to consideration of internal pressure only. Since that time, however, many rules have been added and consideration is made of external pressure and other

1 The use of the words Division 1 and Division 2 are shortened terms for ASME Code sections given in References (1) and (2).

applied loadings. In most instances, a detailed stress analysis is required only to determine the adequacy of a component when no design formulas, rules, nor charts are given.

The design philosophy of Section VIII, Division 2, is considerably different from the philosophy of Section VIII, Division 1. The strength theory is the maximum shear stress theory. In reviewing the literature[3] when Section VIII, Division 2, was being developed, it was determined that both the maximum shear stress theory and the distortion energy theory give better results for predicting yielding and fatigue failure than the maximum shear stress theory which is used in Division 1. The maximum shear at a point is defined as one-half of the algebraic difference between the largest and the smallest of the three principal stresses. The theory assumes that yielding occurs when the maximum shear stress is equal to $S_y/2$. This is expressed in the following equation:

$$\tfrac{1}{2}(\sigma_{largest} - \sigma_{smallest}) = \tfrac{1}{2}S_y \text{ or } (\sigma_L - \sigma_S) \leq S_y$$

In Division 2, an appropriate factor of safety is applied to both tensile strength and yield strength data which replaces the value S_y.

In Section VIII, Division 1 construction, a detailed stress analysis is required only when necessary to meet the requirements of U-2(g) and UG-22. A stress analysis report may be requested by the Authorized Inspector for review. In contrast, for Section VIII, Division 2, a stress analysis report is required and must be certified by a Registered Professional Engineer experienced in pressure vessel design.

DESIGN CRITERIA

The design criteria of Section VIII, Division 1, is to provide formulas, curves, charts, and design procedures which set the minimum required thickness and maximum allowable working pressure (MAWP) for common pressure vessel components. These thicknesses and pressures are established on the basis that the primary stresses will not exceed the allowable tensile or compressive stress values. In establishing some of the design methods, consideration was also given to local and secondary stresses where these higher stresses are acceptable, consistent with experience. Although there is a list of minimum required loadings given in UG-22, design procedures are not given for most of the loadings. The amount of analysis and method of analysis is the responsibility of the Stampholder, subject to acceptance of the Authorized Inspector as described in U-2(g).

The design criteria of Section VIII, Division 2, is to provide some design formulas, charts, and rules for the more common configurations of shells and heads for design temperatures below the creep/rupture temperature range. Detailed stress analysis evaluations of actual stresses in complex geometries and/or with unusual loadings are required. Those calculated stresses are assigned to various categories which have different allowable stress values. The stress categories and sub-categories are:

A. Primary Stress

   1. General Primary Membrane Stress
   2. Local Primary Membrane Stress
   3. Primary Bending Stress

B. Secondary Stress

C. Peak Stress

The basic limits for the various categories are based on limit-design theory, assuming no strain-hardening, and applying an adequate factor of safety. The primary stress limit permits no yielding to occur while the primary-plus-secondary stress limit permits only that amount of yielding to occur which will "shakedown" to elastic action after a few cycles, (see Table 2 for a summary).

Division 2 recognizes fatigue as a mode of failure and, different from Division 1, it provides rules to design against fatigue failure. Rules are given which establish whether or not a fatigue evaluation is required. Operating experience under certain restricted conditions may be considered. When operating experience cannot be substantiated, certain design conditions are examined. If those rules are satisfied, no further fatigue evaluation need be made even though the vessel may be subjected to some cyclic operation. If none of the rules can be met, a cyclic analysis is required.

