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346

Tentative Structural Design Basis for Reactor Pressure Vessels and Directly Associated Components

(Pressurized, Water Cooled Systems)

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TENTATIVE STRUCTURAL DESIGN BASIS
FOR REACTOR PRESSURE VESSELS AND
DIRECTLY ASSOCIATED COMPONENTS

(PRESSURIZED, WATER COOLED SYSTEMS)

* * * * *

1 December 1958 Revision

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TENTATIVE STRUCTURAL DESIGN BASIS FOR REACTOR PRESSURE VESSELS
AND DIRECTLY ASSOCIATED COMPONENTS
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1 December 1958 Revision

ADDENDUM NO. 1 - 27 February, 1959

Page 4, paragraph 3.1.2 - Revise last sentence to read, "Nozzle connections are considered in paragraph 6.6".

Page 28, paragraph 5.1.1 - Add to the end of this paragraph the following:

"The use of this paragraph will frequently result in the calculation of a permissible wall thickness thinner than that currently permitted by the ASME Boiler and Pressure Vessel Code. In order to avoid this violation of ASME Code requirements in cases where no appreciable advantage is obtained, the Equipment Specification or the Purchase Order may impose the additional limitation that wall thicknesses used in the cylindrical shell or heads of the reactor vessel itself shall not be less than those required by the applicable portion of the ASME Code".

Page 29, paragraph 5.1.4 - Revise to read:

"5.1.4 Thermal Stresses. All thermal stresses shall be considered as transient conditions and treated in accordance with paragraph 5.2. In addition, thermal stresses which remain constant during any normal steady-state power operation (hereafter called "steady-state" thermal stresses) shall be limited by the following requirements:

(1) The distortion limits of paragraph 1.5 apply.

(2) Whenever the combination of primary-plus-secondary stress, as defined and limited in paragraph 5.1.3, with "steady-state" thermal stress, including the effects of structural discontinuities but not stress concentrations, results in a stress intensity which exceeds the value of S_p in Table 5-1 at the operating temperature, justification for this stress level must be provided showing that no deleterious effects, such as those due to distortion, creep, or stress-rupture, will occur in either the new or irradiated condition of the material".

Page 40, paragraph 6.6 - Revise to read:

"6.6 Nozzle Connections and Openings. Piping connections and openings in the vessel proper (other than the upper head) shall preferably conform to one of the configurations shown in Appendix C. Stress concentration values for these configurations are given and discontinuity effects are known to be acceptable. If other configurations are used, evidence of their adequacy must be furnished.

Page 42, paragraph 8 - Delete paragraph 8 (entire page).

Page 127, paragraph B.2.3 - Add the following to the end of this paragraph:

"The use of S_u as the allowable value of S_s does not mean that no safety factor is being used. As described in paragraph A.1.8.3, no operating points on the fatigue diagram can ever occur to the right of S_b , so that S_s merely establishes the slope of the "safe limit" line in the useful range. This point is illustrated in Figure 5.2-1b)".

Page 30, Figure 5.2-1b - Revise this Figure as follows:

ERRATA

Page 58, paragraph A.4.1 - In next to last line "equation (10)" should read "equation (19)".

Page 103, paragraph A.8.5.2 (c)

$$\frac{M_{\phi}}{M/a} \quad \text{should read} \quad \frac{M_{\phi}}{M/a\beta}$$

$$\frac{N_{\phi}}{M/a^2} \quad \text{should read} \quad \frac{N_{\phi}}{M/a^2\beta}$$

$$\frac{M_x}{M/a} \quad \text{should read} \quad \frac{M_x}{M/a\beta}$$

$$\frac{N_x}{M/a^2} \quad \text{should read} \quad \frac{N_x}{M/a^2\beta}$$

Page 91, paragraph number "A.7.7.2" should read "A.7.2.2).

Page 115, paragraph A.8.11.3 - Reference "(See A.5.4.3)" should read "(See A.5.5.2)".

Page 57, Figure A.3-4 - Add a note as follows:

- + values used for stresses at outer surface
- values used for stresses at inner surface

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TENTATIVE STRUCTURAL DESIGN BASIS FOR REACTOR
PRESSURE VESSELS AND DIRECTLY ASSOCIATED COMPONENTS
(PRESSURIZED, WATER COOLED SYSTEMS)

1 December 1958 Revision

1. INTRODUCTION

1.1 Object. The structural design basis outlined herein establishes general loading assumptions, performance and design criteria, and essential minimum limitations for assuring acceptable structural reliability and safety of reactor pressure vessels and directly associated components under the individual and designated combinations of loading requirements specified for a particular installation (except for shipboard shock and vibration to which reference should be made in the applicable Equipment Specification).

1.2 Scope

1.2.1 For purposes of this document the term "reactor pressure vessel and directly associated components" shall be interpreted to cover (a) the pressure containing shell, complete with top and bottom heads, including flanges and means for fastening and sealing, reinforcements, openings and nozzles (including control drive mechanism housings) penetrating the shells, heads and closures; (b) all structures internal to the pressure retaining members with the exception of (1) those containing fissionable or poison materials and (2) those structures which are integral with structures containing fissionable or poison materials; and (c) external structures permanently attached to the vessel, such as those required for handling, operation, servicing and support. Generally speaking, the limits of the vessel at external piping connections shall be taken as the first circumferential weld beyond the limits of nozzle reinforcement; however, where a transition weld between ferritic and austenitic materials is involved, it shall be the first circumferential weld beyond the transition weld. The exact point of demarcation between the reactor vessel support and the foundation will be specified in the Equipment Specification covering each application.

1.2.2 This structural design basis was prepared for application in the Naval Reactor Program. In such application, the provisions herein are invoked by reference to this document by a Military Specification for a particular component. That specification also invokes detailed requirements with respect to design objectives and requirements, materials, fabrication, inspection, and testing which are compatible with this design basis. Application of this structural design basis to components not considered by such a specification, and for which such compatible detailed requirements are not otherwise prescribed, is not recommended.

1.3 Survey of Design Procedure. A narrative description of the design procedure contained herein is presented in Appendix A in order that the basic philosophy and method may be described without the encumbrance of technical details. It is advisable for the reader to review that description prior to applying the requirements of this design basis.

1.4 Except when superseded by information contained herein, the design of reactor pressure vessels shall, as a minimum requirement, comply with the latest edition (including revisions, addenda, and applicable cases, such as Case No. 1234) of Section VIII of the ASME Boiler and Pressure Vessel Code ("Unfired Pressure Vessels"), as pertaining to a vessel containing lethal substances. This requirement does not necessarily constitute an endorsement of the design procedures contained therein, and satisfaction of Code requirements or the following of Code procedures are not considered to be complete evidence of the suitability of the design.

In addition to the other requirements contained herein, construction to the following Code paragraphs is specifically prohibited: U-2(d); UG-6(d); UG-10; UG-19(c); UG-23(d); UG-25(d),(e); UG-27(e); UG-28(g),(i); UG-32(g),(h),(i),(k),(m),(r); UG-33(f),(g); UG-43(c),(d),(g); UG-46; UG-47 through UG-50; UG-83; UG-101; UW-9(e),(f); UW-15(c); UW-16(e); UW-17; UW-18; UW-19; UW-52; Part UR; Part UB; UCL-11(c); UCL-23(c); UCL-25(b); UCL-46.

1.5 As a limitation on dimensions and clearances is necessitated by safety and operational considerations, the specifications on certain components may establish (a) fabrication and installation dimensions and tolerances, and (b) maximum operational deformation and/or resulting clearances for maintained or transient conditions. These dimensions, tolerances and clearances must not be exceeded under any condition or combination of emergency operation.

1.6 The general design procedures and allowable stress values contained herein are predicated upon minimum specification properties. (see paragraph 4). They have been adjusted to take into account operating temperature, any normal variation in materials properties, and surface irregularities characteristic of mill finish on ferritic and austenitic steel plate. No allowance is made for stress-corrosion fatigue or fretting fatigue; these problems are not considered to be of substantial significance with austenitic steel linings in contact with main coolant except possibly where local corrosion cells or couples or local fretting may occur. The latter conditions shall be avoided where practicable, particularly where failure would allow leakage of main coolant to the atmosphere. No allowances have been made for inaccuracies in stress evaluation or in the assumed conditions of operation. Therefore, conservative conditions must be assumed, and empirical or limited solutions for calculation of stress must be explored as to the maximum range of error they involve, which then establishes the correction factor to be applied. Areas for which acceptable theoretical stress analysis is lacking must be subjected to experimental stress analysis.

1.7 Waiver Conditions. Undoubtedly, it will be found that, in certain cases, the optimum design of certain components from a functional standpoint will require use of higher allowable stresses, new materials, or structural requirements other than those specified herein. In event that need for a waiver from the requirements contained herein becomes apparent, a written

request for such waiver shall be submitted to the Naval Reactors Branch, through applicable channels, with sufficient supporting information upon which to base a decision. Such procedure is not intended to discourage requests for waiver of these requirements, but rather to maintain an adequate control over the design until such time as design requirements and knowledge become well established.

1.8 Experience Factors. It is recognized that this structural design basis requires an extensive amount of detailed analytical and experimental analysis. However, it is not the intent of this document to require re-evaluation of configurations where detailed experience has been obtained and detailed analytical or experimental evidence exists. When such situations occur, vendors are invited to submit the applicable evidence to the Naval Reactors Branch through applicable channels for a ruling as to whether or not such evidence is sufficient to permit partial waiver of the detailed procedures required herein.

2. LOADING CONDITIONS

2.1 Loading conditions to be considered in the design of reactor pressure vessels shall include the following:

- a. Internal pressure
- b. External pressure
- c. Mechanical forces (non-pressure loadings), including effects of hull distortions, static forces due to fluid flow, and internal and external attachments.
- d. Pipe reactions due to thermal expansion of piping system and vessels to which piping is attached.
- e. Thermal conditions resulting from steady-state heat flow at any level of power operation.
- f. Thermal conditions due to internal (gamma) heat generation.
- g. Thermal conditions resulting from normal power changes.
- h. Thermal conditions resulting from abnormal power changes.
- i. Thermal conditions resulting from normal start-up and shutdown.
- j. Thermal conditions resulting from emergency shutdown ("scrams") and subsequent recovery.
- k. Thermal conditions resulting from emergency cooling conditions. The requirements contained herein pertain only to those emergency cooling conditions from which recovery to operational condition is required by component specification. No requirements are established regarding the safeguards problem where operational recovery is not required or expected.
- l. Thermal effects on components constructed of materials with different thermal expansion coefficients.
- m. Vibration due to flow effects, which originate either within the reactor or are transmitted to it by attachments.

3. GENERAL STRUCTURAL REQUIREMENTS

3.1 Definition of Terms

3.1.1 Design Pressure. The design pressure will be as specified in the equipment specification. Ordinarily such design pressure will be approximately

10-25% in excess of the normal steady-state operating pressure and perhaps 5-10% in excess of the maximum estimated surge pressure. Ordinarily, the first set safety or relief valve in the primary system will be set at or below design pressure, and the maximum pressure on the system (neglecting nuclear and chemical incidents) will be limited to 10% above design pressure, with relief devices open and blowing at rated capacity, as specified in Section VIII of the ASME Code. When this condition applies, it shall not be necessary to analyze for over-pressure effects except as qualified in paragraph 3.2.3.

3.1.2 Structural Discontinuities. The term "structural discontinuities" refers to major changes in shape, which are largely inherent in the design of a vessel and which lead to stress intensifications involving relatively large areas or volumes of material; examples of such discontinuities are head-to-shell and shell-to-flange junctions, the juncture between two shells of differing thickness, and reducing sections. Stress values resulting from the consideration of ligaments are specifically discussed in paragraph 5.5.2. Nozzle connections are not considered to be structural discontinuities.

3.1.3 Stress Concentrations. The term "stress concentrations" will refer to those structural discontinuities which lead to stress or strain intensification, but which involve relatively small areas or volumes of material such that no deformation of the structure as a whole could possibly be involved. Examples of "stress concentrations" are openings and nozzle connections, small fillet radii, small attachments, incomplete weld penetrations, etc., which are often the result of poor design and fabrication practices. It is emphasized that the use of this structural design basis is predicated upon elimination or minimization of this type of discontinuity, insofar as practicable, through the use of the best possible design, fabrication and inspection techniques. Recommended methods and references for evaluation of stress concentration factors are given in paragraph A.7. Theoretical stress concentration factors shall be used, but in no case need the stress concentration factor be taken as greater than 5.0. (See paragraph A.7.7).

3.1.4 Operational Cycle. The term "operational cycle" means the initiation and establishment of a new operating condition and the return to the original operating conditions.

3.1.5 Primary Stress. A direct or shear stress developed by the imposed loading which is necessary to satisfy only the simple laws of equilibrium of external and internal forces and moments. Note that both structural discontinuities and stress concentrations are neglected.

3.1.6 Secondary Stress. A direct stress or a shear stress developed by the constraint of adjacent parts or by self-constraint of a structure. Note that stress concentrations are neglected.

3.1.7 Stress Intensity. The term "stress intensity" is used herein as a short form of the term "equivalent intensity of combined stress". This quantity is defined herein as the difference between the algebraically largest principal stress and the algebraically smallest principal stress at a given location. The stress intensity is numerically equal to twice the maximum shear stress and is directionless.

3.2 Design Criteria

3.2.1 Stress Limits. The stress quantity upon which limitations are established herein is the "stress intensity". (See paragraph 3.1.7).

3.2.2 Criteria for Evaluating the Effect of Cyclic Stresses. The combined effect of steady-state and cyclic stresses shall be evaluated in accordance with a modification of the Goodman diagram, as explained and illustrated in paragraph 5.2 and in Appendix A. The cumulative effect of various loading cycles will be evaluated in accordance with Miner's hypothesis, as explained in paragraph 5.2 and in Appendix A.

It should be noted that under certain combinations of steady state and cyclic loading there is a possibility of distortions occurring in the form of a ratchet. That is, the deformation increases with time by a near equal amount for each cycle. Although the procedures presented herein do not explicitly consider this phenomenon, the requirements of paragraph 1.5 shall be specifically considered with regard to this problem. Appendix D considers one specific case of this phenomenon.

3.2.3 Transient Over-Pressure Limits. Relief valve settings, as described in paragraph 3.1.1, are not necessarily sufficient to protect the components from the effects of pressure transients. In addition to the fatigue criteria of paragraph 5.2, the requirements of paragraph 1.5 shall be specifically considered with respect to this problem.

3.3 Stress Analysis. A detailed stress analysis of all major structural components shall be prepared covering both primary and secondary stresses under all significant conditions of operation, including stress concentration effects where applicable, and in sufficient detail to permit independent checking. Where practicable to do so, the results of stress analysis shall be checked for continuity of loadings and deformations at junctions and for compatibility of deformations of a structural component (plate, shell, or flange, etc.) with loadings imposed on the component. Examples of stresses to be investigated include (1) stresses at inside and outside of vessel wall due to internal pressure and thermal effects, (2) stress at discontinuities such as head-to-shell joints, changes in section and openings, (3) stresses in flanges, seals, bolts and other elements of closure assemblies, (4) stresses due to external loads such as pipe and foundation reactions, and (5) effects of stress concentrations such as head knuckles, fillet radii, etc., if any. The following notes, conditions, and requirements are listed for guidance in making these analyses:

3.3.1 Certain of the allowable stresses permitted in this structural design basis are such that the yield strength of the material is exceeded, particularly for combinations of steady-state effects and transient thermal conditions. In making comparison with these allowable stress values, stresses will be calculated on an elastic basis. The stress limits S_a cited for fatigue in paragraph 5.2, are based on strains being proportional to the stresses calculated on an elastic basis. When the calculated stress range exceeds twice the yield strength, the stress value to be compared with paragraph 5.2 shall include the effects of local increases in strain due to plastic flow. Unless this is done, the allowable alternating stress for fatigue loadings shall be no greater than the tabulated values of S_y .

3.3.2 The variation of the stress intensity shall be evaluated as a function of time for one complete operational cycle. Typical complete operational cycles are:

- 3.3.2.1 Start-up and shutdown, including both pressure and thermal effects.
- 3.3.2.2 Variation of reactor power through full range and return.
- 3.3.2.3 Scram and scram-recovery.
- 3.3.2.4 Emergency Cooling cycle.
- 3.3.2.5 Flow-induced vibration.

3.3.3 All conditions outlined in paragraph 2.1 must be considered from the standpoint of fatigue damage.

3.3.4 For purposes of design, the various transient conditions of plant operation will be divided into three categories, and the resulting stresses considered in accordance with paragraph 5.2.7. The provision of temperature and pressure limits in the classes as defined below is for the purpose of standardization of operating requirements and structural evaluation procedures and these have no direct structural significance. More detailed requirements concerning the nature of these loading cycles in terms of power change, pressure range, temperature range, time, etc. (as appropriate) will be furnished in the equipment specification or in supplementary data furnished to the manufacturer.

(Note: If the operating requirements for a specific project differ from those considered herein, paragraphs 3.3.4.1, 3.3.4.2, and 3.3.4.3 and Table 5-1 will be revised accordingly by the equipment specification to suit the requirements of paragraph B.2.4).

3.3.4.1 Class I - All conditions of plant operation during which the variation of system pressure and coolant temperature ($^{\circ}\text{F}$) is less than plus or minus 15% of the values existing at the start of the transient. This category is intended to cover all operations incident to normal power production. The allowable stress values, S_{aI} , in Table 5-1 assume that the total number of such operations result in no more than 100,000 cycles.

3.3.4.2 Class II - All start-up operations, other than those included in Class I or Class III. This category is intended to cover "warm" start-up operations. Unless otherwise prescribed in the equipment specification, it shall be assumed that all such start-up operations are from 150°F . Other operational conditions during which the system temperature remains above the minimum temperature prescribed for warm start-ups but below 115% of the full power operating temperature ($^{\circ}\text{F}$) may be included in this category if so prescribed by the equipment specification. The allowable stress values, S_{aII} , in Table 5-1, assume that the total number of operations result in no more than 2500 cycles.

3.3.4.3 Class III - All conditions of plant operation during which the coolant temperature is below the minimum temperature prescribed for warm start-ups (in paragraph 3.3.4.2, or in the equipment specification) or above 115% of the full power operating temperature. This category is primarily intended to cover "cold" start-up and emergency operations. The allowable

stress values, S_{allow} , in Table 5-1, assume that the total number of such operations result in no more than 500 cycles. Other operational conditions of infrequent occurrence may be included in this category if so prescribed by the equipment specification.

3.3.5 For purposes of obtaining plant operating limitations, the alternate procedure outlined in paragraph 5.2.8 may be followed.

3.3.6 The condition outlined in paragraph 2.1, item (m), shall be considered as a transient mechanical condition.

3.3.7 Calculation of thermal stresses in irregular sections (such as closure flanges, nozzle connections, etc.) will involve a determination of temperature distribution under various transient conditions. Whereas certain simplified analytical approaches to this problem may serve the purposes of preliminary design, for final design, the temperature distribution may have to be determined by means of analog or digital computers.

3.4 Materials. Data pertaining to properties and allowable stresses for certain commonly used materials are contained in paragraphs 4 and 5 following. If other materials are required in any structural component, written approval shall be requested from the Naval Reactors Branch for the use of such material and establishment of applicable stress limits. Such request should contain complete details concerning the composition and processing of the material, its intended use, and recommended allowable stress values for the usage in question based upon past experience or standard engineering practices. No materials shall be used without such approval.

3.5 Welding. Except as required below, all welding, including design details, procedure and inspection shall comply with the requirements of Military Standard MIL-STD-278 (SHIPS) "Welding and Allied Processes for Machinery for Ships of the USN", MIL-STD-22 (SHIPS) "Welded Joint Design", and such supplementary specifications as may be called for in the equipment specifications.

3.5.1 Pressure-Retaining Strength Welds

3.5.1.1 Main seam welds in shells and heads shall comply with the following requirements:

- a. Full weld penetration.
- b. Welds ground smooth and flush (minus zero, plus 1/16 inch reinforcement).
- c. Full radiographic and magnetic powder or liquid penetrant inspection of completed weld plus liquid penetrant inspection of first pass.
- d. Efficiencies of such welds may be taken as 100%, except as may be required by paragraph 3.5.5.

3.5.1.2 Welds at openings and nozzle connections

- a. When all or part of the required compensation is attributable to the nozzle, the nozzle shall be attached by full

penetration welds through either the vessel or the nozzle thickness, as shown in Figure 3.5-1; attention being invited to the limitations in location and angularity of the welds with respect to the nozzle wall contained thereon. When no portion of the nozzle or welds is required for opening compensation, the nozzle need not be attached by full penetration welds, except as limited by subparagraph (b) following.

- b. It is the intent of the foregoing that full penetration welds be used wherever possible, for the purpose of achieving continuity of metal and facilitating the required radiographic examination. Non-full-penetration welds are intended to be allowed only when special problems of design or fabrication make their use advisable. Designs of non-full-penetration welds must be approved by the Naval Reactors Branch.
- c. All welds shall be ground smooth.
- d. All welds which are subject to stress caused by pressure shall be radiographically inspected to specified standards, except where this is impractical for non-full-penetration welds, as permitted in subparagraph (b) above. Wherever possible, all weld joints shall be designed to permit radiographic examination to specified standards. When the weld joints cannot be designed to permit radiography to specified standards, radiographic examination shall nevertheless be made in accordance with best obtainable practice. In addition, when radiography to specified standards is not feasible or when any radiography is impractical, welds shall be examined by other non-destructive methods, such as liquid penetrant, magnetic particle, or ultrasonic, to prove their soundness. None of this requirement applies to seal welds.

