THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

FOUNDED 1880

THIS CERTIFICATE IS AWARDED TO

TAE M. AHN

FOR ATTENDING THE COURSE

ASME B&PV CODE: SEC VIII, DIV. 1 + ALTERATION & REPAIR

October 1, 2001
Houston, TX
3.75 CEU

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Vice President
Professional Development

ASME® PROFESSIONAL DEVELOPMENT

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Education and Training

ASME
INTERNATIONAL
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Programs

ASME B&PV Code: Section VIII, Division 1, Design & Fabrication of Pressure Vessels (Includes Alteration & Repair)

Instructor(s): Kamran Mokhtarian
John Mooney

Dates: 10/1/01-10/5/01
Location: HOUSTON, TX

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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Amy Geffen, Ph.D.
Director, Professional Development

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Family Name:	Given Name:	Initial:	Gender: ☐ M ☐ F (Optional)	Birthdate:

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PROFESSIONAL HISTORY AND DATABASE QUESTIONS

COLLEGE(S)/UNIVERSITY(IES) ATTENDED

College/University-Location	Degree	Area of study	Date Degree Awarded

CHECK BOXES FOR ALL EARNED DEGREES

1.☐ No Degree 2.☐ B.A. 3.☐ B.S. 4.☐ M.A.
5.☐ M.S. 6.☐ M.E. 7.☐ Ph.D. 8.☐ Sc.D.
9.☐ Other (Describe)________________

PROFESSIONAL RECORD

Name of current employer	
Job Title	
Length of time in current job	years: months:

Attach resumé or curriculum vitae

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	State/ Country	License number	Date of Issue
☐ P.E.			

REFERENCE

Please complete this section to the best of your ability. Check box if ASME Member

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As a member of The American Society of Mechanical Engineers, you may select five technical divisions from the following list (in priority order).

☐ ☐ ☐ ☐ ☐

27 Advanced Energy Systems
01 Aerospace
02 Applied Mechanics
09 Bioengineering
31 Computers & Information Engineering
14 Design Engineering
25 Dynamic Systems and Control
34 Electronic and Photonic Packaging
23 Environmental Engineering
36 Fluid Power Systems and Technology
08 Fluids Engineering
06 Fuels and Combustion Technology
11 Heat Transfer
38 Information Storage and Processing Systems
05 Internal Combustion Engine
22 International Gas Turbine Institute
03 Management
13 Manufacturing Engineering
10 Materials
04 Materials Handling Engineering
39 Micro Electro Mechanical Subdivision
33 NDE Engineering
32 Noise Control and Acoustics Analysis
17 Nuclear Engineering
35 Ocean, Offshore & Arctic Engineering
16 Petroleum
40 Pipeline Subdivision
21 Plant Engineering and Maintenance
19 Power
28 Pressure Vessels and Piping
12 Process Industries
18 Rail Transportation
07 Safety Engineering and Risk Analysis
30 Solar Energy
29 Solid Waste Processing
24 Technology and Society
20 Textile Engineering
15 Tribology

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NMME01
C6 0:

☐ Select a code based on your PRINCIPAL JOB FUNCTION

01 General or corporate management
02 Product design including applied research and development
03 Systems design including applied research and development
13 Design of Equipment for in-plant use
14 Other Design Engineering Functions
04 Production engineering (processing or manufacturing)
05 Testing, reliability assurance, quality control and standards
06 Plant engineering, operations and maintenance
11 Basic research & development
15 Other Process Engineering Functions
12 Consulting
07 Education
08 Other please specify: ________________

☐ Select a code that best describes your TITLE.

Engineering
31 Corporate V.P. for Engineering, Director, etc.
32 Chief Engineer, Senior Engineer, etc.
33 Project Manager
34 Other (e.g. Engineer, Allied Scientist)

Non-Engineering
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36 Vice President, Senior Manager, etc.
37 Project Manager, Manager or Supervisor
38 Other (e.g., sales, marketing)

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*<u>2001 - 2002 Membership Dues - $111</u>

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(A) I authorize the Society to verify the information contained in this application and, to that end, to contact any educational institution, professional society, publisher, employer or other entity named or identified in this application or in any document submitted in support of this application. I hereby consent to and authorize the release and disclosure to ASME of any information records or correspondence as required to verify the information in the application which is held by any such entity identified in this application.