ALLOWABLE TENSILE STRESSES AND FACTORS OF SAFETY

In the early codes, allowable tensile stresses were set at fixed values as follows: 11,000 psi for steel plate which was stamped 55,000 psi; 10,000 psi for steel plate stamped less than 55,000 psi; and 9,000 psi for seamless shells. By the 1931 Edition, a uniform factor of safety of five, applied to the specified minimum ultimate tensile strength, was used. At that time, no factor of safety was applied to the yield strength. However, for most ductile steel, the yield strength was somewhere close to one-half of the tensile strength. Consequently, the factor of safety on yield strength was close to two and one-half. During World War II, some special provisions were added to the Unfired Pressure Vessel Code to permit the use of a factor of safety of four on tensile strength with certain kinds of construction. The factor of safety of five was continued through the 1949 Edition. However, with the successful experience in using the factor of safety of four, a new edition of the Code was issued in 1950 which reduced the general factor of safety on tensile strength to four.

With the 1952 Edition, the criteria for setting allowable tensile stresses was expanded to include a factor of 62.5 percent of the yield strength and as high as 90 percent yield strength for some materials, such as austenitic stainless steels and nickel alloys. At the same time, criteria for elevated temperature design were added. These consisted of a factor of safety applied to creep and to rupture data. The lowest value of these values was set as the allowable tensile stress. Two years ago, the factor of safety on yield strength was increased from 62.5 percent to 67 percent. The basis for establishing the allowable tensile stresses in Section VIII, Division 1, is given in Appendix P of that division and is listed in Table 1.

For Section VIII, Division 2, the allowable tensile stresses, or design stress intensity values, for all materials--except bolting material--is set by a factor of safety of three on tensile strength and 67 percent of the yield strength to as high as 90 percent yield strength for some materials, such as austenitic stainless steels and some nickel alloys. The basis for establishing the design stress intensity values for Section VIII, Division 2, is given in Appendix 1 of that division and is listed in Table 1. No design stress intensity values nor design rules have been established in the creep/rupture temperature range for Division 2. When a vessel is to be designed at a temperature which is above the highest temperature permitted for that material (as given in the tables of design stress intensity values) for restricted applications where fatigue is not a consideration, stresses may be established according to Code Case 1489. This Case uses allowable tensile stresses from Division 1 at temperatures where no stresses are given in Division 2.

ALLOWABLE COMPRESSIVE STRESSES FOR EXTERNAL PRESSURE AND FOR AXIAL LOADING ON A CYLINDRICAL SHELL

Rules for the design of vessels subjected to external pressure first appeared in the 1934 Edition of Section VIII. They were very smilar to the present rules but were limited to a few materials. This early Code also contained rules for stiffening rings and permissable out-of-roundness of a cylinder under external pressure. In the 1943 Edition, a special figure and design method was added which permitted the calculation of the maximum allowable external pressure for tubes and pipes. This figure remained in Division 1, as Figure UG-31, until the 1974 Edition; at which time, it was removed due to lack of background data to substantiate its applicability to all of the materials which are permitted in Subsection C of Division 1. At the request of the ASME Code Committee, the Pressure Vessel Research Committee of the Welding Research Council initiated a test program to review and retest tube specimens according to the rules of Fig. UG-31 at Southwest Research Institute. This report[4] is being reviewed by the PVRC to determine if the old figure should be reinstated or revised. In the 1952 Edition, rules were added to determine the allowable compressive stress for cylinders under axial loading. This method utilizes the same design curves and similar design rules to those which are used to set the maximum allowable external pressure.

At the present time, exactly the same rules and factors of safety are used in Division 1 and Division 2 to establish both the maximum allowable external pressure for cylinders and spheres and to determine the maximum allowable axial compressive stress in a cylinder. These are summarized in Table 3. Due to the nonlinear aspects of the criteria used to establish the allowable compressive stresses, the Code uses a trail-and-error type of solution. A trial thickness is selected for the shell or head; and by following the appropriate design rules, an allowable compressive stress or external pressure is determined. If the result is not suitable, the thickness is increased or stiffening rings are added to cylindrical shells until an acceptable answer is determined.