3.5.2 Welded Structural Attachment. All structural welds between two non-pressure retaining members, or between a non-pressure retaining member and a pressure retaining member, shall be full penetration welds. Consideration shall be given to non-destructive inspection of such welds where the service may warrant.

3.5.3 Seal Welds. Seal welds are those provided for a closure function only. Strength must be provided by other means, such as by a threaded joint. Seals without flexible diaphragms may be of the configuration shown in Fig. 3.5-2. It shall be noted that such a seal weld must develop sufficient forces and moments to constrain the relative motions between the two components to which the weld is attached, the effects of which shall be evaluated on a fatigue basis. Where use of a flexible sealing diaphragm is necessary, full penetration welds through the thickness of the diaphragm shall be used. Grinding of seal welds is not required; if performed, care must be taken to avoid reduction in thickness below minimum required values. Completed seal welds shall be examined by liquid penetrant inspection.

FIG 3.5-1 ACCEPTABLE NOZZLE WELD LOCATIONS

3.5-2(a)

3.5-2(b)

FIG. 3.5-2. ACCEPTABLE SEAL WELDS FOR SEALS WITHOUT FLEXIBLE DIAPHRAGMS

3.5.4 Bimetallic Welds. Welds between materials of differing mean coefficients of thermal expansion (where one is less than 85% of the other) shall be removed a distance $2.5\sqrt{rt}$ from severe changes in section or environmental condition wherever practicable. (In this relationship, "r" is the mean radius of the cylinder and "t" is its thickness). An example of such environmental condition is the thermal condition which occurs at the end of a cooling coil. In addition to the foregoing, for primary coolant piping, such welds must be placed beyond the transition section of a reinforced nozzle. (The termination of such a transition is not considered to be a "severe change in section", as specified above).

3.5.5 Weld Quality Factor for Fatigue. Since the weld metal may not have as high a fatigue strength as the base metal, it is necessary to account for this weakening effect when present. This is accomplished by the introduction of a weld quality factor, Q, in the table of material properties, Table 4-2. When considering the stress condition in a weld, the appropriate allowable alternating stress S_a is multiplied by the quality factor Q in order to obtain the allowable alternating stress intensity for the weld metal and heat affected zone. The heat affected zone shall be considered to extend one material thickness or one inch, whichever is smaller, from the edge of the weld groove.

3.6 Cladding. For purposes of structural design, austenitic or other dissimilar metal (such as Inconel) cladding on ferritic base metal shall be considered as follows:

- a. No structural strength shall be attributed to the cladding, subject to the contents of paragraph 5.3. The inside diameter of the vessel may, however, be taken as the diameter at the inner surface of the cladding when computing the required thickness.
- b. Stresses and deformations due solely to differences in thermal coefficients of expansion between the cladding and the base metal may be neglected if the cladding thickness is less than 10% of the base metal thickness.
- c. In considering the thermal conditions which will occur in structural members, the presence of the cladding will not be considered unless such a consideration would lead to higher temperatures or higher thermal stresses in the base metal.

4. MATERIALS DATA

Materials designations contained herein are referred to only as a means of material identification. The applicable materials specifications will be listed in each equipment purchase specification. The designations herein cover all forms of the materials (plate, sheet, forgings, weldments, etc.) as long as the composition, mechanical, and physical properties are consistent with the cited specifications.

4.1 Ferritic Materials. The following ferritic materials are applicable to such components as the reactor pressure vessel, the support skirt, and the primary shield tank. These materials may or may not be clad with an austenitic

stainless steel, such as AISI 304, 304 L, or 347, or other dissimilar metal, such as Inconel. For consideration of the cladding, see paragraph 3.6 above.

- a. ASTM A-212, Grade B - carbon silicon steel
- b. ASTM A-204, Grade C - carbon molybdenum steel
- c. ASTM A-304, Grade B - manganese molybdenum steel
- d. MIL-S-16113, Grade HT - high tensile structural steel
- e. SAE 4340 alloy steel
- f. AISI Type 410 stainless steel

Pertinent materials data for these steels are included in Table 4-1 and in Figures 4.1-1 to 4.1-6, inclusive.

4.2 Austenitic Materials. The following austenitic stainless steels may be required for certain components in contact with the primary coolant. These materials may be used as cladding for base metals. For consideration of the cladding in stress calculations, see paragraph 3.6 above.

- a. AISI Type 304 stainless steel
- b. AISI Type 304 L stainless steel
- c. AISI Type 347/348 stainless steel

Pertinent materials data for these steels are included in Table 4-1 and in Figures 4.2-1 and 4.2-2.

4.3 Other Materials. The following other materials may be required for certain components in contact with primary coolant. These materials may be used as cladding for base metals.

- a. Inconel
- b. Inconel X

Pertinent materials data for these materials are included in Table 4-1 and in Figures 4.3-1 and 4.3-2.

5. STRESS LIMITS

Unless otherwise stated, the maximum shear stress theory will be used as a basis for design, but for convenience, formulae and allowable stresses will be expressed in terms of an "equivalent intensity of combined stress". (see paragraph 3.1.7). Unless specifically stated otherwise, all references herein to the strength of a given material (yield, ultimate, fatigue, etc.) refer to minimum strength at design metal temperature based upon minimum specification values. The maximum allowable stress intensities for the various conditions specified below are contained in Table 5-1 for materials listed in paragraph 4 above. The general basis for arriving at these allowable stress figures is outlined in Appendix B as a matter of record and for guidance in arriving at recommended allowable stress figures for any new materials which may be proposed (see paragraph 3.4 above). The following list of symbols will be used to designate various strengths or stress limits herein (some of which are defined more fully in subsequent paragraphs):

TABLE 4-1

	<u>Poisson's Ratio,</u>	<u>Thermal Diffusivity, Sq. Inches/Sec.</u>
Ferritic Steel Except AISI 410	0.30	0.018*
AISI 410	0.30	0.011*
Austenitic Steels	0.30	0.006*
Inconels	0.29	0.006*

* Based on Conductivity and Specific Heat at about 200°F.

TABLE 4-2

WELD QUALITY FACTORS FOR FATIGUE

(See Paragraph 3.5.5)

<u>Material</u>	Room Temperature	
	Minimum Tensile Strength In Kips	<u>Q</u>
A 302B	80	1.0
A 204C	75	1.0
A 212B	70	1.0
MIL-S-16113 HT	70	1.0
SAE 4340	100	1.0
SAE 4340	150	0.85
AISI 410	100	1.0
AISI 410	160	0.85
AISI 304	75	1.0
AISI 304L	70	1.0
AISI 347/348	75	1.0
Inconel	80	1.0
Inconel X	160	0.85

FIG. 4J-1. MATERIALS DATA FOR A-302 GRADE B STEEL

FIG 4/2 MATERIALS DATA FOR A-204, GRADE C STEEL

FIG 4.1-3 MATERIALS DATA FOR A-212 GRADE B STEEL

FIG 41-4. MATERIALS DATA FOR MIL-5-1613, GRADE HT STEEL

FIG 4.1-5. MATERIALS DATA FOR SAF 4340 STEEL

FIG. 43-2 MATERIALS DATA FOR INCONEL X

TABLE 5-1 - MAXIMUM ALLOWABLE STRESS INTENSITIES (1000 psi)

Material	Metal Temperature °F	S _m	S _p	S _y	S _u and S _s	S _b	S _{ae}	S _{aI}	S _{aII}	S _{aIII}
A-302-B	70	27	45	50	80	50	24	24	45	66
	200	↓	44	49	↓	49	↓	↓		
	400		43	47		48				
	600		40	45		48				
	700	↓	38	43	↓	48	↓	↓		
A-204-C	70	25	39	43	75	46	23	23		
	200	25	38	42	↓	↓	↓	↓		
	400	25	36	40						
	600	23	33	37						
	700	21	31	34	↓	↓	↓	↓		
A-212-B	70	23	34	38	70	42	21	21		
	200	22	32	36	70	↓	↓	↓		
	400	20	29	32	70					
	600	18	25	28	70	↓	↓	↓		
	700	16	23	25	66					
MIL-S-16113 HT	70	23	38	42	70	42	21	21		
	200	23	36	39	70	↓	↓	↓		
	400	22	33	35	70					
	600	18	26	29	70	↓	↓	↓		
	700	16	23	26	66					
SAE 4340 (100,000 psi tensile strength)	70	33	60	75	100	75	24	24		
	200	33	60	72	100	72	↓	↓		
	400	32	59	68	99	68				
	600	31	56	64	94	64	↓	↓		
	700	30	53	61	90	61	↓	↓	↓	↓

TABLE 5-1 - MAXIMUM ALLOWABLE STRESS INTENSITIES (1000 psi) (Continued)

Material	Metal Temperature °F	S _m	S _p	S _y	S _u and S _s	S _b	S _{ae}	S _{aI}	S _{aII}	S _{aIII}
SAE 4340 (heat-treated to provide 150,000 psi tensile strength)	70	50	90	130	150	130	34	34	55	75
	200	50	89	125	149	125	↓	↓	↓	↓
	400	49	88	118	147	118	↓	↓	↓	↓
	600	47	84	110	140	110	↓	↓	↓	↓
	700	44	80	105	134	105	↓	↓	↓	↓
AISI 410 (heat-treated to provide 100,000 psi tensile strength)	70	33	60	80	100	80	24	24	25	66
	200	31	56	79	94	79	↓	↓	↓	↓
	400	30	53	75	89	75	↓	↓	↓	↓
	600	28	50	69	84	69	↓	↓	↓	↓
	700	26	47	65	79	65	↓	↓	↓	↓
AISI 410 (heat-treated to provide 160,000 psi tensile strength)	70	53	96	140	160	140	34	34	55	75
	200	50	90	130	150	130	↓	↓	↓	↓
	400	50	90	123	150	123	↓	↓	↓	↓
	600	50	90	120	150	120	↓	↓	↓	↓
	700	50	90	119	150	119	↓	↓	↓	↓
AISI 304	70	19	27	30	75	46	23	23	45	66
	200	16	23	26	67	42	21	21	↓	↓
	400	13	19	21	62	38	19	19	↓	↓
	600	11	16	18	60	36	18	18	↓	↓
	700	10	15	16	59	34	17	17	↓	↓
AISI 304-L	70	16	22	25	70	44	22	22	↓	↓
	200	13	19	21	63	38	19	19	↓	↓
	400	11	15	17	59	34	17	17	↓	↓
	600	9	13	14	57	32	16	16	↓	↓
	700	8	12	13	56	30	15	15	↓	↓

TABLE 5-1 - MAXIMUM ALLOWABLE STRESS INTENSITIES (1000 psi) (Continued)

Material	Metal Temperature °F	S _m	S _p	S _y	S _u and S _s	S _b	S _{ae}	S _{aI}	S _{aII}	S _{aIII}
AISI 347/348	70	19	27	30	75	46	23	23	45	66
	200	17	24	27	68	42	21	21	↓	↓
	400	15	22	25	62	38	19	19	↓	↓
	600	14	21	23	60	36	18	18	↓	↓
	700	14	20	22	59	34	17	17	↓	↓
INCONEL	70	19	27	30	80	48	24	24	↓	↓
	200	18	25	28	78	48	↓	↓	↓	↓
	400	17	24	27	77	46	↓	↓	↓	↓
	600	17	24	27	77	46	↓	↓	↓	↓
	700	16	23	26	77	46	↓	↓	↓	↓
INCONEL X (heat-treated to provide 160,000 psi tensile strength)	70	53	81	90	160	90	34	34	55	75
	200	53	80	89	159	89	↓	↓	↓	↓
	400	52	79	88	157	88	↓	↓	↓	↓
	600	52	77	86	154	86	↓	↓	↓	↓
	700	51	76	85	152	85	↓	↓	↓	↓

S_y	= minimum yield strength of material
S_b	= stress intensity at limit of elastic behavior (see B.2.5)
S_u	= minimum ultimate tensile strength of material.
S_e	= endurance limit (or endurance strength at 10^7 cycles) of the material.
S_f	= fatigue strength of material at some finite number of cycles.
S_m	= allowable membrane stress intensity. (see Table 5-1)
S_p	= allowable primary-plus-secondary stress intensity. (see Table 5-1)
S_a	= allowable amplitude of alternating stress intensity.
$S_{aI}, S_{aII}, S_{aIII}$	= allowable S_a value for Classes I, II, III, respectively. (see 3.3.4) (see Table 5-1)
S_{ae}	= allowable S_a value for vibratory conditions. (see Table 5-1)
S_s	= allowable mean stress intensity for fatigue evaluation. (see Table 5-1)
S	= stress intensity (see 3.1.7).
S_r	= calculated range of alternating stress intensity.
S_{alt}	= calculated amplitude of alternating stress intensity = $0.5 S_r$.
S'_{mean}	= basic value of the calculated mean stress intensity.
S_{mean}	= adjusted value of the mean stress intensity. (see 5.2.4)

5.1 Steady-State Conditions.

5.1.1 Membrane Stresses (internal design pressure). The stress intensity based upon the average value of the Primary Stress (see 3.1.5) across the thickness of a section, neglecting structural discontinuities and stress concentrations, due to design values of internal pressure, mechanical forces, or their combinations, shall not exceed the allowable values of membrane stress designated by the symbol " S_m ", in Table 5-1.

5.1.2 Membrane Stresses (external design pressure). The wall thickness of members subjected to external pressure shall be determined in accordance with Appendix V of Section VIII, ASME Boiler and Pressure Vessel Code, 1956 edition, except that the factor B, used therein, shall be multiplied by the quantity 1.33.

5.1.3 Primary-plus-Secondary Stresses. The primary-plus-secondary stress intensity (see 3.1.5 and 3.1.6) at any location due to internal or external design pressure, mechanical forces, pipe reactions, or their combinations, including the effect of any structural discontinuities (but not stress concentrations), shall not exceed the allowable values designated by the symbol " S_p ", in Table 5-1.

5.1.4 Thermal Stresses. No stress limitations need be considered with regard to steady-state thermal stresses but distortion limits (paragraph 1.5) apply. All thermal stresses shall be considered as transient conditions and treated in accordance with paragraph 5.2.

5.2 Combinations of Steady-State and Transient Conditions. The steady-state and transient conditions (see paragraph 3.3.3) shall be combined in accordance with the modification of the Goodman diagram illustrated in Fig. 5.2-1 (hereinafter referred to as the "fatigue diagram"). Operating cycles which must be investigated are as defined in paragraph 3.1.4.

The procedure for the analysis of combined steady-state and transient conditions is as follows:

5.2.1 Principal Stresses. Consider the value of each of the three principal stresses at the point in question versus time for a complete cycle. The effects of both structural discontinuities and stress concentrations are included. Call these stresses σ_1 , σ_2 , and σ_3 for later identification.

5.2.2 Stress Differences. Consider the stress differences defined by

$$S_{12} = \sigma_1 - \sigma_2; \quad S_{23} = \sigma_2 - \sigma_3; \quad S_{31} = \sigma_3 - \sigma_1$$

versus time for the complete cycle. Use the symbol S_{ij} to represent any one of the three stress differences, where i and j take on the values of 1, 2, 3 as required. At any given time, the absolute value of the largest of these stress differences is the stress intensity.

5.2.3 Alternating Stress Differences. For each S_{ij} , locate the extremes of the range through which it fluctuates and find the absolute magnitude of this range. Call this magnitude S_{rij} and let $S_{alt\ ij} = 1/2 S_{rij}$. (Note that consistent values of i and j must be used throughout any one equation). (For example, if S_{12} fluctuates from +20,000 psi to -30,000 psi, $S_{r12} = 50,000$ psi and $S_{alt\ 12} = 25,000$ psi). At any given time, the absolute value of the largest of these quantities is the alternating stress intensity.

5.2.4 Mean Stress Differences. For each S_{ij} , locate the mean value of the range through which it fluctuates and call its absolute value $S'_{mean\ ij}$. (For the example given above $S'_{mean\ 12} = 5000$ psi). This is the basic value of the mean stress difference and the actual value for use in the fatigue diagram depends on whether or not S_p has been exceeded during the cycle.

- (1) For any S_{ij} for which $S_{alt\ ij} + S'_{mean\ ij} \leq S_p$ the mean stress difference is equal to its basic value, or

$$S_{mean\ ij} = S'_{mean\ ij}$$

FIG. 5.2-1 FATIGUE DIAGRAM FOR COMBINED
STEADY STATE AND TRANSIENT CONDITIONS

- (2) For any S_{ij} for which $S_{alt\ ij} + S'_{mean\ ij} > S_b$,
and $S_{alt\ ij} < S_b$, then

$$S_{mean\ ij} = S_b - S_{alt\ ij}$$

- (3) For any S_{ij} for which $S_{alt\ ij} \geq S_b$, then

$$S_{mean\ ij} = 0.$$

Attention is called to the discussion of the mean stress quantity in paragraph A-2. If the mean stress is obviously compressive, the value zero may be used in the fatigue diagram without requesting a waiver.

5.2.5 Use of the Fatigue Diagram. The fatigue diagram is a plot which uses the absolute value of the alternating stress difference as the ordinate, and the absolute value of the mean stress difference as the abscissa. (see Fig. 5.2-1(b)). The safe region is that which lies below a line whose intercepts are S_a on the ordinate, and S_g on the abscissa. (The method of determining these limiting points is described in Appendix B.2). It is necessary that each of the three points

$$(S_{mean\ 12}, S_{alt\ 12}); (S_{mean\ 23}, S_{alt\ 23}); (S_{mean\ 31}, S_{alt\ 31})$$

be within the safe region for the design to be adequate or the operating condition acceptable.

Further explanation of this procedure and an illustrative example are given in Appendix A.

5.2.6 The above rules lead to the general conclusion that when the applicable allowable value of alternating stress intensity, S_a , is equal to, or greater than, S_b , the alternating stress intensity determines whether or not the stress condition is safe. For this case, if the alternating stress intensity, S_{alt} , is equal to, or less than S_a , the stress condition is safe. (see paragraph A-1).

5.2.7 For design, the allowable values of the alternating stress intensity for the three loading classes, designated by the symbol, S_a , are given in Table 5-1. The development of these allowable stress values is given in Appendix B.

5.2.8 As an alternate method to paragraph 5.2.7 for establishing plant operating limitations, the limitations on alternating stress intensity may be established and the effects of cumulative fatigue damage evaluated in accordance with a modification of Miner's hypothesis. The following procedure shall be followed in making such evaluation:

- (1) For every operating condition, make a conservative but realistic estimate of the number of times that each operation will be performed during the life of the vessel. For conditions "1", "2", "3", etc., call these values n_1 , n_2 , n_3 , -- etc., respectively.

- (2) Plot the corresponding three mean and alternating stress differences for each operating condition on the fatigue diagram. Draw straight lines through these points (from S_g) to their intercepts on the ordinate (alternating stress) axis and considering only the largest value of the intercept for each operating condition, call these intercepts S_{a1} , S_{a2} , S_{a3} , etc., as shown in Figure 5.2-2a.
- (3) Next, for each of these intercepts on the vertical axis, and using the applicable fatigue strength curve for design purposes (see Figs. 5.2-3 and 5.2-5), determine the number of repetitions which would be required to produce an unsafe condition if this operating cycle were the only one involved, as illustrated in Fig. 5.2-2b. Call these values N_1 , N_2 , N_3 , etc.
- (4) In accordance with Miner's hypothesis, the cumulative usage factor (U) is then $U = U_1 + U_2 + U_3 + \dots + U_n$, or

$$U = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} + \dots + \frac{n_n}{N_n}$$
- (5) The cumulative usage factor so determined shall not exceed 0.80.

5.3 Bearing Stresses

5.3.1 The average bearing stress for resistance to crushing under the maximum attainable load shall be limited to 1.5 times the values of " S_y " listed in Table 5-1, except as qualified by the following paragraphs. No fatigue evaluation is required.

5.3.2 In the case of clad surfaces, the stress limits of paragraph 5.3.1 shall apply both to the base metal and to the cladding. In calculating the bearing stress in the base metal, the assumed breadth of the loaded area shall be increased beyond that used for the cladding by twice the cladding thickness. Full consideration shall be given to the possibility of excessive distortion of the clad metal, such as could occur in narrow bands without lateral support.

5.3.3 In the case of hard-faced surfaces, the average bearing stress shall be limited to the respective values of " S_y " listed in Table 5-1 for both the facing material and the base metal.

5.4 Bolting Stresses

5.4.1 Number of Bolts and Nominal Stress. The number and cross sectional area of bolts required for resistance to internal pressure shall be determined in accordance with the procedures of Section VIII, Appendix II, of the ASME Boiler and Pressure Vessel Code. When gaskets are not used for sealing, the gasket factor, m , and the minimum design seating stress, y , may be taken as zero. The maximum allowable bolt design stresses, as used in the Code formulas, may be based on heat treated properties for operating metal temperatures 100°F or more below the tempering temperature, provided that the stresses do not exceed 1/3 of the yield strength at temperature. It should be noted that the

Code method here prescribed does not in any way consider or limit the stresses which actually exist in the bolts due to preload or service.

5.4.2 Preload Stress. In calculating bolting stresses due to preload, consideration must be given to the method of bolt tightening, since tightening methods which utilize mechanical torque may establish significant residual torsional stresses. The preload shall be sufficient to prevent separation of the seating surfaces under service conditions. Differential thermal expansions and flange distortions must be considered.