(B) I subscribe to the above and, if elected to any grade, will be governed by the Constitution and By-Laws of the Society and its Code of Ethics as long as I continue as a member. I furthermore agree to promote the objectives of the Society so far as shall be in my power.

Signature in Ink________________ Date________________

Membership grades will be assigned based on information provided with this form. Additional information may be required for other grades.

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ASME International

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Along with writing and distributing industry codes and standards and providing activities and programs to advance the field of mechanical engineering, ASME provides a vast array of professional resources to its members. From aerospace to fluid power systems to solar energy and more, ASME offers quality forums for technical professionals.

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2000-2001 Membership Dues Schedule

- Member; Executive Affiliate; Associate Member; Affiliate (age 30 and over) – $107
- Member; Associate Member; Affiliate (to age 30) – $80

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ASME Grades of Membership

C3.1.5 A **Member**, at the time of admission or advancement, shall have attainments amounting to the equivalent of at least twelve years of active practice in the profession of engineering or engineering teaching, five years of which shall have been in a position of responsible charge of engineering work. The Member shall be qualified to direct such work or to carry on important research or design in the field of engineering.

Attainment of a degree in an approved engineering curriculum or a baccalaureate degree in an approved engineering technology curriculum shall be accepted as equivalent to eight years of active practice. Furthermore, appropriate credit will be given for the following: graduation from an unapproved engineering or engineering technology curriculum, completion of portions of such curricula, completion of a course of study in a technical institute or other recognized educational programs. (Applies to Member and Associate Member Grades)

The experience of one who has not been graduated from an approved curriculum should show variety, progress and promotion in the performance of engineering functions.

MSWD08

C3.1.6 An **Associate Member**, at the time of admission or advancement to that grade, shall have attainments amounting to the equivalent of at least eight years of engineering experience of a character satisfactory to the Council on Member Affairs.

C3.1.7 An **Executive Affiliate** need not have a formal education in engineering or have practiced as an engineer, but must have (a) attained a position of policy making authority and recognized leadership in some pursuit related to engineering, and (b) demonstrated an aptitude for cooperating closely with engineers and a marked enthusiasm for furthering engineering advances.

C3.1.8 An **Affiliate** should be a person who is capable of and interested in rendering service to the field of engineering; and whose work should be so related to applications of engineering that admission to this grade will contribute to the welfare of the Society.

C3.1.10 A license to practice as a Professional Engineer issued by a legally authorized body whose requirements for licensing are considered adequate by the Council on Member Affairs shall be considered equivalent to eight years of active practice toward any grade of Society membership.

ASME Continuing Education Institute Certificate Program
Pressure Vessel & Piping Program

Name: **Company:**

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❑ WHICH COURSES ARE IN THE CERTIFICATE PROGRAM?