OTHER DESIGN FORMULAS AND PROCEDURES

CYLINDRICAL, SPHERICAL, AND CONICAL SHELLS UNDER INTERNAL PRESSURE

Both Division 1 and Division 2 contain formulas for calculating the minimum required thickness and maximum allowable working pressure of cylindrical, spherical, and conical shells under internal pressure. The formulas are similar in form except that the Division 2 formulas are adjusted, due to the maximum shear stress effect, instead of the maximum stress effect. Also, the Division 1 formulas contain a weld joint efficiency factor, E, while the Division 2 formulas do not. Mandatory weld examination requirements in Division 2 result in a weld joint efficiency of E = 1.0; consequently, no term is included because full strength is assumed.

FORMED HEADS UNDER INTERNAL PRESSURE

For formed heads under internal pressure, there is a considerable difference between Division 1 and Division 2. In Division 1, formulas are given for hemispherical, ellipsoidal, torispherical, conical, and toriconical heads. Each type of head may be calculated by a design formula and procedure which permits variations in the geometry, such as a 2.1:1 ellipsoidal head instead of the usual 2:1 ellipsoidal head. Division 2, on the other hand, is limited to formulas for only hemispherical, torispherical, and 2:1 ellipsoidal formed heads. The Division 2 formed heads which have design formulas also require a substantial skirt at the head-to-shell juncture to keep the fatigue and discontinuity stresses at an acceptable level. For other types of formed heads for Division 2 construction, a complete stress analysis is required.

CYLINDRICAL AND SPHERICAL SHELLS UNDER EXTERNAL PRESSURE

As was previously noted, the method for designing cylindrical and spherical shells under external pressure is exactly the same in Division 1 and Division 2.

FORMED HEADS UNDER EXTERNAL PRESSURE

Although Division 2 limits the design rules for formed heads under external pressure to hemispherical, ellipsoidal, and torispherical heads, while Division 1 contains not only those but also conical and toriconical heads, both apply the same sort of design method. That is, the radius is adjusted depending upon the type of head. With this adjusted radius, the rules for the spherical shell under external pressure are followed.

FLAT HEADS AND COVERS

The design formulas for calculating the minimum required thickness of flat heads is identical in both Division 1 and Division 2 except for the weld joint efficiency term, E, in the Division 1 formula. However, in the input information to the formulas, the values of c and, of course, S are different in each division. When these are used in the formula, the differences in the minimum required thicknesses are close, although the thickness required for Division 1 is slightly more than for Division 2.

OPENINGS AND REINFORCEMENT

Both divisions determine the reinforcement requirements by the area replacement method. That is, an area equal to the minimum required thickness times the opening diameter have to be replaced by excess area within the reinforcement limits. However, one difference between the two divisions is that Division 2 requires a second calculation for all nozzle which examines 2/3 of the required area within a closer limit. Division 1 requires this examination only when the openings are large, relative to the shell. In this second calculation, the horizontal limits in Division 2 are set by a formula which is determined by the beam on an elastic foundation theory.

Division 2, however, has several requirements which are different from those in Division 1. First, the maximum ratio of opening diameter divided by the shell diameter, d/D, is limited to 0.5. Division 1 has no limit. Second, Division 2 contains a special calculation procedure which can permit a nozzle configuration which does not give 100 percent replacement of area. This method has several provisions which must be met before the procedure may be used. Also, since Division 2 may require a fatigue analysis, there are special provisions for establishing stress concentration factors in the region of a nozzle.

SOME RECENT DESIGN PROCEDURE ADDITIONS

In Division 1, a design procedure has been added to design reverse flanges. Its use is limited to a diameter of opening divided by shell diameter of 2 because results become conservative. In a companion procedure, design rules are now included for the calculation of a flat head with a large, centrally-located opening which uses a similar analysis procedure. Rules for dimpled and embossed assemblies have been incorporated. Depending upon the method of fabrication, the maximum allowable working pressure is established either by a proof-test or by calculations. The nozzle reinforcement rules have been rewritten; several additional configurations have been added which permit nozzle which are welded from the inside only, under limited conditions of loading, and nozzle-to-shell clearance.