5.4.3 Service Stress. The acceptability of stresses calculated for service conditions shall consider the possibilities of fatigue and bolt relaxation.

5.5 Interpretations

5.5.1 For various illustrations of the application of stress limits S_m , S_p , and the fatigue method, see paragraphs A.8 and A.9.

5.5.2 Perforated Plates or Shells

5.5.2.1 Typical ligament in a uniform pattern:

- a. The stress intensity produced by pressure or other mechanical load averaged across the width of the ligament and through the thickness of the plate is limited to S_m .
- b. The stress intensity produced by pressure or other mechanical load averaged across the width of the ligament but not through the thickness of the plate is limited to S_p .
- c. The peak stress intensity at any point produced by all effects is evaluated by means of the fatigue diagram.

5.5.2.2 Isolated ligament or thin ligament in a nominally uniform pattern:

- a. The stress intensity produced by pressure or other mechanical load averaged across the width of the ligament and through the thickness of the plate is limited to S_p .
- b. The peak stress intensity at any point produced by all effects is evaluated by means of the fatigue diagram.

6. DETAILED STRUCTURAL REQUIREMENTS

The following requirements cover the design of pressure vessel details, which, in certain cases, are not susceptible to adequate design on an analytical basis. Ordinarily experimental evidence (either photoelastic tests or strain gage measurements on metal models, or both) will be required to establish

the adequacy of design of such parts. The design requirements specified herein do not preclude the proposal and use of alternate design procedures but the burden of proof of the adequacy of such alternate design rests with the manufacturer. It should be anticipated that experimental evidence will be required for any case involving a departure from the requirements contained herein, and specific approval of the Naval Reactors Branch (via applicable channels) must be obtained for any such departure. The requirements for analytical or experimental analysis contained herein do not preclude the possibility that the fabricator may already possess adequate evidence based upon previous experience with similar configurations which makes repeated investigation unnecessary. In certain cases, specific requirements for such tests will be spelled out in the Equipment Specification.

6.1 Basic Pressure Vessel Shell. Formulas for calculation of pressure and thermal stresses in the basic pressure vessel shell are listed in Appendix A.

6.2 Fixed Heads

6.2.1 Fixed heads shall be hemispherical or ellipsoidal in form, except that approximate ellipsoidal or torispherical heads will be considered when it can be definitely shown that the primary-plus-secondary stress intensity for such head is within the prescribed limits.

6.2.2 The minimum required thickness of an unperforated 2:1 ellipsoidal head is the same as the minimum required thickness of an attached unperforated cylindrical shell.

6.2.3 Computer codes are in process of development for the calculation of stresses in ellipsoidal heads due to internal pressure, and information will soon be available on the stress indices which should be used. These indices are required for (1) the calculation of primary-plus-secondary stress for comparison with S_p and (2) the calculation of the combined effect of pressure, mechanical, and thermal loads for comparison with fatigue curves. The data now available show that in a 2:1 ellipsoidal head with a thickness equal to that of the cylindrical shell, the maximum primary-plus-secondary stress intensity is less than 1.5 times the membrane stress in the attached cylinder, provided the diameter-to-thickness ratio is not greater than 50. Since $S_p/S_m \approx 1.5$ for most materials, and since diameter/thickness is less than 50 in most pressurized-water reactors, the rule of paragraph 6.2.2 above is conservative.

6.2.4 Any pipe connection, including local reinforcement, or other attachment on ellipsoidal or torispherical heads shall be located completely within a circle, concentric with the axis of the vessel, having a diameter equal to 40% of the inside diameter of the attached cylinder.

6.2.5 The centerline of all primary coolant pipes shall be normal to the head at the point of entry.

6.3 Upper (Removable) Heads. The design of the upper heads shall conform to the requirements of the specific installation, as outlined in the Equipment Specification. When these requirements call for a head differing

appreciably from past practice, adequacy of the design must be verified experimentally until better methods become available for analytical determination of stresses. This may be accomplished by direct test or by detailed comparison with other head configurations previously tested, data for which will be furnished to the manufacturer.

6.4 Flanges, Reducers, Transitions, etc.

6.4.1 Stresses in flanges and reducers shall be determined by detailed stress analysis, comparison with similar existing designs, or by experimental techniques. ASME Boiler and Pressure Vessel Code procedures are not necessarily adequate.

6.4.2 Abrupt changes in contour due to forming, fit-up, etc., shall be avoided. Where unequal thicknesses are involved in a vessel (head-to-shell, shell-to-flange, etc.), the transition between the two sections should conform to the following rules:

- (1) Where possible, the centerlines of plates of differing thickness should coincide; otherwise, bending stresses will be introduced, because of eccentric loadings.
- (2) The slope of the transition section shall not exceed 1:3 in the longitudinal direction without experimental verification of stresses.
- (3) A smooth fillet must be provided between the transition section and the thinner plate, with the fillet having a radius of at least 50% of the thickness of the thinner plate.
- (4) When the centerlines do coincide and conditions (2) and (3) are met, analysis for discontinuity pressure stresses is not required.
- (5) Any weld located in or adjacent to such transition must be so located and of such configuration that prescribed non-destructive examination of welds will not be adversely affected.

6.4.3 In the design of bearing surfaces, such as seating and support ledges, particular care shall be taken to fillet or undercut the corners of such ledges to minimize stress concentrations.

6.5 Closures (Bolting, Seals, etc.) The design of closures varies widely between different vessel designs and between manufacturers, making it impracticable to outline specific requirements. Therefore recommendations concerning proposed stresses and pre-load practices on bolted joints, shear blocks, sealing members (such as Belleville springs) shall be submitted to the Naval Reactors Branch for approval as a part of the design report for the vessel. These recommendations shall include a detailed justification of the methods proposed. Written approval of the design and procedures must be obtained before use.

6.6 Nozzle Connections and Openings. Piping connections and openings in the vessel proper (other than the upper head) shall comply with the following requirements.

6.6.1 Compensation for Openings. All openings in the pressure vessel will require compensation in accordance with Appendix C, regardless of size. Acceptable types of compensation, for which experimental verification of stresses will not be required, are furnished for guidance in that Appendix. For any case where the nozzle or opening is located closely adjacent to a transition or other discontinuity which might materially affect the stress pattern, or where the entry of major openings or piping connections is non-radial, the adequacy of the design may require experimental verification.

6.6.2 External Loadings on Nozzles. The external loadings for nozzle connections will be specified in the Equipment Specifications. Unless otherwise approved, calculation of shell stresses due to external loadings shall be based upon Bijlaard's theory, as summarized in paragraph A-5 of the Appendix, together with any subsequent extensions or modifications of the references cited therein.

6.6.3 Threaded Connections. The restrictions of ASME Boiler and Pressure Vessel Code, paragraph UG-43 (F) do not apply to threaded connections used for the attachment of control drive mechanism housings, fuel ports, or instrumentation ports.

7. INSPECTIONS AND TESTS

7.1.1 Requirements for inspection of welding are covered by paragraph 3.5 above.

7.1.2 Hydrostatic Tests. Subject to the limitations given below, hydrostatic tests shall conform to the requirements of Section VIII, ASME Boiler and Pressure Vessel Code, paragraph UG-99.

- (1) The hydrostatic pressure test of each pressure containing structure shall be conducted at a pressure equal to 1.5 times the design pressure multiplied by the lowest ratio (for any of the materials involved in pressure containment) of the S_m value for the test temperature to the S_m value for the design temperature, where the S_m values are taken from Table 5-1.
- (2) The foregoing requirements apply to members which are primary pressure retaining members, but not to members which fulfill a sealing rather than a strength function. Sealing members need only be tested to a pressure equal to 110% of the system design pressure.

- (3) In order to decrease the probability of brittle fracture of ferritic steels, the following load restrictions shall be used in relation to the Nil-Ductility Transition (NDT) temperature (as defined below). When the metal temperature is less than 60°F above the NDT temperature, the maximum applied load, including internal pressure, shall be restricted to 20% of the design value. When the metal temperature is more than 60°F above the NDT temperature there is no restriction upon the load from the aspect of brittle fracture prevention. However, under no circumstance shall a hydrostatic test be conducted at a temperature exceeding 200°F. There is no restriction on low temperature loading of materials other than ferritic steels.
- (4) The Nil-Ductility Transition (NDT) temperature is that temperature below which the metal fails in a brittle manner in a drop weight test.* Correlations with Charpy V-Notch test are a satisfactory means of determining the NDT temperature. The following correlations may be used:

<u>Material</u>	<u>Charpy V-Notch Energy Absorption (ft-lb) Correspondence to NDT Temperature</u>
ASTM A-212	15
AISI Type 410	20
ASTM A-302	30
SAE 4340	35

Correlation data for other ferritic steels listed in paragraph 4.1 are not available and should be obtained if they are to be loaded at temperatures which may fall below NDT + 60°F. The following correlation between yield strength and Charpy V-Notch energy absorption is believed to be conservative and may be used as a guide in the absence of better data:

<u>Material Class By Yield Strength</u>	<u>Charpy V-Notch Energy Absorption (ft-lb)</u>
Less than 45,000 psi	20
45,000 to 75,000 psi	30
greater than 75,000 psi	40

It may be found in some cases, especially for high strength steels, that the Charpy V-Notch Energy Absorption is at all times below the tabulated value. In such cases, the Charpy V-Notch correlation shall be determined experimentally.

* "The Evaluation of the Significance of Charpy Tests for Quenched and Tempered Steels", by P. P. Puzak and W. S. Pellini, Welding Journal, Research Supplement, June 1956.

8. INSTALLATION AND OPERATING PRECAUTIONS

8.1 Brittle Fracture Prevention. In order to decrease the probability of brittle fracture, the following load restriction shall be used in relation to the nil-ductility transition (NDT) temperature. When the metal is less than 60°F above the NDT temperature, the primary-plus-secondary tensile stress due to pressure, mechanical forces, pipe reactions, preload effects, or their combination, shall not exceed 20% of the allowable values designated by S_p , (Table 5-1). When the metal temperature is more than 60°F above the NDT temperature, there is no restriction upon the load from the aspect of brittle fracture prevention. These restrictions shall apply to hydrostatic testing, assembly and operational conditions.

APPENDIX A

STRESS CRITERIA AND CALCULATIONS

A.1* SURVEY OF DESIGN PROCEDURE

It has been deemed advisable to include a narrative description of this design philosophy in order that the basic method may be described without the encumbrance of technical details. Therefore, this appendix presents such a description for a representative component, the reactor pressure vessel.

A.1.1. The functional requirements for the vessel are assumed to be known. These include such items as the height, required internal diameter, desired location of inlet and outlet pipes, preferred method of support, removable head configuration and penetration pattern, preferred type of closure design, and the type and location of internal or external attachments.

A.1.2 The operational requirements and objectives are also assumed to be known. It is essential that a definite separation of requirements and objectives be recognized. Requirements are items which are necessitated by the basic characteristics of the plant. Objectives are generally desired operational characteristics.

A.1.3 It is also assumed that the structural material, its properties, and allowable stress values are known in the temperature range of interest.

A.1.4 The first step in the design procedure is that of establishing the basic configuration. This step requires consideration of the functional requirements, the structural material, and the operational requirements with regard to mechanical (non-thermal) loadings and metal temperatures insofar as they affect allowable stress values. The basic metal thicknesses are determined by consideration of the allowable stress quantities designated by the symbols S_m , S_p , and S_y and by consideration of the detailed structural requirements. The influence of these considerations is as follows:

A.1.4.1 S_m (see paragraph 5.1.1) limits the membrane (average) stress through the thickness such that gross deformation will not occur during hydrostatic pressure testing. The effects of structural discontinuities (paragraph 5.1.2) and "stress concentrations" are not considered. This determines basic wall thickness.

A.1.4.2 S_p (see paragraph 5.1.3) limits the stress at any location through the thickness such that gross yielding does not occur as a result of subsequent pressure excursions which are limited to 110% of design pressure. The effects of "stress concentrations" are not considered. This determines basic geometry at flanges, changes in thickness, etc.

* Appendix A.1 is for information only and does not contain requirements with respect to structural limitations or design procedures. Section A.1 is not intended to either supersede or modify design requirements or objectives which form a part of equipment specifications.

A.1.4.3 S_y (see paragraph 5.3) limits bearing stresses in order to prevent significant deformations.

A.1.4.4 Detailed structural requirements. Paragraph 6 contains details for configurations which are recurrent and which are acceptable under the conditions stated therein. For example:

- a. Fixed heads (paragraph 6.2)
- b. Removable heads (paragraph 6.3)
- c. Flanges, reducers, transitions, etc. (paragraph 6.4)
- d. Closures (paragraph 6.5)
- e. Nozzle connections and openings

The foregoing requirements supplement the results of calculations considering S_p where analysis is difficult or unnecessary.

A.1.4.5 If bolts are to be used, paragraphs 5.4.1 and 5.4.2 will be of interest in determination of the basic configuration.

A.1.5 The design procedure steps which follow all require the evaluation of various operating conditions on a fatigue basis. Since the fatigue analysis method is the same for each of the following steps of the design procedure, a more detailed discussion of the method will follow the discussion which outlines these steps and their significance. In order to facilitate the design procedure, each transient operation is placed into one of these operating classes, depending upon the severity of the transient and the number of anticipated occurrences. These classes are defined in paragraph 3.3.4.

A.1.6 The second step in the design procedure is that of ensuring satisfaction of operational requirements. In addition to the considerations involved in determining the basic configuration, it is now necessary to include operational requirements with regard to thermal loadings and other repeated loads. The design is considered to satisfy the operational requirements if the desired number of applications of loading may be safely applied for all conditions.

A.1.7 The third step in the design procedure is that of evaluating combinations of steady-state and transient conditions which result from operational objectives and the establishment of plant operating procedures. In addition to the considerations involved in determining the basic configurations, it is now necessary to consider the operational objectives. For example, one objective might be that of attaining a certain start-up and shutdown rate from cold conditions. By definition, (see paragraph 3.3.4) this is a Class III operation; that is, it is assumed to occur 500 times in the plant life. The stress conditions resulting from start-up and shutdown, using the desired rates, are then evaluated and the fatigue analysis method applied in order to determine whether or not the operation is safe. If not, slower rates are analyzed until those that give the limiting stress are found. This rate is then used as the plant operating procedure. This step is repeated until all conditions of operation have been considered and all plant operating procedures determined.

A.1.8 The fatigue analysis method of the design procedure may be described as follows:

A.1.8.1 It is assumed that the stress condition is known versus time for a complete cycle of operation. (see paragraph 3.1.4). For simplicity in this presentation it will be assumed that a simple uniaxial stress exists in the component of interest. The method for handling complex stress conditions is given in paragraph 5.2. The stress variation with time considering structural discontinuities and stress concentrations may be as sketched. (Fig. A.1-1). The quantities indicated on the sketch are the applied stress conditions required for the analysis of the fatigue condition.

A.1.8.2 It is also necessary to determine if the maximum stress during a cycle exceeds the yield strength of the material for fatigue loading. It must be recognized that these materials which have an endurance strength above their static yield strength strain harden during cycling so that the yield strength for fatigue loading approaches the endurance limit. The symbol S_p is used in the material property tables to designate the yield strength for fatigue loading. This value is the larger of the static yield strength or the endurance limit without safety factor.

A.1.8.3 Although this design basis follows a procedure whereby all stresses are calculated on an elastic basis, a method is used to account for the shift in mean stress when the maximum stress of the cycle exceeds S_p . In order to describe the significance of this procedure, consider a simple uniaxial stress application with the stress varying from zero to a maximum, designated by S_{max} . Then

$$S_{alt} = \text{alternating stress} = S_{max}/2$$

$$S'_{mean} = \text{basic mean stress} = S_{max}/2$$

The stress condition may be represented graphically on an alternating stress versus basic mean stress plot as in the following sketch. (Fig. A.1-2). It is obviously a straight line with a slope of plus 1, as shown. The dotted line joining the points S_p on the ordinate and abscissa represents the condition that the material is at its yield value. If $S_{max} < S_p$, the points for the case considered lie along the line between Points O and A. If $S_{max} > S_p$ and the material is subjected to the calculated stress-time condition shown in dotted lines on the following sketch (Fig. A.1-3), the actual behavior will be (approximately) that shown by solid lines. Note that although the alternating stress has not changed, the actual mean stress (S_{mean}) is the basic mean stress, (S'_{mean}) reduced by the amount by which S_{max} exceeds S_p . If the values of S_{alt} and S_{mean} [$= S'_{mean} - (S_{max} - S_p) = S_{max}/2 - S_{max} + S_p = S_p - S_{max}/2$] are plotted on the alternating stress versus mean stress diagram, it is found that the points will lie along the line between Points A and B, (Fig. A.1-2). This procedure is realistic as long as $S_{max} < 2S_p$; that is, when yielding occurs only in tension. (Note that the stress point plots at Point B if $S_{max} = 2S_p$). However, if $S_{max} > 2S_p$, yielding will also occur in compression and the actual mean stress becomes zero. That is, the stress points plot along the ordinate axis above Point B.

FIG. A.I-1

FIG. A.I-2

FIG. A.1-3

It is concluded that for the case of a calculated stress varying from zero to S_{max} , the possible stress points lie between Points O and A if $0 < S_{max} < S_b$, between Points A and B if $S_b < S_{max} < 2S_b$, and on the ordinate axis above Point B if $2S_b < S_{max}$. Similar procedures may be used to show that as a consequence of the procedures followed herein all possible stress points on the S_{alt} versus S_{mean} plot lie either within the triangle OCB or along the ordinate axis above Point B.

A.1.8.4 In order to determine if the fatigue condition is safe, an allowable condition line is superimposed upon the alternating stress versus mean stress plot just discussed. The end points of this line are designated by S_a on the ordinate and by S_s on the abscissa where:

S_a = the allowable value of the alternating stress (S_{alt}) for zero mean stress (S_{mean}). Values are given for each of the three classes of loading.

S_s = the ultimate tensile strength, S_u . Stress points which fall below the line $\overline{S_a - S_s}$ are considered to be safe whereas points above this line are considered to be unsafe.

A.1.8.5 Two possible situations occur:

- a. If $S_a \geq S_b$, the effects of mean stress may be neglected and the condition is safe if $S_a \geq S_{alt}$. In this case the limiting line will always lie above the line $S_b - S_b$ below which all stress points must lie for $S_{mean} > 0$.
- b. If $S_a < S_b$, the effects of mean stress must be considered. In this case the limiting line will lie below the line $S_b - S_b$ for some values of S_{mean} .

A.1.8.6 The fatigue analysis method of the design procedure, as just described, is conservative for the following reasons:

- a. It is assumed that the worst condition in each class of loading occurs the full number of cycles assumed in deriving the allowable stress for that class of loading; whereas it actually occurs only a fraction of the assumed number of cycles.
- b. It is assumed that the worst condition for each class of loading occurs at the same point in the structure; whereas it is improbable that this will actually occur.
- c. This conservatism is offset an unknown amount since the existence of additional loading conditions, other than the worst condition, within each class, is neglected.

A.1.9.1 An alternate procedure is also contained herein (paragraph 5.2.8) for the determination of plant operating limitations by consideration of accumulative fatigue damage. This procedure tends to remove some of the conservatism in permissible operating conditions which results from the simple, more approximate design procedure. It is generally not practicable to follow this more complete procedure during the design stages unless there are a limited number of operating conditions of significance to the component considered. This method makes use of the fatigue diagram in order to determine an equivalent alternating stress with zero mean stress for use on the fatigue design curve in place of the actual alternating and mean stresses. The safe number of cycles for this condition is then evaluated from the design fatigue curve. The ratio of the anticipated number of cycles to the safe number of cycles is determined for each operating condition. The component is considered to be safe for all of the imposed operating conditions if the sum of these ratios for all conditions at the point of interest is less than 0.8.

A.1.9.2 The procedure may be illustrated graphically by consideration of the following sketch (Fig. A.1-4). To the left of the ordinate axis is a fatigue design curve; whereas to the right is a fatigue diagram. Note that the allowable stress values S_a which were used for the various classes of loadings, do not appear and are not considered in any way in this procedure. It should also be noted that as a consequence of the reduction of the mean stress due to yielding, all possible points on the fatigue diagram fall on or within the triangle S_p OS_p or along the ordinate axis, above S_p .

A.1.9.3 The procedure for a given operation, say that designated by 2, is as follows:

- a. Plot the point ($S_{\text{mean } 2}$, $S_{\text{alt } 2}$).
- b. Draw a line between this point and the point S_g and determine the intercept, S_{a2} , of this line with the ordinate axis.
- c. Draw a horizontal line to the left of the ordinate axis to intercept the fatigue design curve.
- d. Draw a vertical line through this intercept and determine the number of safe cycles, N_2 , for this operational condition.
- e. Form the ratio of the conservative but realistic number of cycles anticipated for this operational condition to this safe number of cycles. This ratio is the usage factor, U_2 , for operation 2.

A.1.9.4 This procedure is repeated for each operation at the location considered in order to evaluate the cumulative usage factor. The sum of the individual usage factors must be kept below the value 0.8.