PD #:	Course Title:	Days/ CEUs:
❑ 008	Alteration and Repair Procedures for Boilers and Pressure Vessels	1/.75
❑ 013	ASME B31.1: Power Piping Design and Fabrication	5/3.75
❑ 014	ASME B31.3: Chemical Plant and Petroleum Refinery Piping	4/3.0
❑ 077	Failures, Failure Prevention and Repairs of Pressure Vessels, Piping, Boilers and Rotating Machinery, and Life-Extension Considerations	3/2.25
❑ 181	ASME Boiler and Pressure Vessel Code: Section I, Power Boilers	2/1.5
❑ 184	An Introduction to and Refresher on ASME Section III Div. 1 Requirements for Design and Manufacture of Nuclear Power Plant Components	4/3.0
❑ 186	ASME Boiler and Pressure Vessel Code: Section VIII, Division 1, Design and Fabrication of Pressure Vessels	4/3.0
❑ 187	ASME Boiler and Pressure Vessel Code: Section VIII, Division 1, Design and Fabrication of Pressure Vessels (with Alteration & Repair)	5/3.75
❑ 190	ASME Boiler and Pressure Vessel Code: Section IX, Welding and Brazing Qualifications	3/2.25
❑ 192	ASME Boiler and Pressure Vessel Code: Section XI, In-Service Inspection of Nuclear Power Plant Components	5/3.75
❑ 345	Design, Inspection and Repair of ASME Section Viii, Division 1, Pressure Vessels	3/2.25
❑ 359	Practical Welding Technology	4/3.0
❑ 363	ASME B&PV Code: Section II, Material Issues Revealed	3/2.25
❑ 370	ASME B31.8 Gas Transmission and Distribution Piping Systems	2/1.5
❑ 382	How to Predict Thermal-Hydraulic Loads on Pressure Vessels and Piping	2/1.5
❑ 386	Design of Bolted Flange Joints	2/1.5
❑ 389	Section V	2/1.5
❑ 391	ASME B31.4	2/1.5
❑ 392	Section VIII, Division 2	2/1.5
❑ 394	Seismic Design & Retrofit of Process/ Power Plant Fluid Systems	2/1.5
❑ 395	Recommended Practice for Fitness-For-Service & Continued Operation of Equipment	3/2.25
❑ 398	Operation, Maintenance and Repair of Plant Piping Systems	4/3.0
❑ 401	The Layout of Piping Systems and Process Equipment	2/1.5
❑ 406	Pressure Relief System Design for Section I, IV, and VII Applications	2/1.5

❑ HOW CAN I EARN THE CERTIFICATE?

* You must complete 8 Continuing Education Units (CEUs) within a seven-year period. The CEUs can be obtained by completing either an in-company or public course. The time period is retroactive to January 1993.
* Only courses approved and provided by ASME qualify for the Certificate program.
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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

SHORT COURSE

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

John L. Mooney, PE
Member ASME

REPAIRS AND ALTERATIONS OF BOILERS

AND PRESSURE VESSELS

GENERAL COMMENTS

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

GENERAL

Introduction

The comments that will be offered in this short course will be the instructor's with his understanding of the latest Editions of The National Board inspection Code (including Addendum), API 510 - Pressure Vessel Inspection Code: Maintenance Inspection, Rating, Repair and Alteration, and the laws, rules, and regulations of jurisdictions having pressure vessel laws covering boilers and/or pressure vessels. These will be supplemented with data, past experience, and proven field practices.

The National Board inspection Code is published as a loose-leaf note book and addenda pages will be issued annually and become mandatory six months after publication. Revisions will be made periodically and will be published in the National Board Bulletin.

On request National Board will render interpretations to Code users. When requesting an interpretation it shall be in the following format:

Subject	Cite applicable paragraph number and give a concise description. Specify the edition.
Question	Direct and to the point. Do not request approval.

Appendix E of API 510 provides information on requests for interpretations of API 510 and consideration of revisions to the standard based on new data or technology. The format for interpretations is similar to the above NBIC format.

The following abbreviations will be used in the course notes:

ASME	American Society of Mechanical Engineers
NB	National Board of Boiler and Pressure Vessel inspectors
NBIC	The National Board inspection Code
API	American Petroleum institute
API 510	American Petroleum institute Pressure Vessel inspection Code: Maintenance Inspection, Rating, Repair, and Alteration
Inspector	Authorized inspector (Commissioned by NB) or API 510 Certified Inspector

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

In these course notes, and in the oral presentation, the general term 'pressure vessel' will be used and in most cases the comments will apply equally to boilers and to pressure vessels. The course notes will be used as the basis for the presentation; however, the oral presentation may go into more detail and may cover material not included in the course notes. Paragraphs of a routine nature will not be covered in the notes or the oral presentation.

Both NBIC and API 510 are approved American National Standards Institute (ANSI) documents. The NBIC is accepted by most of the jurisdictions in the United States that have laws covering repairs and/or alterations to boilers pressure vessels, and is mandatory in some jurisdictions. API 510 has been adopted as an alternative document in about 15 states. With the change in the API 510 Sixth edition, that requires a written examination to qualify and certify and API authorized pressure vessel inspectors, it is expected that API 510 will be adopted by other jurisdictions having pressure vessel laws.