The addition of design procedures to Division 2 has been limited because that division has provisions to do a detailed analysis on most configurations which are not given.

REFERENCES

1. ASME Boiler and Pressure Vessel Code, ANSI/ASME BPV-VIII-1, Section VIII, Division 1, Rules for Construction of Pressure Vessels, 1980 Ed., ASME, New York, July 1980.

2. ASME Boiler and Pressure Vessel Code, ANSI/ASME BPV-VIII-2, Section VIII, Division 2, Alternative Rules for Construction of Pressure Vessels, 1980 Ed., ASME, New York, July 1980.

3. Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2, ASME, New York, 1969.

4. Tschoepe, E. and Maison, J.R., "The External Pressure Collapse Tests of Tubes," SwRI Project No. 03-5113-001, January 1981, Southwest Research Institute, San Antonio, TX. (A private report to the PVRC.)

TABLE 1 - MULTIPLYING FACTORS ON MATERIAL PROPERTIES TO DETERMINE MAXIMUM ALLOWABLE STRESS AND DESIGN STRESS INTENSITY VALUES

Code Sections	Notes	Tensile Strength		Yield Strength		Creep Rate of .01% in 1,000 hrs.	Stress to Rupture in 100,000 hours	
		Minimum Spec. @ Room Temp.	Strength @ Temp. [8]	Minimum Spec. @ Room Temp.	Strength @ Temp.	Average	Average	Minimum
Section VIII Division 1								
Vessels	3	1/4	1/4	2/3	2/3	1.00	0.67	0.80
	4	1/4	1/4	2/3	0.90	1.00	0.67	0.80
Bolting	1	1/4	1/4	2/3	2/3	1.00	0.67	0.80
	2	1/5	1/4	1/4	2/3	1.00	0.67	0.80
Division 2								
Vessels	3	1/3	1/3	2/3	2/3	--	--	--
	4	1/3	1/3	2/3	0.90	--	--	--
Bolting	5	1/4	1/4	2/3	2/3	--	--	--
	6	1/5	1/4	1/4	2/3	--	--	--
	7	---	---	1/3	1/3	--	--	--

NOTES FOR TABLE 1

1 - Minimum for all bolting except heat treated bolting

2 - Minimum for all heat treated bolting

3 - Minimum for a-1 materials except bolting

4 - Increase in minimum permitted for austenitic stainless steels and nickel in nickel alloys except bolting

5 - Minimum for Appendix 3 bolting except heat treated material

6 - Minimum for Appendix 3 heat treated bolting

7 - Minimum for Appendix 4 bolting

8 - Values in this column must be multiplied by 1.1

TABLE 2 - BASIC STRESS INTENSITY LIMITS FOR VARIOUS STRESS CATEGORIES, DIV. 2

Stress Intensity Category	Allowable Value	Factor Based On Yield Strength	Factor Based On Tensile Strength
General Primary Membrane (P_m)	S_m	$\leq 2/3\ S_y$	$\leq 1/3\ S_u$
Local Primary Membrane (P_L)	$1.5\ S_m$	$\leq S_y$	$\leq 1/2\ S_u$
Primary Membrane Plus Bending ($P_m + P_b$)	$1.5\ S_m$	$\leq S_y$	$\leq 1/2\ S_u$
Primary Plus Secondary ($P_L + P_b + Q$)	$3\ S_m$	$\leq 2\ S_y$	$\leq S_u$

TABLE 3 - MULTIPLYING FACTORS ON PROPERTIES TO DETERMINE ALLOWABLE COMPRESSIVE STRESS

Condition	Critical Buckling Stress	Yield Strength	Creep Rate of 0.01% in 1,000 Hrs.	Allowable Tensile Stress
Axial Load on Cylinder	1/4	1/2	1.00	1.00
External Pressure on Cylinder	1/3	1/3	0.67	1.00
External Pressure on Sphere	1/4	1/4	0.50	1.00

<u>Note</u>

Factors have been incorporated into the external pressure material's charts.