A.2 Combinations of steady-state and transient conditions. In the evaluation of combined steady-state and transient conditions, the quantities S_{alt} and S_{mean} are used. The quantity $S_r = 2S_{alt}$ is the full range through which the stress intensity fluctuates during the cycle and S_{mean} is the mid-point between the two extremes. Stresses are calculated on an elastic basis but the resulting values are compared to strain-range test values so the occurrence of yielding during the cycle does not affect the value of S_{alt} which is used. It does, however, affect the value of S_{mean} as discussed in paragraph A.1.

In the specified procedure of paragraph 5.2, only absolute values of S_{mean} and S_{alt} are used. This procedure is based on the assumption that mean compressive stress has the same deleterious effect on fatigue life as mean tensile stress, which is probably not true. If only uniaxial stress were to be involved, it would be safe to assume that mean compressive stress has no deleterious effect at all. The procedure used must, however, be general enough to handle biaxial and triaxial stress states. Cases, such as fluctuating torsion, can arise in which the sign of the stress intensity is ambiguous and it is impossible to decide whether the mean intensity is or is not important. More work is in progress on this subject and it may be possible in the future to devise a less restrictive procedure. For the present, however, the specified procedure is known to be safe. As noted in paragraph 5.2d, when the mean stress is obviously compressive, it should be taken as zero.

A.3 Basic Formulas for Stress Calculation. This section gives a few of the basic formulas for stress calculation which are commonly used in the design of a reactor vessel. It is included merely as a convenience for the designer and is not a check-list of all points which must be covered. The basic formulas for pressure stress are included because the use of the maximum-shear criterion results in expressions different from those found in the ASME Boiler Code. The formulas given here for thermal stress are included because they are not readily available in standard texts. A list of standard texts and published references is given in the Bibliography, Appendix E.

NOMENCLATURE

S	=	equivalent intensity of combined stress, lb/in ²
σ_r	=	radial stress component
σ_h	=	hoop stress component
σ_a	=	axial stress component
P	=	internal pressure, lb/in ²
D_i	=	inside diameter of vessel, inches
R_i	=	inside radius of vessel, inches
t	=	thickness of vessel wall, inches
x	=	distance from inner wall, inches
ν	=	Poisson's ratio (dimensionless)
E	=	Young's modulus, lb/in ²
α	=	linear coefficient of thermal expansion, in/in °F
k	=	thermal conductivity, Btu/hr ft °F
h	=	film coefficient, Btu/ft ² hr °F
β	=	linear absorption coefficient, per inch
q	=	heat generation per unit volume, Btu/hr in ³
ϵ	=	diffusivity, in ² /sec.
T_f	=	temperature of coolant, °F
T_i	=	temperature of inner wall, °F
T_o	=	temperature of outer wall, °F
T_m	=	mean wall temperature, °F
T_x	=	temperature at any location in wall, °F
ΔT	=	change in temperature at any point, °F
θ	=	time, sec.
Y	=	radius ratio = $\frac{R_i + t}{R_i}$

- A.3.1 (1) Required minimum thickness of simple hollow cylinder or sphere under internal pressure, based on membrane stress intensity:

$$\text{Cylinder: } t = \frac{2PR_1}{2S_m - P} \quad \text{----- (1)}$$

$$\text{Sphere: } t = \frac{PR_1}{2S_m - P} \quad \text{----- (2)}$$

- (2) Membrane stress intensity due to internal pressure.
(S must not exceed S_m):

$$\text{Cylinder: } S = \frac{P(2R_1 + t)}{2t} \quad \text{----- (3)}$$

$$\text{Sphere: } S = \frac{P(R_1 + t)}{2t} \quad \text{----- (4)}$$

- A.3.2 Stress intensity at wall surfaces of simple hollow cylinder or sphere due to internal pressure. (S must be less than S_p):

$$\text{Cylinder, inside: } S = \frac{2PY^2}{Y^2 - 1} \quad \text{----- (5)}$$

$$\text{Cylinder, outside: } S = \frac{2P}{Y^2 - 1} \quad \text{----- (6)}$$

$$\text{Sphere, inside: } S = \frac{3P}{2} \left[\frac{Y^3}{Y^3 - 1} \right] \quad \text{----- (7)}$$

$$\text{Sphere, outside: } S = \frac{3P}{2} \left[\frac{1}{Y^3 - 1} \right] \quad \text{----- (8)}$$

- A.3.3 Principal stress components at wall surfaces of simple hollow cylinder or sphere due to internal pressure:

$$\text{Cylinder, inside: } \begin{cases} \sigma_r = -P \\ \sigma_h = P \left[\frac{Y^2 + 1}{Y^2 - 1} \right] \end{cases} \quad \text{----- (9)}$$

$$\quad \quad \quad \begin{cases} \sigma_h = P \left[\frac{Y^2 + 1}{Y^2 - 1} \right] \end{cases} \quad \text{----- (10)}$$

$$\quad \quad \quad \begin{cases} \sigma_a = \frac{P}{Y^2 - 1} \end{cases} \quad \text{----- (11)}$$

$$\text{Cylinder, outside: } \begin{cases} \sigma_r = 0 \\ \sigma_h = \frac{2P}{Y^2 - 1} \end{cases} \quad \text{----- (12)}$$

$$\quad \quad \quad \begin{cases} \sigma_h = \frac{2P}{Y^2 - 1} \end{cases} \quad \text{----- (13)}$$

$$\quad \quad \quad \begin{cases} \sigma_a = \frac{P}{Y^2 - 1} \end{cases} \quad \text{----- (14)}$$

$$\begin{aligned} \text{Sphere, inside:} \quad & \begin{cases} \sigma_r = -P & \text{-----(15)} \\ \sigma_h = \sigma_a = \frac{P}{2} \left[\frac{Y^3 + 2}{Y^3 - 1} \right] & \text{-----(16)} \end{cases} \end{aligned}$$

$$\begin{aligned} \text{Sphere, outside} \quad & \begin{cases} \sigma_r = 0 & \text{-----(17)} \\ \sigma_h = \sigma_a = \frac{3P}{2} \left[\frac{1}{Y^3 - 1} \right] & \text{-----(18)} \end{cases} \end{aligned}$$

A.3.4 Thermal stress due to internal heat generation. In the cylindrical portion of a reactor vessel adjacent to the core, thermal stresses occur due to internal heat generation. The heat generation varies in accordance with $q_x = q_1 e^{-\beta x}$. The temperature of the inner wall is known, and the outer wall is insulated. The maximum tensile and compressive stresses occur at the inner and outer walls respectively. Their magnitudes can be obtained from the curves of Figure A.3-1 which show non-dimensional tangential stress, (η) , plotted as a function of radius ratio, Y , with βR_1 as a parameter.

Stress at the inner or outer wall is

$$\sigma_h = \sigma_a = \eta \left[\frac{12 E \alpha q_1}{(1-\nu) k \beta^2} \right] \text{-----(19)}$$

For thin-walled vessels in which the radius ratio is close to unity and the curves cannot be read with good accuracy, more accurate results can be obtained from the formulas

$$\text{Inside wall, } \sigma_h = \sigma_a = \frac{12 E \alpha q_1}{(1-\nu) k \beta^2} \left[1 - \frac{1}{\beta t} + \left(\frac{1}{\beta t} - \frac{\beta t}{2} \right) e^{-\beta t} \right] \text{(20)}$$

$$\text{Outside wall, } \sigma_h = \sigma_a = \frac{12 E \alpha q_1}{(1-\nu) k \beta^2} \left[-\frac{1}{\beta t} + \left(\frac{\beta t}{2} + 1 + \frac{1}{\beta t} \right) e^{-\beta t} \right] \text{(21)}$$

In the case of a thermal shield the coolant temperatures are equal on the inside and outside. The maximum tensile and compressive stresses can be obtained from equation (19) with the η values obtained from

$$\eta = \eta_1 + \frac{Fk}{12R_1h} \text{-----(22)}$$

where η_1 is obtained from Fig. A.3-2 or A.3-3, and the correction factor, F , is obtained from Fig. A.3-4:

For thin-walled shields the maximum tensile stress is

$$\sigma = \frac{6 E \alpha q_1}{(1-\nu) k \beta^2} \left[1 - \frac{2}{\beta t} + e^{-\beta t} \left(\frac{2}{\beta t} + 1 \right) \right] \text{-----(23)}$$

If the actual distribution of heat generation through the wall thickness cannot be represented well enough by a single term, $q_1 e^{-\beta x}$, it may be considered as the sum of two or more terms, each having its own values for q_1 and β . The stresses calculated for each term are added to give the total stress.

FIG. A.3-1 MAXIMUM NON-DIMENSIONAL STRESS (η)
vs. RADIUS RATIO (Y) WITH βr_1 AS A PARAMETER

FIG. A.3-2 MAXIMUM NON-DIMENSIONAL STRESS (σ_r)
 vs. RADIUS RATIO (Y) WITH (r_1/r_2) AS A PARAMETER
 (For $\nu = 0.05$) ($\nu = 0.05$. See Fig. A.3-3)

FIG. A.3-5 MAXIMUM NON DIMENSIONAL STRESS (τ_1) vs. RADIUS RATIO (Y)
WITH βr_1 AS A PARAMETER (Expansion of Center Portion of Fig.A.3-2)

FIG. A.3-4 CORRECTION FACTOR (F) VS RADIUS RATIO (Y)
WITH βr_i AS A PARAMETER

A.3.5 Thermal stress due to a temperature transient. When a vessel wall is insulated on the outside and is in contact with a fluid on the inside, and the temperature of the fluid increases, linearly with time, by an amount ΔT_f in a time θ seconds, the stress due to the increase in fluid temperature is:

$$\text{Inside} - \sigma_h = \sigma_a = - \frac{E\alpha(\Delta T_f)}{1-\nu} \eta_1 \quad \text{-----} \quad (24)$$

$$\text{Outside} - \sigma_h = \sigma_a = \frac{E\alpha(\Delta T_f)}{1-\nu} \eta_0 \quad \text{-----} \quad (25)$$

Where η_1 and η_0 are stress factors taken from Figures A.3-5 and A.3-6 respectively.

A.4 Sample Calculations

A.4.1 Thermal stress due to internal heat generation. A reactor vessel made of A-302 material has an 80-inch inside diameter and a 4-inch thick wall. It is insulated on the outside. The inside wall temperature is held essentially constant at 450°F. The heat generation at the inner surface is 56.0 Btu/hr in.³ with an absorption coefficient of $\beta = 0.467/\text{in.}$ What maximum thermal stress is produced?

The material properties needed for the solution of the problem are:

$$E\alpha = 210 \frac{\text{psi}}{^\circ\text{F}}, \nu = 0.3 \quad k = 23 \text{ Btu/hr ft } ^\circ\text{F}$$

$$(\beta R_1 = (0.467) (40) = 18.67$$

$$\text{Radius ratio} = \frac{44}{40} = 1.10$$

From Fig. A.3-1, $\eta = 0.42$ for maximum tension at inside wall

$$\frac{E\alpha}{1-\nu} = 300 \frac{\text{psi}}{^\circ\text{F}}, \frac{12q_1}{k\beta^2} = 134^\circ\text{F}$$

Therefore the maximum stress, from equation (10) is

$$\sigma = (300) (134) (0.42) = 16800 \text{ psi}$$

A.4.2 Thermal stress due to a temperature transient. Assume vessel wall 4.0 inches thick of A-302 material, insulated on the outside surface, subjected to a drop in coolant temperature of 20°F in 15 seconds:

$$\begin{aligned} t &= 4.0 \text{ inches} \\ h &= 10,000 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F} \\ k &= 23 \text{ Btu/hr ft } ^\circ\text{F} \\ \epsilon &= 0.018 \text{ in}^2/\text{sec} \\ E\alpha &= 210 \text{ psi/}^\circ\text{F} \\ \nu &= 0.3 \\ \theta &= 15 \text{ sec.} \end{aligned}$$

Fig. A.3-6

STRESS FACTOR FOR OUTSIDE
WALL OF CYLINDRICAL VESSEL
SUBJECTED TO A CHANGE IN
INTERNAL FLUID TEMPERATURE

$$\frac{12k}{ht} = 0.0069$$

$$\lambda = \frac{\epsilon \theta}{t^2} = 0.0169$$

From Figures A.3-5 and A.3-6 with $\frac{12k}{ht} = 0.0069$ and $\lambda = 0.0169$,
 $\eta_i = 0.89$ and $\eta_o = -0.09$ (from curve for $\frac{12k}{ht} = 0.0$)

$$\text{From equation (24), } \sigma_i = \frac{210}{0.7} (20) (0.89) = 5340 \text{ psi (tension)}$$

$$\text{From equation (25), } \sigma_o = \frac{210}{0.7} (20) (-0.09) = -540 \text{ psi (compression)}$$

For a 20°F drop in 60 seconds:

$$\lambda = 0.067$$

$$\sigma_i = \frac{210}{0.7} (20) (0.785) = 4710 \text{ psi (tension)}$$

$$\sigma_o = \frac{210}{0.7} (20) (-0.183) = -1100 \text{ psi (compression)}$$

Data are not available for values of η_o at finite heat transfer rates. Generally speaking, the error in using the curve for $\frac{12k}{ht} = 0.0$ will be small and on the conservative side. Therefore, this curve may be used for all cases when evaluating the stress at the outside surface of a simple cylinder under temperature transients.

A.5 Stresses From Radial Loads and External Moments Acting on Spherical and Cylindrical Shells

A.5.1 Professor P. P. Bijlaard, of Cornell University, has published a number of papers on this subject as the result of research sponsored by the PVRC. The papers most pertinent to this application are listed in A.5.7. The remainder of this Appendix section contains a discussion of his results in order that the designer may work directly from his summary curves.

A.5.2 In the stress analysis of shells, one obtains the stresses from the following relation:

$$\sigma_1 = K_N \frac{N_1}{t} \pm K_M \frac{6M_1}{t^2} \quad \text{-----} \quad (\text{A.5-1})$$

where

σ_1 is the stress acting in the 1 direction.

N_1 is the membrane force per unit length acting in the 1 direction.

M_1 is the bending moment per unit length acting in the 1 direction.

t is the thickness of the shell.

K_N is the stress concentration factor, if required, on the membrane stress (see A.5.6).

K_M is the stress concentration factor, if required, on the bending stress (see A.5.6).

Since it is generally necessary at some stage of the problem to apply unequal values of K_N and K_M , it is not useful to combine the two sources of stress in a general solution. In addition, it is often necessary to evaluate the stress on both the inner and outer surfaces so that both signs must be considered with respect to the bending stresses.

A.5.3 In the case of external loads on a spherical shell (which method is also applicable to hemispherical and ellipsoidal heads or a cylindrical shell), one must consider two directions for 1. In addition, in the case of moment loadings, the stresses will have a different sign on one side of the nozzle than on the other.

A.5.4 Bijlaard's summary curves, contained in the references listed in A.5.7, plot non-dimensional membrane and bending stress components in terms of other dimensionless parameters. The geometric factors considered are

the shell and nozzle radii, thickness, and given shell length to radius ratios. The effect of the exact ratio of shell length to radius is small and need not be considered. Bijlaard has also considered the effect of applied internal pressure upon the stresses due to external loads. These effects are not significant in spherical shells or in cylindrical shells whose radius to thickness ratio is less than fifty.

A.5.5 The information contained in this paragraph is useful in evaluating the maximum stresses which result from external loadings in order to check compliance with the allowable stress intensity S_p . If information regarding the membrane stress and bending stress components for the spherical shell is desired, it must be obtained from References A.5.5.1, A.5.5.2 and A.5.5.6 directly. For the cylindrical shell, this section contains curves for the membrane and bending components. This procedure assumes that the attachment is rigid and that the shell is uniform in thickness in the vicinity of the nozzle and that the nozzle is sufficiently far removed from the ends of the shell.

A.5.5.1 Spherical Pressure Vessels and Heads

A.5.5.1.1 General. The results published by Bijlaard for external loads and moments acting on a spherical shell consist of curves for bending moments and membrane forces. The maximum stress occurs at the intersection of the attachment and the shell. With this in mind, the bending moments and membrane forces were combined by the use of relation A.5-1 for both the external moment and external load. The results which are presented in Figure A.5-2 are expressed in terms of the parameter S , given by:

$$S = 1.82 \frac{r_o}{R} \sqrt{\frac{R}{t}} \quad \text{----- (A.5-2)}$$

where

r_o is the outer radius of the attachment

R is the radius of the middle surface of the shell

t is the thickness of the shell

Note: It should be pointed out that the method to be explained below applies also to ellipsoidal heads. In such a case, R would be the radius of the shell at the point of attachment of the nozzle.

Figure A.5-1 shows the above dimensions on a typical spherical shell as well as the mode of application of the external moments and loads as considered here.

FIG. A. 5-1 SPHERICAL SHELL SUBJECTED TO BOTH A RADIAL LOAD P AND AN EXTERNAL MOMENT M ACTING ON A PIPE, CONSIDERED AS A RIGID CYLINDRICAL BODY

A.5.5.1.2 Calculation of the Stress due to a Radial Load P Acting on a Spherical Shell (Refer to Fig. A.5-2)

- a. Calculate the value of the parameter S from relation A.5-2.
- b. Enter Fig. A.5-2 at the value of S calculated above, and using the curve entitled "Radial Load P", determine the value of the ordinate $\frac{\sigma t^2}{P}$
- c. Using the desired value of the load P and thickness t, multiply $\frac{\sigma t^2}{P}$, as found in the previous step, by $\frac{P}{t^2}$ and obtain the maximum stress due to a load P. This stress will be constant around the intersection between the shell and the nozzle. It is compressive when P acts toward the shell, and it is tensile when P acts away from the shell.

A.5.5.1.3 Calculation of the Stress due to an External Moment M Acting on a Spherical Shell. (Refer to Fig. A.5-2)

- a. Calculate the value of the parameter S from relation A.5-2.
- b. Enter Fig. A.5-2 at the value of S calculated above, and using the curve entitled "External Moment M", determine the value of the ordinate $\frac{\sigma t^2}{\frac{M}{R} \sqrt{\frac{R}{t}} \cos \theta}$

- c. Using the desired value of the moment M, the thickness t, and the shell radius R, multiply $\frac{\sigma t^2}{\frac{M}{R} \sqrt{\frac{R}{t}} \cos \theta}$ as found in the previous step, by $\frac{M}{Rt^2} \sqrt{\frac{R}{t}} \cos \theta$ and obtain the maximum stress

due to an external moment M. As seen in the above, the stress varies around the intersection of the shell and the nozzle in proportion to $\cos \theta$. For a definition of the angle θ , see Fig. A.5-1. The maximum stress due to a given moment occurs at the outside of the shell at $\theta = 0$ and $\theta = \pi$. At $\theta = 0$, the stress is compressive whereas at $\theta = \pi$, the stress is tensile.

A.5.5.2 Cylindrical Pressure Vessels

A.5.5.2.1 General. The results published by Bijlaard for external loads and moments acting on cylindrical pressure vessels (3) (4) (5) did not lend themselves to a simple combination by the use of relation A.5-1, which would allow presentation of curves for the maximum stress values, as was done for spherical shells in paragraph A.5.5.1. Furthermore, the direction of the maximum stress due to external moment was not evident as in the previous case. As a result of these considerations, one must utilize all of the curves published by Bijlaard for all the bending moments and membrane forces caused by an external moment.

A further consideration is that Bijlaard calculated his data for two kinds of moment acting on a cylindrical shell. These are a circumferential moment acting on the shell and a longitudinal moment acting on the shell. It is also not evident from the data if the stresses from one of these is consistently higher than the stresses from the other.

For the cylindrical shell, Bijlaard's curves are presented in terms of the parameters β and γ which are given by:

$$\beta = \frac{0.875r_o}{a} \quad \text{----- (A.5-3)}$$

$$\gamma = a/t \quad \text{----- (A.5-4)}$$

where

a is the radius of the middle surface of the shell

r_o is the outer radius of the attachment

t is the thickness of the shell

Figure A.5-3 shows the above dimensions on a typical cylindrical shell as well as the mode of application of the external moments and loads considered here.

A.5.5.2.2 Calculation of the Maximum Stress Due to a Radial Load P Acting on a Cylindrical Shell (Refer to Figs. A.5-4 and A.5-5)

1. Calculate the value of β and of γ using equations A.5-3 and A.5-4 respectively.
2. Enter Fig. A.5-4 at the value of β calculated above and using the appropriate curve for the value of γ read the value of $\frac{M}{P} \phi$.
3. By a procedure similar to that employed in step 2 above, find $\frac{N \phi}{P/a}$ from Fig. A.5-5.
4. Multiply $\frac{M}{P} \phi$ by $\frac{6P}{t^2}$ and multiply $\frac{N \phi}{P/a}$ by $\frac{P}{at}$, using the desired values of P , a , and t . This yields $\frac{N \phi}{t}$ and $\frac{6M}{t^2} \phi$.
5. Combine $\frac{N}{t} \phi$ and $\frac{6M}{t^2} \phi$ by means of relation A.5-1 to yield the maximum stress in a cylindrical vessel due to a load P . The maximum stress is compressive when P acts toward the shell, and it is tensile when P acts away from the shell.