In addition to those jurisdictions which have adopted API 510, it is used widely for repairs and alterations by the oil, petrochemical, and chemical industries in those jurisdictions that do not have pressure vessel laws.

NBIC and API 510 provide definitions of repairs and alterations that are similar and generally accepted by regulatory people, owner/users, and repair/alteration organizations.

A repair constitutes rework involving replacement material, and/or replacement parts to restore a pressure vessel to a safe and serviceable condition without deviation from the original construction.

A change in the physical condition (geometry) of the original construction or a change of service constitutes an alteration to a pressure vessel.

NBIC classifies rerating of a pressure vessel, increasing the pressure, increasing the design temperature, or lowering the minimum design temperature, as an alteration. It does not address down rating a pressure vessel. API 510 classifies rerating as a change up or down in the pressure or temperature. Reratings that increase the design pressure or temperature or selected decreases in temperature are also classified as alterations.

The physical size and/or location of a pressure vessel that is installed in the field can present a challenge in making repairs and/or alterations. Shop fabrication permits positioning of pressure vessels to provide access and favorable working conditions for the crafts. Working on a pressure vessel in place at the point of installation may prevent compliance with the literal requirements of the original construction Code and substitutions of alternative methods, techniques and/or procedures may be necessary.

NBIC and API 510 recognize these hardships and permit deviations to the rules where laws exist. Concurrence is usually required from the jurisdictional authority, engineer and the inspector, and there must be agreement by all parties involved that the alternative method will provide safety at least equal to that provided by the rules of the original construction Code.

Jurisdictional Requirements

It is essential that owner/users and organizations repairing and/or altering pressure vessels be knowledgeable regarding boiler and pressure vessel laws, rules, and/or regulations at the place of installation.

A document that is helpful for information regarding pressure vessel laws for new construction, repairs, alterations, and inservice inspection is the SYNOPSIS OF BOILER AND PRESSURE VESSEL LAWS, RULES, AND REGULATIONS, published by the Uniform Boiler and Pressure Vessel Laws Society; located at 1230 Hustbourne Lane, Suite 125C, P. 0. Box 24382, Louisville, KY 40224. This document is only available through membership in the Society. By means of Data Sheets and Bulletins, members are kept informed of actions of governmental bodies adopting or revising laws, rules and regulations for the construction, installation, repair/alteration and inservice inspections of boilers and pressure vessels. In addition, the Society assists its members by providing help with specific problems relating to requirements and enforcement of these laws, rules, and regulations. Membership is available to all who subscribe to the objectives of promoting safe, practical and uniform laws, rules and regulations affecting the manufacturer and the users of boilers, pressure vessels and their appurtenances and who make voluntary contribution to the support of the Society.

The American Petroleum institute also publishes a summary of laws and regulations that pertain to boilers and pressure vessels. The document "DIGEST OF STATE BOILER AND PRESSURE RULES AND REGULATIONS" may be obtained from API; located at 1220 L Street, Northwest, Washington, D.C. 20005.

Most jurisdictions that have pressure vessel laws hold the owner/user responsible for the safe operation of installed pressure vessels and for compliance with the law for procurement, installation, maintenance, repair, alterations, and/or rerating of pressure vessels. It is most important that the owner/user be cognizant of the details of all repairs, alterations, and rerating whether or not laws, rules; and/or regulations exist at the point of operation of the pressure vessel.

Most jurisdictions require the owner/user to file the repair/alteration report even though the responsibility for completing the report rests with the concern that did the work. A few jurisdictions hold the organization that did the work responsible for filing the report. NBIC requires that copies of reports covering alterations to pressure vessels stamped with a NB registration numbers be filed with their office to supplement the original data report. Filing reports covering repairs is optional.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

Many jurisdictions require a repair concern to have the NB 'R' Stamp, an ASME stamp covering a similar scope of work, or department approval to make repairs or alterations to pressure vessels subject to the law. NBIC no longer makes reference to the ASME stamp in reference to a repair or alteration. It is expected that jurisdictions that require NBIC will delete reference to the ASME stamp as their pressure vessel laws are updated.