(A) LONGITUDINAL MOMENT M AND RADIAL LOAD P

(B) CIRCUMFERENTIAL MOMENT M AND RADIAL LOAD P

FIG. A.5-3 CYLINDRICAL SHELL SUBJECTED TO AN EXTERNAL MOMENT AND A RADIAL LOAD ACTING ON A PIPE,

FIGURE A-5-4

BENDING MOMENT $\frac{M_\phi}{P}$ DUE TO AN
EXTERNAL RADIAL LOAD P ON A
CIRCULAR CYLINDER

FIGURE A.5-5

MEMBRANE FORCE $\frac{N\phi}{P/a}$ DUE TO AN
EXTERNAL RADIAL LOAD P ON A
CIRCULAR CYLINDER

A.5.5.2.3 Calculation of the Maximum Stress Due to an External Moment M Acting on a Cylindrical Shell (Refer to Figs. A.5-6 thru A.5-13)

NOTE: The following procedure must be repeated twice. It is done once to determine the maximum stress due to an external longitudinal moment M, and a second time to determine the maximum stress due to an external circumferential moment M. The greater of these two stresses will represent the worst effect that could be produced by any given external moment.

In the following, the figures mentioned without parentheses refer to the external longitudinal moment and those figures mentioned in parenthesis refer to the external circumferential moment. The procedure to be followed is the same in both cases.

1. Calculate the value of β and of γ using relations A.5-3 and A.5-4 respectively.
 2. Enter Fig. A.5-6 (A.5-10) at the value of β calculated above, and using the appropriate curve for the value of γ read the value of $\frac{M_x}{M/a\beta}$.
 3. By a similar procedure, find $\frac{M_\phi}{M/a\beta}$, $\frac{N_x}{M/a^2\beta}$, and $\frac{N_\phi}{M/a^2\beta}$ using Figures A.5-7 (A.5-11); A.5-8 (A.5-12); A.5-9 (A.5-13), respectively.
 4. Multiply $\frac{M_x}{M/a\beta}$ and $\frac{M_\phi}{M/a^2\beta}$ by $\frac{6M}{a\beta t^2}$, using the appropriate values of M, a, t, and β , to yield values of $\frac{6M_x}{t^2}$ and $\frac{6M_\phi}{t^2}$.
 5. Multiply $\frac{N_\phi}{M/a^2\beta}$ and $\frac{N_x}{M/a^2\beta}$ by $\frac{M}{ta^2\beta}$ using the appropriate values of M, a, t, and β to yield values of $\frac{N_x}{t}$ and $\frac{N_\phi}{t}$.
 6. Using relation A.5-1 combine $\frac{N_\phi}{t}$ and $\frac{6M_\phi}{t^2}$ as well as $\frac{N_x}{t}$ and $\frac{6M_x}{t^2}$ to yield σ_ϕ and σ_x . The greater of these is selected as the maximum stress due to a longitudinal (circumferential) moment.
- (Recall that the procedure must be repeated for both types of moment to determine the maximum stress).
- The sign of the maximum is determined as follows (see Fig. A.5-3). At points Q₁ and R₁ the applied moment has the same effect as a load toward the shell; hence the induced membrane forces and moments both lead to compressive stresses on the outside of the shell. At points Q₂ and R₂ the applied moment has the same effect as a load away from the shell; hence the induced membrane forces and moments both lead to tensile stresses on the outside of the shell.

FIGURE A.5-6

FIGURE A.5-7

MOMENT $\frac{M\phi}{M_0\beta}$ DUE TO AN
EXTERNAL LONGITUDINAL MOMENT M ON A
CIRCULAR CYLINDER

FIGURE A.5-8

FIGURE A.5+9

MEMBRANE FORCE $\frac{N_\phi}{M/a^2\beta}$ DUE TO AN
EXTERNAL LONGITUDINAL MOMENT M ON A
CIRCULAR CYLINDER

FIGURE A.5-10

MOMENT $\frac{M_x}{M/\alpha\beta}$ DUE TO AN
EXTERNAL CIRCUMFERENTIAL MOMENT M ON A
CIRCULAR CYLINDER

FIGURE A.5-11

FIGURE A.5-12

FIGURE A.5+13

A.5.5.2.4 Limitations on the Application of the Preceding

1. Any External Moment Acting on a Circular Cylinder. It should be pointed out that Bijlaard's calculations were made on the basis that the moment acts on an attachment at the center of the length of the cylinder. His results are applicable to an "off center" attachment (a more usual case) provided that the attachment is at least a distance equal to one half of the shell radius away from the end of the cylinder.

Furthermore, the results are applicable only to cylindrical shells whose length is at least equal to their radius.

2. External Load Acting on a Circular Cylinder. There is no restriction on the point of application of the load in this instance.

The stresses are affected very little by the ratio of shell length to shell radius. Bijlaard has found that a change in this ratio never affects the stresses by more than 10%. The curves we have presented for this loading are exact for the ratio equal to 8. Those who wish to make a correction in the case of other values of the length to radius ratio are referred to correction curves given on page 8 of reference (4).

A.5.6 The stress concentration factors to be used in evaluating the effects of pipe reactions are discussed in paragraph A.7.3.

A.5.7 BIBLIOGRAPHY

1. P.P. Bijlaard, "Computation of the Stresses from Local Loads in Spherical Pressure Vessels or Pressure Vessel Heads", Welding Research Council Bulletin Series, No. 34, March, 1957.
2. P.P. Bijlaard, "Local Stresses in Spherical Shells from Radial or Moment Loadings", The Welding Journal Research Supplement, May, 1957.
3. P.P. Bijlaard, "Stresses from Local Loadings in Cylindrical Pressure Vessels", Transactions of the American Society of Mechanical Engineers, Vol. 77, August, 1955, pp. 805-816.
4. P.P. Bijlaard, "Stresses from Radial Loads in Cylindrical Pressure Vessels", The Welding Journal Research Supplement, December, 1954.
5. P.P. Bijlaard, "Stresses from Radial Loads and External Moments in Cylindrical Pressure Vessels", The Welding Journal Research Supplement, December, 1955.

6. P.P. Bijlaard, "On the Stresses in Spherical Vessels from Local Loads Transferred by a Pipe", September, 1958. Report to PVRC (to be published). This paper presents summary curves without derivation for all spherical shell considerations to date, including consideration of the pipe flexibility.
7. P.P. Bijlaard, "Additional Data on Stresses in Cylindrical Shells Under Local Loading". Report to PVRC (to be published). This paper presents summary curves without derivation for all cylindrical shell considerations to date, including the stress away from the point of attachment but not including pipe flexibility.

A.6 Thermal Stress at Nozzles

A.6.1 An outline is given for the method of calculating thermal stresses at nozzles.

FIG. A.6-1a GIVEN NOZZLE CONFIGURATION

FIG. A.6-1b FREE-BODY DIAGRAM
(POSITIVE DIRECTIONS SHOWN)

Nomenclature to be used in this section is as follows:

- R_v = Inside radius of hemispherical or cylindrical shell, in.
 T = Thickness of head, in.
 R_1 = Inside radius of nozzle, in.
 R_o = Outside radius of nozzle, in.
 t = Thickness of nozzle
 M = axisymmetric discontinuity moment, in-lb/in.
 V = axisymmetric discontinuity force, lb/in.
 δ = radial displacement, in., positive as shown
 θ = angular rotation, positive as shown
 S, R, C, T = subscripts for shell, ring, cylinder, & thermally induced
 E = Young's modulus, psi
 σ_x = axial stress
 σ_y = circumferential (hoop) stress

A.6.2 Calculation Procedure

A.6.2.1 On the basis of average temperatures and assumed linear gradients, calculate the deformations in each of the three bodies: δ_{ST} , θ_{ST} , δ_{RT} , θ_{RT} , δ_{CT} , θ_{CT} .

A.6.2.2 Write the equations for each of the three bodies for deformation in terms of the discontinuity reactions.

(a) For the cylinder, assuming Poisson's ratio = 0.3:

$$\delta_C = \frac{2 R_1 + t}{0.605 E t^2} \left[M_C - 0.550 \sqrt{(2 R_1 + t) t} V_C \right]$$

$$\theta_C = \frac{2 R_1 + t}{0.605 E t^2} \left[\frac{3.64 M_C}{\sqrt{(2 R_1 + t) t}} - V_C \right]$$

(b) For the ring:

$$\delta_R = \frac{R_R}{A E} \left[V_C \left(R_1 + \frac{t}{2} \right) - V_S R_S \right]$$

$$\theta_R = \frac{M R^2}{E I} = \frac{-R_R}{E I} \left[(M_C + L_C V_C) \left(R_1 + \frac{t}{2} \right) + (M_S + L_S V_S) R_S \right]$$

where

R_S is the radius at which V_S acts, in.

R_R is the radius to the centroid of the ring cross-section, in.

A is the cross-sectional area of the ring, in.²

I is the moment of inertia of the ring cross-section with respect to the neutral axis.

M is the moment about the neutral axis from V_C , M_C , V_S , M_S .

I_C , I_S are as shown on Fig. A.6-1 (b).

(c) For the shell:

It is assumed for purposes of analysis that the shell is hemispherical. Using the paper "Influence Coefficients for Hemispherical Shells with Small Openings at the Vertex", by G. D. Galletly, (Trans ASME*), for the Geckeler Approximation.

$$\delta_S = \frac{2\lambda \cos \alpha}{ET} \left[\left(R_V + \frac{T}{2}\right) (\cos \alpha) V_S - \lambda M_S \right]$$

$$\theta_S = \frac{2\lambda^2}{ET} \left[-V_S \cos \alpha + \frac{2\lambda M_S}{\left(R_V + \frac{T}{2}\right)} \right]$$

where

$$\lambda = \left[\frac{2.73 \left(R_V + \frac{T}{2}\right)^2}{T^2} \right]^{\frac{1}{4}}$$

$$\alpha = \frac{\pi}{2} - \phi_0$$

$2\phi_0$ = total central angle subtended by the opening

$$\therefore \phi_0 = \sin^{-1} \frac{R_S}{R_V + \frac{T}{2}}$$

It should be noted that this method is approximate. When more accurate results are required, Method II of Galletly's paper should be used.

A.6.2.3 Solve the compatibility equations of the structure for the four unknowns V_C , M_C , V_S , M_S . In writing these equations it should be noted that whereas the calculated displacements and rotations are at the cut surface for the shell and the cylinder, they refer to the centroid of the ring. Hence, where I_C and I_S are as defined on Fig. A.6-1(b), the compatibility equations are:

$$\delta_C + \delta_{CT} = \delta_R + \delta_{RT} - I_C (\theta_R + \theta_{RT})$$

$$\delta_S + \delta_{ST} = \delta_R + \delta_{RT} + I_S (\theta_R + \theta_{RT})$$

* Journal of Applied Mechanics, March, 1955, pp. 20-24.

$$\theta_C + \theta_{CT} = \theta_R + \theta_{RT}$$

$$\theta_S + \theta_{ST} = \theta_R + \theta_{RT}$$

A.6.2.4 Calculate the discontinuity stresses which exist in the three components.

(a) For the cylinder, at the end

The axial stress, σ_x' is

$$\sigma_{xC}' = \pm \frac{6 M_C}{t^2}, \text{ where the upper sign refers to the outer surface.}$$

The hoop stress, σ_y' is

$$\sigma_{yC}' = \frac{3.30}{t^2} \left[M_C - 0.550 \sqrt{(2R_1 + t)t} V_C \right] + 0.3 \sigma_{xC}'$$

(b) For the ring, which is considered to rotate about its centroid so that for a positive moment M , all points to the left of the neutral axis are in tension and all points to the right of this axis are in compression; the hoop discontinuity stress is

$$\sigma_{yR}' = \frac{x R_R M}{I} = - \frac{x}{I} \left[(M_C + I_C V_C) \left(R_1 + \frac{t}{2} \right) + (M_S + I_S V_S) R_S \right]$$

where x is the distance from the neutral axis at which the point considered is located.

(c) For the shell, at the junction with the ring:

$$\sigma_{xS}' = \pm \frac{6 M_S}{t^2}, \text{ where the upper sign refers to the outer surface.}$$

$$\sigma_{yS}' = \frac{2 \lambda \cos \alpha}{T(R_V + \frac{T}{2})} \left[(R_V + \frac{T}{2})(\cos \alpha) V_S - \lambda M_S \right] + 0.3 \sigma_{xS}'$$

A.6.2.5 Calculate the "free-body" stresses which exist in each of the three components. The deformations δ_{ST} , θ_{ST} , δ_{RT} , θ_{RT} , δ_{CT} , θ_{CT} were calculated on the basis of the average temperature and a linear gradient in the average temperature in the direction of the nozzle centerline. The "free-body" stresses result from the point being considered as having a different temperature than the average temperature at the cross-section considered. If this temperature difference from average is designated as ΔT , the "free-body" stress is

$$\sigma_x'' = \sigma_y'' = 1.43 E \alpha \Delta T$$

A.6.2.6 Calculate the total thermal stress which is the sum of the discontinuity thermal stress and the "free-body" thermal stress with consideration given to the appropriate stress concentration factor. Stress concentration factors for bending at the outer fillet radius may be determined from the method presented in paragraph A.7.2.6. Note that a stress concentration factor must be applied both to the cylinder, K_C , and to the shell, K_S . This is an approximate procedure which is necessitated by the inability of the ring analysis method to account for the fillet effects.

The total stresses are then:

(a) In the cylinder

$$\sigma_{xC} = \mp \frac{6K_C M_C}{t^2} + \sigma_{xC}''$$

$$\sigma_{yC} = \frac{3.30}{t^2} \left[M_C - 0.550 \sqrt{(2R_1 + t)t} V_C \right] + 0.3 \sigma_{xC} + \sigma_{yC}''$$

(b) In the ring

$$\sigma_{xR} = + \frac{6K_C M_C}{t^2} \text{ at cylinder, or } = - \frac{6K_S M_S}{T^2} \text{ at shell}$$

$$\sigma_{yR} = - \frac{x}{I} \left[(M_C + L_C V_C) \left(R_1 + \frac{t}{2} \right) + (M_S + L_S V_S) R_S \right] + 0.3 \sigma_{xR} + \sigma_{yR}''$$

(c) In the shell

$$\sigma_{xS} = - \frac{6K_S M_S}{T^2} + \sigma_{xS}''$$

$$\sigma_{yS} = \frac{2 \lambda \cos \alpha}{T(R_V + \frac{T}{2})} \left[(R_V + \frac{T}{2})(\cos \alpha) V_S - \lambda M_S \right] + 0.3 \sigma_{xS} + \sigma_{yS}''$$

A.6.3 Cylindrical Shell Stresses due to End Moment and Shear

A.6.3.1 The calculation of discontinuity effects in structures made up of cylindrical shells results in known values of the axisymmetric end shear Q (lb/in) and moment M (in-lb/in). In most cases the important stress conditions occur at the end of the cylinder, particularly when stress concentration effects are important. However, there are important cases when the maximum stress does not occur at the edge or when the stress must be determined at other locations.

A.6.3.2 The location of the maximum stress may be determined from the following equations:

$$\text{Axial Stress:} \quad \lambda x = \cot^{-1} \left[1 + \frac{2\lambda M}{Q} \right] \text{ or at } x = 0$$

$$\text{Hoop Stress (Membrane):} \quad \lambda x = \tan^{-1} \left[\frac{-2\lambda M}{Q} - 1 \right] \text{ or at } x = 0$$

(Note that the two conditions must be stated because the function is not defined for negative values of x).

Q and M are positive if directed outward and

$$\lambda^{-1} = 0.778 \sqrt{(\text{Mean Radius}) \times (\text{Thickness})}$$

A.6.3.3 The stress at any distance from the end, x , may be evaluated from the equations:

$$\sigma_x = \frac{6}{t^2} e^{-\lambda x} \left[M(\cos \lambda x + \sin \lambda x) + \frac{Q}{\lambda} \sin \lambda x \right]$$

$$\sigma_h = \frac{2\lambda^2 R}{t} e^{-\lambda x} \left[M(\cos \lambda x - \sin \lambda x) + \frac{Q}{\lambda} \cos \lambda x \right] + 0.3 \sigma_x$$

The functions required are tabulated in convenient form in the book of "Beams on Elastic Foundations", by M. Hetenyi, University of Michigan Press, 1946.

A.6.4 Influence Coefficients for Short Cylindrical Shells

A.6.4.1 Short shells are those which are too long to be considered rings and too short to be considered semi-infinite cylinders. These may be distinguished as follows:

- (a) If $0 < \lambda L \leq 0.75$ ring equations apply.
- (b) If $0.75 < \lambda L < 2.5$ short shell equations apply,
- (c) If $2.5 \leq \lambda L < \infty$ long shell equations apply.

A.6.4.2 Figure A.6-2 gives the influence coefficients for a short shell loaded by the axisymmetric shear, Q , and moment, M , at one end and free at the other. These results are taken from the paper "Tables of Functions for Short Cylindrical Shells", by R. B. McCalley, Jr. and R. G. Kelly (ASME Paper No. 56-F-5).

A.6.5 Thermal Stress in Bi-Metallic Welds

A.6.5.1 A bi-metallic weld is defined (see 3.5.4) as a weld which joins two materials whose thermal coefficients of expansion differ by more than 15%.

FIGURE A.6-2
SHORT SHELL COEFFICIENTS
($\nu=0.3$)

A.6.5.2 When the weld and the mating members are isothermal, thermal stresses result from:

A.6.5.2.1 The tendency for the material on one side of the weld to have a different thickness than the material on the other side.

A.6.5.2.2 The tendency for the material on one side of the weld to be displaced from the material on the other side. For example, in a pipe the relative displacement is the product of the mean radius, the temperature change, and the difference between the two thermal coefficients of expansion.

A.6.5.4 The problem of A.6.5.2.1 may be solved by the method of reference E.2.a. The problems of A.6.5.2.2 and A.6.5.3 are solved by standard discontinuity analysis.

A.7 Stress Concentration Factors

A.7.1 Curves for the determination of stress concentration factors for recurring cases are developed in paragraph A.7.2. Their application to particular problems is discussed in the remaining paragraphs under A.7. Other stress concentration factors may be obtained from:

A.7.1.1 "Stress Concentration Design Factors", by R. E. Peterson, John Wiley and Sons, Inc., New York, 1953.

A.7.1.2 "Designing by Photoelasticity", by R. B. Heywood, Chapman and Hall, London, 1952.

A.7.1.3 Technical papers and texts. In particular, the Proceedings of the Society for Experimental Stress Analysis.

A.7.2 The following stress concentration effects occur commonly:

A.7.2.1 Stepped bar in tension. Consider a flat bar of unit thickness subjected to a tensile load. Designate the width of the left end of the bar by D and the width of the right end of the bar by d and assume that $d < D$. Assume that there is a step change in thickness at the middle of the bar except for a shoulder radius r . Figure 57* of Reference A.7.1.1 then gives the stress concentration factor as a function of r/d for values of D/d between 1.01 and 3.0.

A.7.2.2 Stepped bar in bending. Consider the bar of A.7.2.1 to be subjected to an in-plane bending moment. Figure 60* of Reference A.7.1.1 gives the stress concentration factor for values of D/d between 1.01 and 6.0.

A.7.2.3 The cases considered in A.7.2.1 and A.7.2.2 are also of interest when D becomes very large compared to d . Figure A.7-1 shows stress concentration factors for this case. (These curves were derived from information in Reference A.7.1.2, pages 178-180).

A.7.2.4 The case of an inclined shoulder is also of interest, and is discussed in Reference A.7.1.2, page 179. If the stress concentration factor, as obtained from Figures 57 and 60 of Reference A.7.1.1 and Figure A.7-1 is designated as K_0 , and the stress concentration factor for the tapered shoulder case as K' , the following relationships apply:

1. If $r < h$

$$\frac{K'-1}{K_0-1} = 1 - \left[\frac{\beta}{90} \right]^{1 + 2.4 \sqrt{\frac{r}{h}}}$$

where β is the angle between the tapered shoulder and the square shoulder. (See Figure A.7-2). In using this relationship, the arbitrary upper limit of $K_0 = 5$ should not be used.

*When using Figures 57 and 60 of Reference A.7.1.1, it should be noted that the symbol t used in the reference corresponds to the symbol h used in this text.

2. If $r > h$

$$\frac{K'-1}{K_0-1} = 1 - \left[\frac{\beta - \alpha}{90 - \alpha} \right]^{1 + 2.4 \sqrt{\frac{r}{h}}}$$

$$\text{where } \alpha = \sin^{-1} \left[1 - \frac{h}{r} \right]$$

3. If $D \gg d$, for use with Figure A.7-1

$$\frac{K'-1}{K_0-1} = 1 - \frac{\beta}{90}$$

The quantities on the left hand side of the above equations are plotted in Figure A.7-2 versus the quantity on the right hand side.

A.7.2.5 The following procedure shall be used if the left hand end of the taper (see Figure A.7-2) terminates in a plate of width D_1 , which is greater than the width of the taper, D . (See, for example, the nozzle connection shown in Figure A.7-3(b)). The fillet radius at that junction will be designated by r_p .

- Calculate the stress concentration factor that would exist if the bar to the right of the junction between the plate and the taper were of uniform width D .
- Account for the taper by using the quantity $(90 - \beta)$ in place of β in the abscissa of Figure A.7-2.

A.7.2.6 The above methods may also be applied to calculate the stress concentration factor when the bar is offset only on one side, by using the method of reflection. Consider a bar of width T to the left and width t to the right such that $T > t$ and that one edge of the bar is a straight line. Then $h = T - t$ on one side and zero on the other. This case is solved by using $D = 2T$ and $d = 2t$ in Figures 57 and 60 of Reference A7.1.1 and Figures A.7-1 and A.7-2.

A.7.3 Application to External Loads on a Nozzle.