Many jurisdictions require repair/alteration concerns to have a license to sell, install, repair, and/or alter pressure vessels. In some cases the license is issued to the repair/alteration concern on the basis of making application and paying a fee. In other cases the representative that will be responsible for the repairs/alterations will be required to take a written or oral examination and the license will be issued in the name of that individual. Since the requirements vary from jurisdiction to jurisdiction it is important to investigate the actual requirements at the point of installation of the pressure vessel.

All jurisdictions that have pressure vessel laws covering repairs and/or alterations require the involvement of the inspector and his acceptance of the repair and/or alteration plan before the start of the work. The inspector may want to make inspections during the course of the work. A major revision in the 1995 NBIC is that the Inspector no longer makes the decision regarding the application of the pressure test following a repair or alteration. The repair/alteration organization is required to pressure test to the MAWP unless exempted by the jurisdictional authority. If the hydrostatic test is carried out, NBIC requires that the inspector witness the test. Upon completion of all work he will, if he is satisfied that all requirements have been met, countersign the repair/alteration report that has been completed and signed by a representative of the organization that did the work.

Pressure vessels that are relocated from one jurisdiction to another require particular attention and a careful review of the law at the new location. Some jurisdictions will require a physical inspection and a favorable report on the condition of the pressure vessel before authorizing the installation. If the pressure vessel will be insured at the new point of installation, the insurance carrier will have an interest in the pressure vessel and will normally furnish a inspector to make the physical inspection to determine the condition of the object.

Pressure vessel laws vary on the recognition of the Owner/User inspector and again this points out the importance of investigating the law where the pressure vessel is installed. Anyone interested in owner/user inspection should carefully review NBIC, PART RA-3000, Accreditation of Owner/User inspection Organizations. The Owner/User inspector recognized in NBIC and ASME Code Section VIII Division 1 should not be confused with the API authorized pressure vessel inspector recognized in the API 510. Prior to the Seventh Edition of API 510 the terminology Owner/User inspector was also used in API 510.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

The laws of several jurisdictions will be covered to illustrate how they differ and to emphasize the importance of knowing the law where work is performed. It is important to understand that the Synopsis and API Digest are only guides and it is the referenced law that will apply in all cases.

ASME Certification - it is important to understand that the ASME Stamp is not to be used to certify repair or alteration. The stamp is for new construction only. The stamp can be used to cover the construction of new parts that are installed to complete a repair or alteration and this provision will be covered in more detail later.

Legal Requirements at the Site - Who can make repairs and alterations to a pressure vessel? The first thing that has to be determined is what is required by the law at the point of installation of the pressure vessel. Most jurisdictions that have pressure vessel laws accept the NBIC as an alternative to rules in their law. The problem is that jurisdictions often adopted NBIC by edition, which causes difficulty when the edition changes.

Prior to the 1992 edition of NBIC, it recognized repair/alteration concerns having the "R" stamp or in possession of an appropriate ASME Code stamp. The reference to the ASME stamp has been deleted from the 1992 and 1995 NBIC. There will be confusion for a time because jurisdictions will need to update laws to be consistent with NBIC. As stated earlier most jurisdictions do not make NBIC mandatory. That being the case, the "R" stamp, ASME stamp, or department approval is not a requirement unless specifically required by the law.

The requirement for API 510 or the "R" stamp also brings forth the requirement for a formal quality control systems and the implementation of the program in carrying out a repair and/or alteration to pressure vessels. Some jurisdictions do not recognize NBIC or API 510 and in such cases their regulations will apply.

Selection of a Repair / Alteration Contractor - The selection of an outside contractor will start after it is determined that the work for whatever reason is outside the capability of the owner/user. Owner/Users should look for all the protection possible to avoid court actions in the event of an accident. Repair/alteration concerns will normally follow recognized standards if they exist for the very same reason. What both parties may try to avoid is the formality of a quality control system. This formality can be a challenge for planned work and it becomes difficult to satisfy for an unscheduled emergency outage. However, it is the only way to insure that the repair/alteration is per the recognized standard.

If the law at the point of installation does not require a recognized standard, then the contract may set forth specific requirements for the acceptance of the repair/alteration concern. This acceptance usually rests with the owner/user and the inspector. The acceptance of the repair or alteration concern by the inspector is going to be by his review and acceptance of the work plan, agreement on inspections during the work, and his final acceptance of the completed work.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

Scope of Work - The scope of work will certainly play a major part in selecting the organization to conduct the repair/alteration to a pressure vessel. Many owner/users have in-house capability and do perform their own work to the degree that it is possible. Usually in-house capability is limited to the routine maintenance work necessary to keep a plant running; the kind of work that will support regular full time employees. Scheduled major outages will require additional manpower to minimize plant down time. Such outages may also require equipment and skills that are beyond the plant capability. Utilities, because of the cost of purchased power, will normally go outside for additional support during planned outages.

Original Equipment Manufacturer - it is common practice to use the original equipment manufacturer to conduct repairs or alterations to utility boilers, large industrial boilers, and complex process vessels and process systems. This arrangement is particularly desirable when the following are required: original drawings, procedures, and/or techniques; engineering support or lab support; specialized tools; replacement assemblies requiring shop fabrication; highly specialized technicians, and design changes.

The original equipment manufacturer's familiarity with his own equipment, availability of special tools, procurement capability, control over manufacturing facilities for replacement assemblies, and his ability to assign experienced people can minimize down time.

There will be occasions when the original equipment manufacturer cannot respond because of work load, geographic location, not staffed for field work, cannot supply supervision, or even union restrictions. Under such conditions, a manufacturer of like equipment might be the next choice. The possibility of a site representative from the original equipment manufacturer as an adviser should be considered. It is also possible to consider the work with in-house people under the control of a representative of the equipment manufacturer.

Other Factors - Other factors enter into the selection of an organization for a repair or an alteration. Probably the next consideration is past experience and the present business relationship with the repair/alteration concern. Ability to mobilize and move in the required equipment, particularly for an unscheduled outage, is an important consideration. A concern may be selected because they are already on site doing other work and the cost of another concern to mobilize and move in equipment is saved. Unfavorable operating experience with the equipment, or even personalities, may rule out the use of the original equipment manufacturer.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

Scheduled Outages

Planning - Repairs and alterations fall into two categories, scheduled and unscheduled. Scheduled outages are planned and organizations exist that specialize in planned outages and will enter into long term maintenance agreements with owner/users. Unscheduled outages usually are the result of unexpected equipment failures. Proper planning and regular scheduled maintenance will minimize unscheduled outages.

Planned inspections, accurate data, and a review of the data precede a scheduled outage. The data is used to order material and develop the work scope for the next outage. The up front inspections and the collection of data can take place months, or years, before carrying out the actual work. A common practice is to make inspections and collect data during a scheduled outage and use it for planning the work scope and material procurement for the next scheduled outage.

There are organizations that specialize in contract maintenance work. These organizations have trained, qualified, and certified technicians, up-to-date techniques and procedures, and equipment to perform all types of nondestructive testing.

New techniques in ultrasonic testing permit the determination, with reasonable accuracy, of the extent of corrosion, hydrogen damage, and caustic gouging as well as obtaining exacting thickness measurements. Evaluation of data obtained from visual inspections, ultrasonic testing, radiography, magnetic particle and liquid penetrant examination permits planning and the procurement of critical, long lead time materials and parts to minimize the outage.

Often the supplier of the original equipment will have historical data on the installed equipment, as well as experience with like equipment at other locations that will permit them to zero in on vulnerable areas. Original equipment manufacturers will normally have programs for life extensions to predict, with reasonable accuracy, the remaining service life of various parts of the equipment.