A.7.3.1 The stress concentration factor for the axial load and for the bending moments at the junction of the pipe and the nozzle (Point A in Figure A.7-3) may be obtained from Figures 57 and 60 of Reference A.7.1.1 and Figure A.7-2 in terms of D , d , r_A , and β .*

A.7.3.2 The stress concentration factors at Point B in Figure A.7-3(a) may be found from Figure A.7-1 in terms of $t = 0.5D$ and r_p . The factors at point B in Figure A.7-3(b) may be found from Figures A.7-1 and A.7-2 (see

*Since less data are available for stress concentration factors in shafts than in plates, it is recommended in Ref. A.7.1.2, page 195, that plate data be used.

FIGURE A.7-3 NOZZLE NOMENCLATURE

FIGURE A.7-4 EXAMPLE-STEPPED CYLINDER

paragraph A.7.2.5) in terms of $t = 0.5D$ and r_p . These two stress concentration factors apply to the bending and axial stresses which would result from the application of the external loads to a pipe of outside diameter D and inside diameter d_1 .

A.7.3.3 The stresses at Point C in Figure A.7-3 to which stress concentration factors must be applied are calculated in accordance with paragraph A.5. These have been derived in terms of a membrane component (N/t) and a bending component ($6M/t^2$). The stress concentration factor to be applied to the bending component is that obtained for the bending case from Figure A.7-1 with $\frac{r}{t} = \frac{r_B}{t}$. The factor for the membrane component is determined from the same figure for the tensile case.

A.7.4 Application to Primary Stresses. Stress concentration factors must be applied to primary stresses when these stresses are perpendicular to a change in section for use in fatigue analysis. An example is given in paragraph A.7.6.

A.7.5 Application to Secondary Stresses. Stress concentration factors must be applied to secondary stresses when these stresses are perpendicular to a change in section for use in fatigue analysis. An example is given in paragraph A.7.6.

A.7.6 Example. Consider the stepped cylinder sketched in Figure A.7-4 when subjected to internal pressure, P .

A.7.6.1 The primary axial stress and hoop stress are σ_{a1} , σ_{a2} , σ_{h1} , σ_{h2} . However, since σ_{a1} acts across the step, a stress concentration factor (see paragraph A.7.2.6) must be applied to that component.

A.7.6.2 Discontinuity stresses exist both because of the step in the wall centerline, which results in a moment $\frac{1}{2} PR_1(T-t)$, and due to the fact that the radial deflection of one side would differ from that of the other if they were not joined. As a result, there is a discontinuity shear force and a bending moment which must be applied to each cylinder. Since the inner surface is continuous, there is no stress concentration factor on that surface. The method of paragraph A.7.2.6 must be applied to determine the stress concentration factor to be applied to the stresses due to the moment on the outer surface.

A.7.7 As stated in paragraph 3.1.3, stress concentration factors need be taken no greater than 5.0. However, this factor must be considered in relation to the immediately surrounding stress field, since under certain circumstances, stress concentration factors may be cumulative.

A.7.7.1 Welding Factors. Stress concentration factors determined by the methods described in this section are to be used with the assumption that radiography and other inspection methods have determined that no significant defects exist. When the inspection is not sufficient to give this assurance, the acceptability of the design may be considered adequate if the fatigue criteria are satisfied after use of a stress concentration factor of 5. Lower values of the fatigue-weakening factor may be used if experimental fatigue data can be cited.

A.8 Sample Calculation - Stress Analysis at a Nozzle

A.8.1 Introduction. The purpose of this sample calculation is to demonstrate the application of this design and evaluation method in a practical and ordinarily difficult design problem. In order to provide a complete example, it will be necessary to establish certain arbitrary, and perhaps impracticable, thermal conditions. The significance of these stresses with respect to fatigue is evaluated in paragraph A.9.

A.8.2 Statement of Problem. Design and evaluate a pressure vessel nozzle to meet the following conditions.

1. Material - A302B clad with 0.25 inch of AISI Type 304 stainless steel.
2. Design pressure - 2500 psi.
3. Design temperature - 600°F.
4. Minimum internal diameter cylindrical pressure vessel - 60 in.
5. Minimum internal diameter of pipe - 10 in.
6. Determine the maximum allowable number of cycles that the nozzle can withstand if the only operation is start-up from 70°F to 570°F at a rate of 500°F/hr and shut-down at the same rate to 70°F.
7. The pipe reactions, which result from thermal expansion of the piping, are

<u>Load</u>	<u>Value at</u>	
	<u>70°F</u>	<u>570°F</u>
Axial Thrust (positive inward)	0	150,000 lb.
Circumferential moment in a plane perpendicular to shell centerline	0	2,000,000 in-lb

8. The problem of A.8.2.6 will be evaluated from the fatigue standpoint in A.9 for the point on the inside of the vessel at the junction of the ring and the shell, although the stresses at all significant points in the vicinity of the nozzle will be calculated.

A.8.3 Determination of Basic Thicknesses, Use of S_m

A.8.3.1 The equation for required thickness, t_r is given by A.3.1 (1), Equation (1) as

$$t_r = \frac{2PR_1}{2S_m - P}$$

In general, $P = 2500$ psi from A.8.2.2

$S_m = 27000$ psi from A.8.2.1, 3.6.a and Table 5-1

So that

$$t_r = \frac{2(2500) R_1}{2(27000) - 2500} = 0.0972 R_1$$

A.8.3.2 For the shell, since $R_{1s} = 30$ (by A.8.2.4 and 3.6.a),

$$0.0972 (30) = 2.91 \text{ in.}$$

A.8.3.3 For the pipe, since $R_{1p} = 5$

$$t_{rp} = 0.0972 (5) = 0.485 \text{ in.}$$

A.8.3.4 Since, by A.8.2.1 the vessel is to be clad with 0.25 Type 304, and since practical numbers would be used, we have

Component	I.R. (total)	O.R. (total)	Thickness (total)	I.R. (base)	O.R. (base)	Thickness (base)
Shell	30	33.25	3.25	30.25	33.25	3.00
Pipe	5	5.75 May be thicker to match pipe	0.75	5.25	5.75	0.500

These values are, of course, subject to change by later considerations.

A.8.4 Determination of Nozzle Configuration

A.8.4.1 Required compensation for opening (see 6.6.1 and Appendix C).
By ASME Boiler Pressure Vessel Code, paragraph UG-37, the required area, A_r , is (see C.1.2)

$$A_r = d t_{rs} = 10.5 (2.91) = 30.5 \text{ in.}^2$$

A.8.4.2 By C.1.2 and ASME Boiler Pressure Vessel Code, paragraph UG-40, the added material must fall within an area bounded by a radius from the pipe centerline equal to 10.5 in. and by a radius from the shell centerline, R_R , equal to the smaller of

$$(a) 33.25 + 2.5 (3.00) = 40.75$$

$$(b) 33.25 + 2.5 (0.500) + 2.5 \Delta t_p + \Delta t_s = 34.5 + 2.5 \Delta t_p + \Delta t_s$$

where Δt_p is any uniform thickness added to pipe

Δt_s is any uniform thickness added to shell

FIG. A.8.4.2-1

A.8.4.3 Compensation Design

- (a) Arbitrarily and so that C.3.1 may be illustrated, we choose to increase the base metal thickness to a value of 4.0 in., then $\Delta t_g = 1.0$. This thickness must be maintained completely around the circumference and for a distance of $1.5(10.5) = 15.75$ in. above and below the pipe centerline. By 6.4.2, there should be a taper for $3(1) = 3$ in. beyond this distance back to the 3.0 in. base metal thickness, and the fillet radius between the taper and the shell should be at least $0.50(3.0) = 1.5$ in.

- (b) Arbitrarily the remaining reinforcement will be added to the pipe wall, as a uniform thickness increase, Δt_p . Then

$$\frac{A_r}{2} = A_1 + A_2 + A_3 = 0.09(10.5-5.25) + 0.015H + \left[1(10.5-5.75) + (H-1) \Delta t_p \right]$$

$$15.25 = 0.472 + 0.015H + 4.75 + (H-1) \Delta t_p = 5.22 + 0.015H + (H-1) \Delta t_p$$

But, by definition, $H = R_R - 33.25$ so that H is the lesser of 7.5 and $1.25 + 2.5 \Delta t_p + 1 = 2.25 + 2.5 \Delta t_p$

In this case the latter governs, and we determine

$$\Delta t_p = 1.75 \quad H = 6.65$$

- (c) The corner radii are then determined in accordance with C.1.3 and C.1.4.
- (d) Figure A.8.4.3-1 gives the final configuration, where the taper beyond the compensation is established in accordance with 6.4.2.

A.8.5 Check of Primary-plus-Secondary Stresses

A.8.5.1 Internal Pressure

- (a) At the inner surface A.3.3 gives the three principal stresses as

$$\sigma_r = -P = -2500 \text{ psi}$$

$$\sigma_h = P \left[\frac{y^2 + 1}{y^2 - 1} \right] = 2500 \left[\frac{y^2 + 1}{y^2 - 1} \right]$$

$$\sigma_a = \frac{P}{y^2 - 1} = \frac{2500}{y^2 - 1}$$

FIG. A. 8.4.3-1

(1) For the shell, $Y = \frac{R_1 + t}{R_1} = \frac{30.25 + 4.00}{30.25} = 1.132$. Therefore

$$Y^2 = 1.28, \quad Y^2 + 1 = 2.28, \quad Y^2 - 1 = 0.28, \quad \frac{Y^2 + 1}{Y^2 - 1} = 8.15,$$

$$\text{and } \sigma_r = -2500 \text{ psi}, \quad \sigma_h = 20,400 \text{ psi}, \quad \sigma_a = 8920 \text{ psi}.$$

(2) For the pipe, $Y = \frac{7.50}{5.00} = 1.5$. Therefore $Y^2 = 2.25$,

$$Y^2 + 1 = 3.25, \quad Y^2 - 1 = 1.25, \quad \frac{Y^2 + 1}{Y^2 - 1} = 2.6, \text{ and}$$

$$\sigma_r = -2500 \text{ psi}, \quad \sigma_h = 6800 \text{ psi}, \quad \sigma_a = 2100 \text{ psi}.$$

(b) At the outer surface, see A.3.3

$$\sigma_r = 0, \quad \sigma_h = \frac{2P}{Y^2 - 1}, \quad \sigma_a = \frac{P}{Y^2 - 1}.$$

(1) For the shell

$$\sigma_r = 0, \quad \sigma_h = 17,900 \text{ psi}, \quad \sigma_a = 8900 \text{ psi}$$

(2) For the pipe

$$\sigma_r = 0, \quad \sigma_h = 4000 \text{ psi}, \quad \sigma_a = 2000 \text{ psi}$$

(c) Note that the stresses at the end of the tapers must be evaluated as required by 6.4.2.

A.8.5.2 Pipe Reaction Effects in Shell. In accordance with 6.6.2, the method of A.5 will be used.

(a) The basic parameters β and γ (see A.5.5.2.1) are:

$$\beta = \frac{0.875r_o}{a} = \frac{0.875(7.50)}{32.25} = 0.203$$

$$\gamma = \frac{a}{t} = \frac{32.25}{4.00} = 8.06$$

- (b) For load P ($= 150,000$ lb, see A.8.2.7) with $t = 4$, $a = 32.25$.
From Figures A.5-4, A.5-5 and Reference A.5.7.4

$$\frac{M_\phi}{P} = 0.13; \quad \frac{N_\phi}{P/a} = -1.3; \quad \frac{M_x}{P} = -0.08; \quad \frac{N_x}{P/a} = -2.0$$

(Note: These values were not accurately determined)

$$\text{Then } \frac{6M_\phi}{t^2} = -7310; \quad \frac{N_\phi}{t} = -1510; \quad \frac{6M_x}{t^2} = -4500; \quad \frac{N_x}{t} = 2320.$$

Applying A.5-1

	<u>Outer Surface</u>	<u>Inner Surface</u>
Axial Stress, psi	-6800	2200
Hoop Stress, psi	-8800	5800

- (c) For circumferential moment M ($= 2 \times 10^6$ in lb, see A.8.2.7)
From Figures A.5-10 through A.5-13

$$\frac{M_\phi}{M/a} = -0.10; \quad \frac{N_\phi}{M/a^2} = -0.20; \quad \frac{M_x}{M/a} = -0.058;$$

$$\frac{N_x}{M/a^2} = -0.5.$$

$$\text{Then } \frac{6M_\phi}{t^2} = -11300; \quad \frac{N_\phi}{t} = -500; \quad \frac{6M_x}{t^2} = -6700; \quad \frac{N_x}{t} = -1200.$$

Applying A.5-1

	<u>Outer Surface</u>	<u>Inner Surface</u>
Axial Stress, psi	-7900	5500
Hoop Stress, psi	-11800	11000

- (d) Total stress due to pipe reactions

	<u>Outer Surface</u>	<u>Inner Surface</u>
Axial Stress, psi	-15000	7700
Hoop Stress, psi	-21000	17000

A.8.5.3 Pipe Reaction Effects in Nozzle

For the axial stress

$$\sigma_x = \pm \frac{M_c}{I} + \frac{P}{A} = \pm \frac{2(10)^6(4)_c}{\pi(7.5)^4 - (5.25)^4} - \frac{150(10)^3}{\pi(7.5)^2 - (5.25)^2}$$

$$= \pm 1070_c - 1660$$

$$= \pm 1070(7.5) - 1660 = -8700 \text{ or } 6300 \text{ at outer diameter}$$

$$= \pm 1070(5.25) - 1660 = -7300 \text{ or } 3700 \text{ at inner diameter}$$

A.8.5.4 In evaluating the fatigue condition, it will be necessary to distinguish the source of the stress. At the outer surface:

	<u>Axial Stress</u>	<u>Hoop Stress</u>
From "N"	- 2320 - 1200 = - 3500;	- 1510 - 500 = - 2010
From "M"	- 4500 - 6700 = -11200;	- 7310 - 11300 = - 18610

A.8.5.5 Primary-plus-Secondary Stress Evaluation

(a) At the inner surface

Loading	Component					
	Shell			Pipe		
	σ_r	σ_a	σ_h	σ_r	σ_a	σ_h
Pressure	-2500	8920	20400	-2500	2000	4000
Pipe Reactions	0	7700	17000	0	-7300 or 3700	0
Total	-2500	17000	37000	-2500	-5300 or 5700	0
Stress Intensity		40000			8200	

(b) At the outer surface

Loading	Component					
	Shell			Pipe		
	σ_r	σ_a	σ_h	σ_r	σ_a	σ_h
Pressure	0	8900	17900	0	2000	4000
Pipe Reactions	0	-15000	-21000	0	-8700 or 6300	0
Total	0	-6000	-3000	0	-6700 or 8300	4000
Stress Intensity		6000			10700	

(c) Conclusion. Maximum stress intensity is 40,000 psi which is equal to the allowable value $S_p = 40,000$ permitted by Table 5-1. Therefore, the design satisfies this stress limit.

A.8.6 Configuration Properties for Thermal Stress Analysis

A.8.6.1 The various coefficients for the equations will be evaluated, since they are needed for the analysis of all conditions.

A.8.6.2 Equations of A.6.2.2a

$$\frac{2R_1 + t}{0.605Et^2} = \frac{2(5.25) + 2.25}{0.605 (2.25)^2 E} = \frac{4.16}{E}$$

$$0.550 \sqrt{(2R_1 + t)t} = 0.550 \sqrt{(12.75)(2.25)} = 2.94$$

$$\frac{3.64}{\sqrt{(2R_1 + t)t}} = \frac{3.64}{5.35} = 0.680$$

So that the equations become

$$E \delta_C = 4.16 M_C - 12.2 V_C \quad (1)$$

$$E \theta_C = 2.83 M_C - 4.16 V_C \quad (2)$$

A.8.6.3 Equations of A.6.2.2b. Figure A.8.6.3-1 shows the cross-section of the ring in the plane transverse to the shell axis

FIGURE A.8.6.3-1

The following dimensions and properties of this section may be calculated by well-known methods from data already given.

$$\begin{aligned}
 A &= 15.5 \text{ in.}^2 & R_s &= 8.45 \text{ in.} \\
 I &= 32 \text{ in.}^4 & R_R &= 6.83 \text{ in.} \\
 L_C &= 2.68 \text{ in.} & R_1 &= 5.25 \text{ in.} \\
 L_s &= 1.18 \text{ in.} & R_V &= 30.25 \text{ in.} \\
 T &= 4.00 \text{ in.} & r_1 &= 1.50 \text{ in.} \\
 t &= 2.25 \text{ in.} & r_o &= 2.00 \text{ in.} \\
 \alpha &= 74.8^\circ
 \end{aligned}$$

Therefore the equations of A.6.2.2b become:

$$E \delta_R = 2.809 V_C - 3.723 V_s \quad (3)$$

$$E \theta_R = -1.361 M_C - 1.803 M_s - 3.65 V_C - 2.13 V_s \quad (4)$$

A.8.6.4 Equations of A.6.2.2c. From dimensions already known,

$$\lambda = \left[\frac{2.73 (32.25)^2}{(4.0)^2} \right]^{\frac{1}{4}} = (177)^{\frac{1}{4}} = 3.64$$

And the equations become

$$E \delta_s = 4.04 V_s - 1.74 M_s \quad (5)$$

$$E \theta_s = -1.74 V_s + 1.50 M_s \quad (6)$$

A.8.6.5 Equations of A.6.2.3. These equations may be written as

$$E \delta_C - E \delta_R + L_C E \theta_R = E \delta_{RT} - E \delta_{CT} - L_C E \theta_{RT}$$

$$E \delta_s - E \delta_R - L_s E \theta_R = E \delta_{RT} - E \delta_{sT} + L_s E \theta_{RT}$$

$$E \theta_C - E \theta_R = E \theta_{RT} - E \theta_{CT}$$

$$E \theta_s - E \theta_R = E \theta_{RT} - E \theta_{sT}$$

Substituting from Equations (1) thru (6) and collecting terms:

$$\begin{aligned}
 0.516 M_C - 4.832 M_s - 24.84 V_C - 1.980 V_s &= E(\delta_{RT} - \delta_{CT} - 2.68 \theta_{RT}) \\
 1.606 M_C + 0.386 M_s + 1.493 V_C + 10.271 V_s &= E(\delta_{RT} - \delta_{sT} + 1.18 \theta_{RT}) \\
 4.190 M_C + 1.803 M_s - 0.517 V_C + 2.128 V_s &= E(\theta_{RT} - \theta_{CT}) \\
 1.361 M_C + 3.307 M_s + 3.646 V_C + 0.386 V_s &= E(\theta_{RT} - \theta_{sT})
 \end{aligned} \quad (7)$$

Once the "free" thermal displacements are determined, these equations may be solved for M_C , M_s , V_C , and V_s , and the thermal stress obtained. This will be illustrated in A.8.9.

A.8.7 Thermal Analysis

A.8.7.1 As stated in 3.3.7, a detailed thermal analysis of each significant operating condition is required. For purposes of initial design, approximation can be obtained by considering Figures A.3-5 and A.3-6, and applying the method of superposition.

A.8.7.2 It is necessary to consider the thermal distribution at various times in order to be sure that the worst condition is investigated.

A.8.7.3 For purposes of this example problem, the temperature distribution, shown in Figure A.8.7-1, has been determined in an approximate manner.

FIGURE A.8.7-1

A.8.8 "Free" Displacements and Stresses

A.8.8.1 For the pipe and temperature distribution of A.8.7.3

$$\begin{aligned}\delta_{CT} &= (R_1 + \frac{t}{2}) \alpha \Delta T_M = 6.375 \alpha (555 - 70) = 3100 \alpha \text{ on startup.} \\ &= 6.375 \alpha (85 - 70) = 95.6 \alpha \text{ on shutdown.}\end{aligned}$$

$$\theta_{CT} = 0$$

On the outside, from A.3.5

$$\sigma_h = \sigma_a = \frac{E\alpha \Delta T_f}{1-\nu} \eta_o$$

where $E\alpha = 186$ from Figure 4.1-1

$\Delta T_f = 500^\circ\text{F}$, from A.8.2.6

$\nu = 0.3$, from Table 4-1

$\eta_o = 0.013$ for startup, from Fig. A.3-6
 $= -0.013$ for shutdown

$$\frac{E\alpha \Delta T_f}{1-\nu} = \frac{186(500)}{0.7} = 133,000$$

Then

$$\sigma_h = \sigma_a = 133(10)^3 (13)(10)^{-3} = 1700 \text{ psi for startup} \\ = -1700 \text{ psi for shutdown}$$

On the inside, from A.3.5

$$\sigma_h = \sigma_a = -\frac{E\alpha \Delta T_f}{1-\nu} \eta_i = -133(10)^3 \eta_i$$

From Fig. A.3-5 $\eta_i = -0.030$ for startup
 $= 0.030$ for shutdown

Then

$$\sigma_h = \sigma_a = 133(10)^3 (-30)(10)^{-3} = -4000 \text{ psi for startup} \\ = 4000 \text{ psi for shutdown}$$

A.8.8.2 For the ring and the temperature distributions of A.8.7.3, the result of the temperature distribution in the ring is that it will undergo a change in mean radius (due to the average temperature change of the ring) upon which is superimposed a rotation without distortion of the cross-section which results from the temperature distribution throughout the ring. The "free" stresses in the ring result from the fact that each circumferential ring element is not at its desired diameter. Note that, because of the assumptions inherent in ring analysis, only hoop stresses can exist.