Service experience also permits long range planning of repairs and/or alterations to pressure vessels along with improved operating efficiency and/or serviceability. As stated earlier, the data developed by inspection and testing prior to and during an outage is reviewed and studied by engineering and metallurgical personnel and recommendations developed for present and future work. Materials and/or fabricated parts procurement lead times may be critical for a planned outage and are always a challenge during an unscheduled outage.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

Financial people must also be involved in the planning of a repairs or alterations. Available cash and/or tax considerations may determine the extent of the repair. Cash flow is important in every business and keeping inventories to a minimum is important. Some repair / alteration organizations, (normally original equipment manufacturers) will enter into contracts with owners and users to pool critical parts so that they will be available for an outage. Pooling replacement parts is more common among nuclear plants.

Replacement materials and parts can be a problem, particularly when small quantities are obtained from a supply house. Materials for all pressure retaining parts must be materials that are included in the applicable Code covering the installed pressure vessel. Only material test reports for plate product forms are required for ASME Section VIII, Division 1 vessels. However, material test reports are required for all product forms in repairs/alterations to Section VIII-Division 2 pressure vessels. Except for Section VIII-Division 2, product forms other than plate shall include the material marking as required by the applicable material specification. If the material test report is not available, or if the required marking does not exist, it will be necessary to obtain lab tests for the chemistry and mechanical properties of each piece used in the repair/alteration. The properties obtained by the lab test will then have to be identified with one of the acceptable materials in the applicable Code covering the installed pressure vessel.

Purchased parts fabricated by welding should be ordered under the rules covering the original construction. When the ASME Code applies, parts should be stamped with the applicable ASME Code stamp and be furnished with the applicable ASME Code parts data report. Some welded pressure retaining parts for ASME pressure vessels can be obtained under existing standards. The most common of such standards used in ASME Code construction will be welded parts under SA-234, SA-403, and/or SA-423. Section I and Section VIII-Division 1 also provide for small parts to manufacturer's standards.

Work Plan - Communication with the owner/user, repair/alteration organization, and the inspector is essential to accomplish a successful outage. Concurrence on the selection of the inspector to be involved should be the first item on the work plan. NBIC holds the repair/alteration concern responsible for obtaining the services of the Inspector, but does not dictate the source to be used. API 510 holds the owner/user responsible for obtaining the services of the inspector. Every effort should be made to use the inspector employed by the inspection agency insuring the equipment and who is responsible for the inservice inspection to the jurisdictional authorities. Having one Inspector overview the work of another inspector may result in both parties being conservative.

The second step in the work plan should be an agreement between the owner/user and the repair/alteration concern on the scope of the work to be carried out during the outage. The development of the repair plan should follow the agreement on the work scope.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

After there is agreement between the owner/user and the repair/alteration concern, the repair/alteration plan must be submitted to the inspector for his acceptance and his selection of inspection points. Effort should be made to minimize or avoid hold points by the Inspector since hold points can impact the work schedule.

Supervisory personnel and craft people need to be selected and, if necessary, trained and qualified.

Substitutions of Materials, Parts & Procedures - it is necessary in the initial stage to reach agreement on substitutions of materials and/or parts, if such substitutions are necessary.

It may not be possible to meet the new construction requirements for radiography. Agreement to waive the inspection or use ultrasonic inspection may be necessary.

Postweld heat treatment to the requirements for new construction may be impossible and alternative methods must be developed and accepted by the inspector.

Nondestructive testing and postweld heat treatment activities are often subcontracted and agreement on the subcontractor's qualifications is necessary. All variances should be cleared before the start of the work to avoid problems during and after completion.

Inspection Records - The inspection record required by NBIC is the R-1 Repair Report or R-2 Alteration Report. These reports must be completed by the repair/alteration concern and countersigned by the Inspector. Copies of parts data reports, if parts were incorporated into the repair or alteration, must to be attached.

API 510 requires that the owner/user maintain records throughout the service life of the equipment. Those records must include all information regarding the repair, alteration and rerate of the equipment.

Quality Control System - Regardless of the law and/or contract requirements, basic controls are required to complete any repair or alteration. Receiving inspection, in-process inspection, material control (in particular weld material control), and document control programs are essential as a minimum at every work site.