From A.8.7.3, the average temperature and temperature gradient (positive increasing in the direction toward the vessel centerline) are:

<u>Condition</u>	<u>Average Temperature (°F)</u>	<u>Temperature Gradient (°F/in)</u>
Startup	552	- 1.39
Shutdown	88	0

The displacement and rotation are then

$$\begin{aligned}\delta_{RT} &= R_R \alpha \Delta T = 6.90 \alpha (552-70) = 3320 \alpha \text{ on startup} \\ &= 6.90 \alpha (88-70) = 124 \alpha \text{ on shutdown}\end{aligned}$$

$$\begin{aligned}\theta_{RT} &= R_R \alpha \left(\frac{\Delta T}{\Delta L} \right) = -6.90 \alpha (-1.39) = +9.6 \alpha \text{ on startup} \\ &= 0 \text{ on shutdown}\end{aligned}$$

The "free" thermal stress in the ring is a hoop stress caused by the fact that the circumference of a given ring element differs from its unrestrained value. The stress is tensile if the temperature at the point considered is less than the average temperature in the plane perpendicular to the ring centerline in which the point lies. The stress distribution is calculated from

$$\sigma = E \alpha \Delta T = 186 \Delta T \text{ and is illustrated below.}$$

Startup

Shutdown

A.8.8.3 For the shell, with the temperature of A.8.7.3, the radial expansion is

$$\Delta_S = (R_V + \frac{r}{2}) \alpha \Delta T = 32.25 \alpha (530-70) = 14800 \alpha \text{ on startup} \\ = 32.25 \alpha (110-70) = 1290 \alpha \text{ on shutdown}$$

Since

$$\delta_{ST} = \Delta_S \cos \alpha_2 = 0.262 \Delta_S = 3880 \alpha \text{ on startup} \\ = 340 \alpha \text{ on shutdown}$$

$\theta_{ST} = 0$ (Note: There will be a very slight rotation due to the small meridional gradient which has been neglected)

On the outside

$$\sigma_h = \sigma_a = \frac{E \alpha \Delta T_f}{1 - \nu} \eta_o = 133(10)^3 \eta_o$$

From Fig.A.3-6 $\eta_o = 0.043$ on startup
 $= -0.043$ on shutdown

and

$$\sigma_h = \sigma_a = 5700 \text{ psi on startup} \\ = -5700 \text{ psi on shutdown}$$

On the inside

$$\sigma_h = \sigma_a = 130(10)^3 \eta_i$$

From Fig.A.3-5 $\eta_i = -0.080$ on startup
 $= 0.080$ on shutdown

and

$$\sigma_h = \sigma_a = -11,000 \text{ psi on startup} \\ = 11,000 \text{ psi on shutdown}$$

A.8.9 Determination of Discontinuity Reactions. The quantities M_C , M_S , V_C , and V_S may now be determined from the equations of A.8.6.5. The known quantities are:

<u>Item</u>	<u>Reference</u>	<u>Startup</u>	<u>Shutdown</u>
δ_{CT}	A.8.8.1	3092 α	95.6 α
δ_{RT}	A.8.8.2	3300 α	122.9 α
δ_{ST}	A.8.8.3	3887 α	338.0 α
θ_{RT}	A.8.8.2	9.46 α	0
θ_{CT}	A.8.8.1	0	0
θ_{ST}	A.8.8.3	0	0

Using $E\alpha = 186$, from Figure 4.1-1, the solution is:

Item	Startup	Shutdown
M_C (in lb/in)	6400	2700
V_C (lb/in)	- 230	400
M_S (in-lb/in)	- 520	- 1000
V_S (lb/in)	-11400	- 4300

A.8.10 Determination of Discontinuity Thermal Stresses. Following A.6.2.4:

A.8.10.1 For the cylinder (pipe), outside

$$\sigma_a = - \frac{6M_C}{t^2} = - \frac{6M_C}{(2.25)^2} = - 1.185 M_C = - 7600 \text{ psi startup}$$

$$= - 1.185(2000) = - 3200 \text{ psi shutdown}$$

$$\sigma_h = \frac{3.30}{t^2} \left[M_C - 0.550 \sqrt{(2R_1 + t)t} V_C \right] + 0.3 \sigma_a$$

$$= \frac{3.30}{(2.25)^2} \left[M_C - 0.550 \sqrt{28.7} V_C \right] + 0.3 \sigma_a$$

$$= 0.652 M_C - 1.92 V_C + 0.3 \sigma_a$$

$$= 0.652(6400) - 1.92(-230) + 0.3(-7600)$$

$$= + 2340 \text{ psi startup}$$

$$= 30 \text{ psi shutdown}$$

A.8.10.2 For the ring. The hoop stress at any point is

$$\frac{\sigma_h}{X} = + \frac{1}{32} \left[(M_C + 2.68V_C)(5.25 + \frac{2.25}{2}) + (M_S + 1.18V_S)8.45 \right] - \frac{0.3\sigma_a}{X}$$

$$= 0.199 M_C + 0.264M_S + 0.534V_C + 0.311V_S - \frac{0.3\sigma_a}{X}$$

$$\sigma_h = 2500X + 0.3 \sigma_a \text{ startup}$$

$$= + 850X + 0.3 \sigma_a \text{ shutdown}$$

where the σ_a at the ends of the ring is assumed to be equal to the σ_a in the directly adjoining member.

Then the hoop and axial stresses are:

a. Outer surface, ring - pipe junction

$$x = -2.68 \quad \sigma_a = \begin{array}{l} -7600 \text{ psi startup) (see A.8.10.1)} \\ -3200 \text{ psi shutdown)} \end{array}$$

and

$$\sigma_h = (-2.68) [2500] + 0.3(-7600) = \begin{array}{l} -9100 \text{ psi startup} \\ -3200 \text{ psi shutdown} \end{array}$$

b. Inner surface, ring - pipe junction

$$x = -2.68 \quad \sigma_a = \begin{array}{l} +7600 \text{ psi startup) (Negative of A.8.10.1)} \\ -3200 \text{ psi shutdown)} \end{array}$$

and

$$\sigma_h = (-2.68) [2500] + 0.3(7600) = \begin{array}{l} -4500 \text{ psi startup) } \\ -1300 \text{ psi shutdown) } \end{array}$$

c. Outer surface, ring - shell junction

$$x = L_A - L_C = 1.93 - 2.68 = -0.75$$

$$\sigma_a = \begin{array}{l} -2325 \text{ psi startup) (see A.8.10.3)} \\ -713 \text{ psi shutdown)} \end{array}$$

and

$$\sigma_h = (-0.75) [2500] + 0.3(-2325) = \begin{array}{l} -2600 \text{ psi startup} \\ -850 \text{ psi shutdown} \end{array}$$

d. Inner surface, ring - shell junction

$$x = L_a - L_C = 3.11 \quad \sigma_a = \begin{array}{l} 2325 \text{ psi startup) (Negative of A.8.10.3)} \\ -713 \text{ psi shutdown)} \end{array}$$

and

$$\sigma_h = (3.11) [2500] + 0.3(2325) = \begin{array}{l} 8500 \text{ psi startup} \\ 2900 \text{ psi shutdown} \end{array}$$

A.8.10.3 For the shell, at the outside

$$\sigma_a = + \frac{6M_S}{t^2} = .375 M_S = \begin{array}{l} -200 \text{ psi startup} \\ -400 \text{ psi shutdown} \end{array}$$

$$\begin{aligned}
 \tau_h &= \frac{2 \lambda \cos \alpha}{T(R_V + \frac{T}{2})} \left[(R_V + \frac{T}{2})(\cos \alpha) V_S - \lambda M_S \right] + 0.3 \sigma_a \\
 &= \frac{2(3.64)(0.262)}{4(32.25)} \left[32.25(0.262) V_S - 3.64 M_S \right] + 0.3 \sigma_a \\
 &= 0.125 V_S - 0.0538 M_S + 0.3 \sigma_a \\
 &= 0.125(-11400) - 0.0538(-520) + 0.3(-200) = -1500 \text{ psi startup} \\
 &= 0.125(-4302) - 0.0538(-1000) + 0.3(-400) = -700 \text{ psi shutdown}
 \end{aligned}$$

A.8.11 Stress Concentration Effects

A.8.11.1 The use of stress concentration factors is treated in 3.1.3, 5.2.1, and A.7. It must be noted that stress concentration factors must be applied to each individual stress component, and cannot be applied to the total stress at a point, or to the stress intensity.

The stress concentration factors to be used in the fatigue evaluation of this problem will now be determined.

A.8.11.2 The stress concentration factors for a nozzle subjected to internal pressure are specifically discussed in Appendix C. In that appendix, stress indices (see C.1.7) are given for various designs. The membrane stress intensity in the shell, upon which these values are based, is

$$S_M = \left[\frac{P}{2t} (2R + t) \right] = \frac{2500}{2(4.0)} [2(30.0) + 4.0] = 20,000 \text{ psi}$$

The stress indices of Figure C.3-6 apply to this nozzle with 33% reinforcement in the vessel wall. Therefore, the stress indices are:

Section	Surface	Index for the Stress Component*		
		σ_h	σ_a	σ_r
Transverse	Outside	1.26	1.26	0
"	Inside	-0.15	0.85	-0.15
Longitudinal	Outside	0.49	0.36	0
"	Inside	2.60	-0.15	-0.15

*The stress is the tabulated value multiplied by $S_M = 20000$ psi in this case.

A.8.11.3 The stress concentration effects for the stresses due to pipe reactions will be considered next (see A.5.4.3). Examination of the results in A.8.5.2 shows that the shell stress in each direction is the sum of a membrane component (due to N) and a bending component (due to M). There is a stress concentration effect on both of the components. The stress concentration factor for the shell depends upon the ratio of the outside fillet radius to the shell thickness ($2.0/4.0 = 0.5$ in this case). Then, from Fig. A.7-1

Bending: $K = 1.5$

Membrane: $K = 1.8$

Stress concentration effects in the pipe depend upon the ratio of the outside fillet radius to the outside diameter of the pipe ($2.0/15.0 = 0.133$) and upon the type of loading. The stress concentration factors to be applied to the axial stress in the pipe are then, from Figure A.7-1

For P: $K = 2.9$;

For M: $K = 2.45$

A.8.11.4 In this example, as well as in most cases, no stress concentration factors need be applied to the "free" thermal stresses.

A.8.11.5 The stress concentration factors for the "discontinuity" thermal stresses will now be considered. These stresses depend upon the effects of both the discontinuity shears and the discontinuity moments. However, only that portion of the stress which results from the discontinuity moment is affected by the stress concentration factor and that only with respect to the axial stresses including the Poisson's ratio effect on the hoop stresses.

The factor for this moment again depends upon the ratio of the outside fillet radius to the wall thickness. Applying Figure A.7-1

For the pipe ($r/t = \frac{2.0}{2.25} = 0.888$); $K = 1.32$.

For the shell ($r/t = \frac{2.0}{4.0} = 0.5$); $K = 1.50$.

A.8.12 Total Stress at Beginning of Startup. All stresses will be assumed to be zero. This implies that the vessel is unpressurized isothermal at 70 °F and the effect of pipe reactions may be evaluated by considering them to be zero when the plant is shut down and proportional to the ratio of fluid temperature to operating temperature at full power for other conditions.

A.8.13 Total Stress at End of Startup. In calculating the pressure stress at the end of startup, it will be assumed that the full power pressure is equal to the design pressure, and that the pressure is developed during startup.

The stresses at the various locations which result from the different conditions are now tabulated in units of 1000 psi. However, in accordance with A.8.2.8, only the points at the junction of the ring and the cylindrical shell will be tabulated. This is done in A.8.18.

A.8.14 Total Stress at Beginning of Shutdown. It is assumed that the plant has been operating at full power for such a length of time that the nozzle region is essentially isothermal. Therefore, the stress results only from internal pressure and pipe reactions.

A.8.15 Total Stress at End of Shutdown. At this time it is assumed that the pressure and pipe reactions have dropped to zero, although the thermal stresses will not yet be negligible.

A.8.16 Total Stress After a Lengthy Shutdown. The nozzle region is isothermal, all stresses are zero, and the cycle is complete.

A.8.17 Precautionary Note: It must be noted that the detailed procedure followed here is only intended as an illustration. There is no certainty that the worst stress occurs at the particular location considered or that the five points in time considered are sufficient to completely describe the cycle.

Also, note that the radial stress at this point is zero.

A.8.18.1 STRESS SUMMARY - Outside of Ring Adjacent to Shell (Transverse Section)(Stress in 1000 psi)

Time (See Par.)	Condition and Source	Longitudinal Stress			Hoop Stress		
		Basic	K	Actual	Basic	K	Actual
Initial (A.8.12)	Isothermal, zero pressure	0	-	0	0	-	0
End of Startup (A.8.13)	Internal pressure, 2500 psi	20.0	1.26	25.2	20.0	1.26	25.2
	Pipe Reactions:						
	From "N"	- 3.5	1.80	- 6.3	- 2.0	1.80	- 3.6
	From "M"	-11.2	1.50	-16.8	-18.6	1.50	-27.9
	"Free" Thermal	0	-	0	7.6	1.0	7.6
	"Discontinuity" thermal:						
	From "M _S "	- 2.3	1.50	- 3.5	0	-	0
	From other effects	0	-	0	- 1.9	1.00	- 1.9
Begin Shutdown (A.8.14)	From " $\sqrt{\sigma_a}$ "	0	-	0	- 0.7	1.00	- 0.7
	Total			- 1.4			- 1.3
	Internal pressure, 2500 psi	20.0	1.26	25.2	20.0	1.26	25.2
	Pipe Reactions:						
End Shutdown (A.8.15)	From "N"	- 3.5	1.80	- 6.3	- 2.0	1.80	- 3.6
	From "M"	-11.2	1.50	-16.8	18.6	1.50	-27.9
	Thermal, assumed isothermal	0	-	0	0	-	0
	Total			+ 2.1			- 6.5
	Zero Pressure & Pipe Reactions	0	-	0	-	-	0
Final (A.8.16)	"Free" Thermal	0	-	0	- 7.8	1.0	7.8
	"Discontinuity" Thermal:						
	From "M _S "	- 0.7	1.50	- 1.0	0	-	0
	From other effects	0	-	0	- 0.6	1.00	- 0.6
	From " $\sqrt{\sigma_a}$ "	0	-	0	- 0.2	1.00	- 0.2
Final (A.8.16)	Total			- 1.0			- 8.6
	Isothermal, zero pressure	0	-	0	0	-	0

A.8.18.2 STRESS SUMMARY - Inside of Ring Adjacent to Shell (Longitudinal Section)(Stress in 1000 psi)

Time (see Par.)	Conditions and Source	Longitudinal Stress			Hoop Stress		
		Basic	K	Actual	Basic	K	Actual
Initial (A.8.12)	Isothermal, zero pressure	0	-	0	0	-	0
End of Startup (A.8.13) $\sigma_r = -2.5$	Internal pressure, 2500 psi	8.9	1.00	8.9	20.0	2.60	52.0
	Pipe Reactions:						
	From "N"	- 3.5	1.00	- 3.5	- 2.0	1.00	- 2.0
	From "M"	11.2	1.00	11.2	18.6	1.00	18.6
	"Free" Thermal	0	-	0	- 2.7	1.00	- 2.7
	"Discontinuity" Thermal:						
	From "M _S "	2.3	1.00	2.3	0	-	0
	From other effects	0	-	0	7.8	1.00	7.8
	From " $\sqrt{\sigma_a}$ "	0	-	0	0.7	1.00	0.7
	Total			18.9			73.8
Begin Shutdown (A.8.14) $\sigma_r = -2.5$	Internal pressure, 2500 psi	8.9	1.00	8.9	20.0	2.60	52.0
	Pipe Reactions:						
	From "N"	- 3.5	1.00	- 3.5	- 2.0	1.00	- 2.0
	From "M"	11.2	1.00	11.2	18.6	1.00	18.6
	Assumed isothermal	0	-	0	0	-	0
	Total			16.6			68.6
End Shutdown (A.8.15) $\sigma_r = 0$	Zero pressure & pipe reactions	0	-	0	0	-	0
	"Free" Thermal	0	-	0	- 3.3	1.00	- 3.0
	"Discontinuity" Thermal:						
	From "M _S "	0.7	1.00	0.7	0	-	0
	From other effects	0	-	0	2.7	1.00	2.7
	From " $\sqrt{\sigma_a}$ "	0	-	0	0.2	1.00	0.2
	Total			0.7			- 0.1
Final (A.8.16)	Isothermal, zero pressure	0	-	0	0	-	0

A.9 Evaluation of the Fatigue Effects

A.9.1 In order to illustrate the application of paragraph 5.2, the method will be applied to the stresses determined in paragraph A.8 at the inside of the ring adjacent to the nozzle. Note that an understanding of the contents of paragraph A.8 is not required for this purpose.

A.9.2 From Table A.8.18.2, and in accordance with 5.2.1, the principal stresses are:

Time	Principal Stresses (1000 psi)		
	σ_1	σ_2	σ_3
I Initial (see A.8.12)	0	0	0
II End of Startup (see A.8.13)	18.9	73.8	- 2.5
III Begin Shutdown (see A.8.14)	16.6	68.6	- 2.5
IV End Shutdown (see A.8.15)	0.7	- 0.1	0
V Final (see A.8.16)	0	0	0

The variation of these stresses, which include all structural discontinuity and stress concentration effects with time is sketched on Figure A.9-1.

A.9.3 The stress differences, see 5.2.2, are plotted on Figure A.9-2.

A.9.4 The alternating stress differences, see 5.2.3, are then found to be:

$$S_{alt\ 12} = \frac{55,500}{2} = 27,800; \quad S_{alt\ 23} = \frac{76,400}{2} = 38,200;$$

$$S_{alt\ 31} = \frac{21,400}{2} = 10,700.$$

A.9.5 The mean stress differences, see 5.2.4, may now be determined as follows:

Component	S'_{mean}	Criterion (see 5.2.4)	S_{mean}
12	27,800	$S_{alt} + S_{mean} > S_b$	20,200
23	38,200		9,800
31	10,700	$S_{alt} + S_{mean} < S_b$	10,700

A.9.6 Each component is then plotted on the fatigue diagram and their "y" axis intercepts determined in accordance with 5.2, as shown in Figure A.9-3.

FIG. A.9-1
PRINCIPAL STRESS PLOT

FIG. A.9-2
STRESS DIFFERENCE PLOT

FIG. A.9-3
FATIGUE DIAGRAM FOR SAMPLE PROBLEM

A.9.7 If the requirements of 5.2.5 are being investigated, it is necessary that each of these three points fall below the line $S_{aIII} - S_s$. Figure A.9-3 illustrates that this criterion is met for this problem.

A.9.8 If the requirements of 5.2.8 are being investigated, the largest intercept on the "Y" axis ($S_a = 44,000$ psi) must be treated in accordance with 5.2.8.3. Then, from Figure 5.2-3, the allowable number of cycles, N , is 6,600.

A.9.9 If the actual number of these cycles is $n = 400$ the usage factor, see 5.2.8.4 is

$$U = \frac{n}{N} = \frac{400}{6600} = 0.0605$$

APPENDIX B

BASIS FOR ESTABLISHING ALLOWABLE STRESSES

B.1 Basic Assumptions Concerning Material Strength. As stated in the design basis, it is intended that all allowable stress figures be based upon minimum properties at maximum service metal temperature. Unfortunately, however, adequate information concerning the strength properties of a specific material at elevated temperatures is not always available. Therefore, where specific test data were not available, the following assumptions have been made:

B.1.1 Ferritic Materials

B.1.1.1 Following common practice for normal pressure vessel materials, the specification minimum ultimate strength can be used at temperatures below 650°F, which should cover our design requirements. In the case of material conforming to MIL-S-16113, Grade HT, a conservative ultimate strength was assumed. For quenched and tempered materials, actual values are considered.

B.1.1.2 The yield strengths at elevated temperatures were taken to be lower than specification minimum properties (room temperature) by not greater than the percentage shown in Fig. B.1-1. The "carbon steel" curve in that figure is applicable to materials containing less than 0.4% Mo.

B.1.2 Austenitic Materials. Reasonably accurate data are available concerning the variation of ultimate strength and yield strength of 304 and 347 stainless steel as a function of temperature. The characteristic curves for 304 L are assumed similar to those for 304, the difference being that the specification minimum strengths are lower. The yield strength curve for 347 has been modified somewhat because of the fact that normal minimum values are well in excess of specification minimum values.

B.1.3 Fatigue Strength of Clad Plate. Available information indicates that the fatigue strength of a clad plate is, to a large degree, determined by the strength of the surface layer (at least where the surface layer is the weaker of the two materials). However, for the materials used herein, it has been arbitrarily assumed that the fatigue strength of the clad plate is the same as that of the base plate.

B.1.4 Construction of Fatigue Design Curves

B.1.4.1 Construction of the design fatigue curve for common structural materials. Figure 5.2-3 presents the design fatigue curve for use with materials whose room temperature tensile strength is between 65,000 and 100,000 psi, including the following materials:

ASTM: A-204, A-212, A-302

AISI: 304, 304 L, 347, 348, 410 (less than 100,000 psi tensile strength)

Others: MIL-S-16113HT, Inconel, SAE 4340 (less than 100,000 psi tensile strength)

Fig. B.1-1 ESTIMATED LOSS IN YIELD STRENGTH WITH TEMPERATURE

This curve was constructed in the following manner:

- (a) Available strain fatigue data for this general class of material were plotted in the form of total strain (elastic plus plastic) range versus cycles-to-failure. Machined specimens without notches that were tested at temperatures less than 600°F were considered. The mean curve for each material was drawn.
- (b) A lower limit of the mean curves was drawn and then converted to a stress amplitude versus cycles-to-failure curve by multiplying the strain range by $E/2$, where E was taken as 26×10^6 psi.
- (c) The design fatigue curve was then constructed by applying a factor of safety of either 2.0 on stress amplitude or a factor of 20 on cycles whichever was more conservative at each point. The factor of 20 on life is the product of the following sub-factors:

Scatter of data (minimum to mean)	2.0
Size Effect	2.5
Surface finish, atmosphere, etc.	4.0
- (d) The design fatigue curve stress amplitude for less than 100 cycles was taken as the value at 100 cycles.

B.1.4.2 The endurance limit of rough-turned specimens as a function of the ultimate tensile strength of a material is given by Figure 5.2-4. A safety factor of 1.5 may be used on these values to obtain the values of S_{ae} . This method was used in developing Table 5-1. This is considered to be essentially equivalent to a factor of safety of two on polished specimens. (see B.1.4.1.c).

B.1.4.3 Construction of the design fatigue curve for materials whose room temperature tensile strength is greater than 100,000 psi.- The design fatigue curve for these materials is obtained by adding two-thirds of the difference between the value of S_e (as obtained from Figure 5.2-4 for the material considered) and 37,000 psi to the stress values of Figure 5.2-3. This procedure has been followed in the construction of the following figures:

<u>Figure</u>	<u>Material</u>
5.2-5	AISI 410, SAE 4340, and Inconel X all heat treated to provide 140,000 - 160,000 psi ultimate tensile strength

B.1.4.4 Construction of the design fatigue curve for materials whose room temperature tensile strength is less than 65,000 psi. The design fatigue curve for these materials may be obtained by multiplying the stress values of Figure 5.2-3 by the ratio of the room temperature tensile strength for the material considered to the quantity 80,000 psi.

B.2 Limits of Allowable Stress

B.2.1 Membrane Stress. The average stress intensity, or membrane stress, S_m shall not exceed 62.5% of the yield strength in tension, or 33.3% of the ultimate strength in tension, whichever is less. However, where this rule results in membrane stress values less than those in common usage, as exemplified by ASME Code practice "S" values, the allowable membrane stress limit may be adjusted to be consistent with ASME Code practice, except that Code procedures for ferrous materials have been applied to non-ferrous materials. Such a procedure was followed in establishing the limit for A-212.

B.2.2 Primary-plus-secondary Stress. The allowable stress intensity, S_p , as defined in paragraph 5.1.3 shall not exceed 90% of the yield strength in tension, or 60% of the ultimate strength in tension, whichever is less.

B.2.3 Mean Stress. The maximum steady-state component or mean stress, S_s , as used in the fatigue diagram and as defined in paragraph 5.2.4, shall not exceed the tabulated value of S_u in Table 5-1 at the temperature considered.

B.2.4 Alternating Stress. The maximum alternating stress component as used in the fatigue diagram and as tabulated in Table 5-1 for use with paragraph 5.2.5 shall not exceed those values determined in the following manner from the applicable fatigue design curve:

B.2.4.1 Assign the number of classes of plant operation (see 3.4.4) to be used, based on the important groups of similar operations that occur. There may be any number of such groups.

B.2.4.2 Apply Miner's Hypothesis (see 5.2.8) as follows:

- (a) Assign a usage factor to each class such that the sum of these factors is no more than 0.8. For Table 5-1, the three classes, I, II, and III, were assigned the usage factors 0, 0.4, and 0.4, respectively.
- (b) Determine the required number of cycles on the design fatigue curve at which the value S_a should be determined by dividing the number of cycles assumed to occur for each class by the value of the usage factor assigned to that class. It should be noted that a usage factor of 0.0 implies that the S_a value is allowable for an infinite number of cycles. For this case, S_a shall be taken as equal to S_{ae} . For the conditions assumed in paragraph 3.3.4:

Class	Cycles for Class	Assigned Usage Factor	Cycles for Determining S_a
I	100,000	0.0	10^7
II	2,500	0.4	6,250
III	500	0.4	1,250

- (c) Determine the allowable S_a values for each class from the applicable design fatigue curve at the number of cycles determined above. For example, for A-302-B (Fig. 5.2-3) and the conditions assumed in paragraph 3.3.4.

Class	Cycles for Class	Applicable S_a (psi)
I	100,000	24,000
II	2,500	45,000
III	500	66,000

B.2.5 Limit of Elastic Behavior. The stress intensity S_p is defined as the limit of elastic behavior and is used in connection with the fatigue diagram to determine the reduction in mean stress produced by plastic flow. For some materials the yield stress, S_y , is higher than the endurance limit, S_e . In these cases $S_p = S_y$. For materials which strain harden by appreciable amounts, the endurance limit may be appreciably higher than the yield stress of the annealed material. Thus S_y is not the true limit of elastic behavior and the endurance limit is a more realistic value to take for S_p . For this purpose, the endurance limit of polished specimens without safety factor is taken as the best estimate for S_p , because the use of too low a value results in an unconservative estimate of the alleviating effects of the plastic flow.

B.3 Allowable Stress Curves and Figures. Curves of allowable stresses for mechanical loadings based upon the above rules are contained in Figures B.3-1 and B.3-2 for purposes of illustration. The specific figures used in Table 5-1 are taken from such curves, rounded off to two significant figures.

FIG B.3-1 ALLOWABLE STRESSES FOR A-302, GRADE B STEEL.

FIG B.3-2. ALLOWABLE STRESSES FOR SAE 4340 STEEL (100,000 PSI TEN. STR)

APPENDIX C

COMPENSATION FOR OPENINGS

C.1 Basic rules for the amount and placement of compensation are as contained in the following subparagraphs.

C.1.1 All compensation shall be integral with the vessel wall or with the nozzle wall (or with both), except as permitted in paragraph 3.5.1.2.

C.1.2 The amount and placement of compensation shall conform to the requirements of Section VIII of the Code, except that the required shell thickness shall be calculated by the equations in A.3.1 (1) and Table 5-1, rather than upon ASME Code rules.

C.1.3 The corner radius r_1 , as shown in Fig. C.1-1, shall not be less than one-third of the shell thickness nor greater than two-thirds of the shell thickness without obtaining verification of stress intensification factors used (that is, stress intensification factors specified herein, are based upon test results where radii were within the ranges specified herein, except for the special case shown in Fig. C.2-3.).

C.1.4 The outer corner radius, r_2 , at the intersection between the compensation and the shell as shown in Fig. C.1-1, shall be not less than the greater of:

- (a) The inside corner radius r_1 (except as permitted for the configuration shown in Fig. C.2-3).
- (b) One-half the shell thickness.
- (c) (For Fig. C.1-1e and C.2-6 only), the quantity $\frac{90 \times d_p}{500}$, where d_p = the outside diameter of the attached pipe.

C.1.5 The radius r_3 , between the reinforced section of the nozzle wall and the pipe wall, as shown in Fig. C.1-1, shall be not less than the greater of:

- (a) $d_p \alpha / 500$, where d_p is the outside diameter of the pipe and α is the angle in degrees between the pipe wall and the adjacent compensation, as illustrated.
- (b) (For variations of Fig. C.1-1a, including C.1-1b) twice the offset times $(\sin \alpha)^3$.

C.1.6 Precautionary Note. The following paragraphs contain stress data concerning specific compensation configurations. These stress data are based upon three-dimensional photoelastic tests at the University of Illinois under the Pressure Vessel Research Committee - BuShips test program, and the test

FIGURE C.1-1.

results have been adjusted to take care of minor variations in corner radii, etc., permitted herein. The models tested had the following dimensional ratios (approximate):

	<u>Cylinder</u>	<u>Sphere</u>
<u>Inside diameter shell</u> <u>thickness of shell</u>	13.7	24.0
<u>Inside diameter pipe wall</u> <u>thickness pipe wall</u>	12.7	11.5
<u>Inside diameter shell</u> <u>inside diameter pipe</u>	6.5	5.0

The stress data contained herein are considered adequate to cover the normal ratio of shell to attachment diameters (D/d) for primary coolant piping in reactor vessels (normally having ratios between, perhaps, 5.0 and 7.0), and for shell diameter to thickness (D/t) ratios as normally obtained in the use of A-212 or A-302 material with 2500 psi design pressure. However, it is emphasized that these stress indices may not be adequate for very small connections in a large vessel (high D/d ratios) or for a branch connection having a diameter approaching that of the shell (small D/d ratios). This is definitely true in the case of stress indices for an "uncompensated" opening, configuration C.2-6. Also, these stress indices may be too low for vessels having low D/t ratios, as would normally be obtained in a solid stainless steel vessel, which has a comparatively low allowable stress at the usual 600-650°F design temperature.

C.1.7 Definition of stress indices. The stress indices specified in the following paragraphs, and on the Figures mentioned therein, are expressed as numerical ratios of the stresses in question to the calculated membrane stress intensity in the shell or head (see paragraph A.3.1.2). Symbols used are as follows (and as illustrated in Figure C.1-2):

- σ_t = the stress component in the plane of the section under consideration and parallel to the boundary of the section.
- σ_n = the stress component normal to the plane of the section under consideration (ordinarily, the circumferential stress around the hole in the shell).
- σ_r = the stress component normal to the boundary of the section.
- S = the stress intensity (combined stress) at the point in question.

C.2 Spherical shells (hemispherical heads).

C.2.1 Stress indices for two forms of nozzle compensation, all on the outside of the vessel wall, complying in general with the above requirements are shown in Fig. C.2-1 and C.2-2. A third configuration, having a larger inside corner radius than permitted by the foregoing rules together with appropriate

FIGURE C.1-2

STRESS INDICES

	<u>INSIDE</u> <u>CORNER ①</u>	<u>OUTSIDE</u> <u>CORNER ②</u>
σ_n	1.75	1.80
σ_t	~ 0.15	1.75
σ_r	~ 0.15	0.0
S	1.90	1.80

**FIG. C2-1. STRESS INDICES FOR NOZZLE
CONFIGURATION IN SPHERICAL SHELL**

STRESS INDICES

	<u>INSIDE CORNER ①</u>	<u>OUTSIDE CORNER ②</u>
σ_n	1.75	1.70
σ_t	~ 0.15	1.80
σ_r	~ 0.15	0.0
S	1.90	1.80

**FIG. C.2-2. STRESS INDICES FOR NOZZLE
CONFIGURATION IN SPHERICAL SHELL**

STRESS INDICES

	<u>INSIDE</u> <u>CORNER</u> ①	<u>OUTSIDE</u> <u>CORNER</u> ②
σ_M	1.75	2.05
σ_t	~ -0.15	2.15
σ_r	~ -0.15	0.0
S	1.90	2.15

**FIG. C.2-3. STRESS INDICES FOR NOZZLE
CONFIGURATION IN SPHERICAL SHELL**

FIG. C.2-5. STRESS INDICES FOR NOZZLE
CONFIGURATION IN SPHERICAL SHELL

STRESS INDICES

	INSIDE ① CORNER	OUTSIDE ② CORNER
σ_n	2.25	2.75
$\sigma_t \sim -0.15$		1.90
$\sigma_r \sim -0.15$		0.0
S	2.40	2.75

DO NOT USE THESE STRESS INDICES FOR DIAM.
SHELL/DIAM PIPE RATIOS MATERIALLY LESS
THAN 50. (SEE PAR. C.1.6)

FIG.C.2-6 STRESS INDICES FOR NOZZLE
CONFIGURATION IN SPHERICAL SHELL

higher stress intensification factors, is shown in Fig. C.2-3. This configuration is acceptable as long as the specified factors are used. Where configurations other than these (but complying with paragraph C.1 above) are used, the stress indices shall be taken as follows:

	<u>Inside Corner</u>	<u>Outside Corner</u>
σ_n	1.85	2.00
σ_t	~ - 0.15	2.00
σ_r	~ - 0.15	0.0
S	2.00	2.00

C.2.2 When balanced compensation of the general form shown in Figures C.2-4 or C.2-5 is used, the shell need not be increased above that required by the allowable stresses contained in this "design basis", provided that (a) the distribution of compensating material, both inside and outside of the shell, falls within the range of 45-55 percent of the total; (b) the inside corner radius, r_1 , shall be not less than one-half the shell thickness (or of the nozzle thickness in the case of the configuration shown in Fig. C.2-5); and (c) other pertinent radii, r_2 , shall be not less than the shell thickness, as shown on Fig. C.2-4 and C.2-5. An acceptable method of fabrication for the configuration shown on Fig. C.2-5 is indicated thereon.

C.2.3 When the required compensating area is provided all in the vessel wall by increasing the thickness of the entire head, the thickness of the head shall be at least twice that required by the allowable membrane stress intensities contained in this "design basis". (Note: For some materials, such as A-302, this permits an appreciable reduction below ASME Code requirements under similar circumstances). Corner radii shall conform to the requirements of subparagraphs C.1.3 and C.1.4. Stress indices for such a configuration are shown in Fig. C.2-6. These stress indices are for an uncompensated opening and, whereas the indices appear high, it must be remembered that the membrane stresses are now one-half their usual values. Particular attention is invited to the precautionary note in paragraph C.1.6 concerning the use of these stress indices.

C.3 Cylindrical Shells

C.3.1 A recommended method for the compensation of openings in a cylindrical shell is by an increase in the shell thickness around the entire circumference of the vessel in the zone of the openings. When this method of compensation is used, the width of the reinforced band shall extend a distance not less than $1.5d$ from the center of the nozzle or opening, where d is the inside diameter of the nozzle or opening, as shown in Fig. C.3-1. Referring to paragraphs C.1.3 and C.1.4 concerning rules for corner radii, references to "shell thickness" contained therein shall mean the thickness of the reinforced section. The slope between the reinforced section and the "design basis" wall thickness shall comply with the requirements of paragraph 6.4.2.

FIG. C.3-1

C.3.2 Stress indices for two configurations having shells of uniform thickness and having all of the required compensation in the nozzle wall, are shown in Figures C.3-2 and C.3-3. Equally acceptable configurations for which the same stress indices may be used are shown in Figures C.3-4 and C.3-5. Although on the basis of tests on spherical models, these configurations are unquestionably preferable from a stress standpoint to those shown in Figures C.3-2 and C.3-3, no credit will be taken for improved stress indices until actual test data become available from cylindrical models.

C.3.3 When configurations other than those generally similar to Figures C.3-2 to C.3-5, inclusive, are used in shells having uniform wall thicknesses, but are otherwise provided with corner radii, etc., in accordance with the foregoing requirements, the stress indices shall be taken as follows:

	<u>Longitudinal Axis</u>		<u>Transverse Axis</u>	
	<u>Inside</u>	<u>Outside</u>	<u>Inside</u>	<u>Outside</u>
σ_n	2.75	1.00	0.85	1.50
σ_t	~ - 0.15	0.60	~ - 0.15	1.50
σ_r	~ - 0.15	0.0	~ - 0.15	0.0
S	2.90	1.00	1.00	1.50

C.3.4 When compensation is provided through use of a reinforcing band as mentioned in paragraph C.3.1, the stress indices may be taken from Figure C.3-6, noting that these indices are expressed in relation to a calculated stress intensity based on the total thickness of the reinforced section (band). These curves (Figure C.3-6b) permit credit to be taken for the fact that, from the standpoint of pressure stresses, a reinforcing band provides more effective compensation than material located in the nozzle wall. (Note: This may not be apparent from casual inspection of the curves. However, as a matter of example, placing 50% of the required compensation in the form of a reinforcing band would permit a reduction of the index for σ_n at the inside corner on the longitudinal section from 2.30 to approximately 1.85 as related to the stress intensity calculated for the basic shell thickness.)

C.4 Experimental Stress Data. Consideration will be given to the use of lower stress indices than those listed above, or to the use of non-radial connections, etc., where supporting experimental stress data, acceptable to the Naval Reactors Branch, are available. Any data which subsequently becomes available from the PVRC - Buships test program will ordinarily be acceptable provided the configuration proposed is closely similar to a configuration tested.

STRESS INDICES

	<u>LONGITUDINAL SECTION</u>		<u>TRANSVERSE SECTION</u>	
	<u>INSIDE CORNER</u> ①	<u>OUTSIDE CORNER</u> ②	<u>INSIDE CORNER</u> ③	<u>OUTSIDE CORNER</u> ④
σ_n	2.30	0.25	0.85	1.15
σ_t	~ -0.15	0.25	~ -0.15	1.15
σ_r	~ -0.15	0.0	~ -0.15	0.0
S	2.45	0.25	1.00	1.15

FIG C.3-2. STRESS INDICES FOR NOZZLE CONFIGURATION IN CYLINDRICAL SHELL OF UNIFORM THICKNESS.

USE DATA FOR FIG C.3-2 UNTIL TEST
OF MODEL SIMILAR TO THIS CONFIGURATION
IS COMPLETED.

FIG.C.3-4. STRESS INDICES FOR NOZZLE
CONFIGURATION IN CYLINDRICAL SHELL

USE DATA FOR FIG. C.3-2 UNTIL TEST OF MODEL SIMILAR TO FIG. C.3-4 IS COMPLETED.

FIG. C.3-5 STRESS INDICES FOR NOZZLE CONFIGURATION IN CYLINDRICAL SHELL.

FIG. C.3-6b. STRESS INDICES FOR NOZZLE CONFIGURATION IN CYLINDRICAL SHELL WITH INTEGRAL REINFORCING BAND.

APPENDIX D

THERMAL STRAIN RATCHET*

D.1 If sufficiently high cyclic thermal strains are applied to a structure subjected to a sustained mechanical loading, a cyclic growth will occur. In the case of a shell with a sustained tensile membrane stress due to pressure and a cyclic heat flux through the shell, there will be an increment of membrane strain produced during each cycle of application and removal of the heat flux.

D.2 The limit which must be imposed on the maximum thermal stress in a shell loaded by internal pressure in order to prevent cyclic growth is defined below, where:

y = maximum thermal stress computed on an elastic basis, divided by the yield strength, S_y .

x = maximum membrane stress due to pressure divided by the yield strength, S_y .

D.2.1 Case A. Linear variation of temperature through the wall.

$$y = \frac{1}{x} \quad \text{for } 0 < x < 0.5$$

$$y = 4(1-x) \quad \text{for } 0.5 < x < 1.0$$

D.2.2 Case B. Parabolic variation of temperature through the wall, corresponding to uniform heat generation in a thin-walled shell and heat removed entirely from one surface or equally from both surfaces.

$$y = 5.2(1-x) \quad \text{for } 0.625 < x < 1.0$$

and approximately,

x	y
0.3	4.65
0.4	3.55
0.5	2.70

D.3 Use of the yield strength in the above relations instead of the elastic limit actually allows a small amount of cyclic growth until strain hardening raises the elastic limit to S_y . With higher stresses growth will occur until the elastic limit is increased enough beyond S_y that no further growth occurs.

* See ASME Paper No. 58-A-129, "Thermal-Stress Ratchet Mechanism in Pressure Vessels", by D. R. Miller.

APPENDIX EBASIC BIBLIOGRAPHY ON THERMAL STRESS ANALYSIS

The following bibliography is intended to include only commonly used results. It is not intended to be a complete bibliography on the subject. Each item listed includes other references of possible interest.

E.1 Texts, Handbooks, and Manuals

- (a) W. I. Berks and A. Chwick, "Bibliography on Thermal Elasticity", Polytechnic Institute of Brooklyn, PIBAL Report No. 216, January, 1953, Contract No. AF 33(616)-116.
- (b) B. E. Gatewood, "Thermal Stresses", McGraw-Hill Book Company, Inc., New York, 1957.
- (c) S. Timoshenko and J. N. Goodier, "Theory of Elasticity", 2nd Edition, McGraw-Hill Book Co., Inc., New York, 1951.
- (d) Reactor Handbook, 2nd Edition, Volume II, "Thermal Stresses" by W. S. Flinn, Chapter 5.2, to be published.
- (e) Reactor Handbook, 1st Edition, Volume II, "Thermal Stress Analysis" by R. A. Daane, Chapter 1.9. "Thermal Shock Shields", by R. J. Fritz, Chapter 2.6.

E.2 Papers

- (a) J. S. Born and G. Horvay, "Thermal Stresses in Rectangular Strips-II, Journal Applied Mechanics, 22, 401-406 (1955).
- (b) R. J. Fritz, "Evaluation of Transient Temperatures and Stresses", Trans. ASME 76: 913-921 (1954).
- (c) J. N. Goodier, "Thermal Stress", Journal Applied Mechanics, September, 1957.
- (d) G. Horvay, "Thermal Stresses in Perforated Plates", Proceedings of First U.S. National Congress of Applied Mechanics, Edwards Brothers, Inc., Ann Arbor, March, 1952.
- (e) G. Horvay, "Transient Thermal Stresses in Circular Disks and Cylinders", Trans. ASME, 54: 127-135 (1954).
- (f) A. Mendelson and S. S. Manson, "Approximate Solution to Thermal Shock Problems in Plates, Hollow Spheres and Cylinders with Heat Transfer at Two Surfaces", ASME Paper No. 54-A-264, 1957.
- (g) G. Sonnemann and D. M. Davis, "Stresses in Long Thick-Walled Cylinders Caused by Pressure and Temperature", ASME Paper No. 57-A-256, presented at ASME Annual Meeting, New York, December 1-6, 1957.