The need for implementing a formal quality control system for a repair or alteration, and the sophistication of the system, must be determined in the planning stage. Both NBIC and API 510 require a complete QC system.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

Pressure Testing - Pressure testing after a repair or alteration is completed requires careful consideration. Water chemistry and water supply should be the first considerations. The method to heat the water to increase the metal temperature above the nil ductility temperature is a primary consideration to minimize the possibility of a brittle fracture failure. A test pump and calibrated test gauge must be available. Precautions must be taken to valve and/or blank off connected piping and to tag and/or lock all stop valves to prevent operation during the test. Proper venting of the pressure vessel and roped areas to keep unauthorized personnel out of the area are necessary. The test pressure needs to be determined during the planning stages and if the repairs or alterations are be done under jurisdictional requirements. The Inspector must be involved and must concur with the decisions and witness the test.

Under the 1995 NBIC, the repair or alteration organization is required to perform a pressure test unless exempted by the jurisdictional authority. Pressure tests above the MAWP of the pressure vessel are permitted. Pressure tests above the MAWP after major repairs or alterations are recommended for vessels with service conditions with a risk of brittle fracture. If replacement parts are being installed it is good practice to test these new parts to 1.5 times the MAWP pressure before the parts are installed. In many cases this practice will satisfy the inspector and will avoid an over pressure test of the entire pressure vessel. If a pressure vessel is to be tested above the maximum allowable pressure shown in the stamping it is essential that a complete inspection of the pressure vessel be made to determine it's physical condition and the data obtained evaluated by engineering to assure that damage will not result. Test requirements are covered under RC-3030 in the NBIC Code and 6.5 in API 510 and it is suggested that these paragraphs be reviewed.

Metal temperature during the pressure test is probably the most important consideration in order to prevent a brittle fracture failure. Catastrophic brittle fracture failures have resulted in severe property damage, serious injuries, and deaths. NBIC, API 510, ASME Section 1, ASME Section VIII, and Section III are not uniform regarding test temperature requirements. Section 1, Power Boilers, addresses the water temperature and not the metal temperature. Heavy wall vessels installed outside will require extended periods of time to obtain a safe metal temperature. Section I states the water temperature shall be at least 70F.

Section VIII-Division 1 for new construction was revised by the December 31,1987 addenda and recommends a hydrostatic test metal temperature at least 30F above the minimum design metal temperature of the pressure vessel. For pneumatic testing Section VIII makes it mandatory that the metal temperature shall be 30F above the minimum design metal temperature. The problem with the revision is that, except for vessels constructed under PART ULT, the minimum design metal temperature will only appear in the stamping of pressure vessels constructed after the 1987 revision to Section VIII-Division 1. Figure UCS-66 can be used as a guide for determining metal temperature for the test.

API 510 requires consideration of metal temperature similar to the requirements of Section VIII-Division 1, except that API 510 will permit 10F above the minimum design metal temperature for thin wall vessels.

Section III has the tightest control and requires that the nil ductility temperature of all material used in the construction be determined and recommends that the material temperature in the completed pressure vessel during the pressure test be 60F above the minimum nil ductility temperature. This practice is not practical for vessels that are in service, unless this information was developed during new construction and is available in the inspection records. The required design specification for Section III and Section VIII Division 2 pressure vessels contains such information and should be reviewed when repairing/altering such pressure vessels.

Metal temperature at the time of a pressure test should not be ignored on the basis that the test pressure will be below the design pressure of the pressure vessel, particularly if new materials and/or fabricated pressure parts have been installed. Brittle fracture failures have occurred below the design or operating pressure.

It will be noted that NBIC and API 510 permit, with jurisdictional approval, waiving the pressure test. It is important that there is concurrence with the inspector. The conditions of each Code should be reviewed.

Unscheduled Outages

The course notes up to this point have been directed at planned outages that cover a considerable scope of work. The overnight or over a weekend repair job is usually more of a challenge, particularly if it is being carried out in a jurisdiction that has adopted the "R" stamp and it is necessary to implement a quality program.

Regardless of the location or the scope of work involved the principles remain the same. The owner/user must have in-house capability, the "R" stamp if a requirement, or enter into a contract with a qualified concern.

Just like a planned outage, the first thing that must be determined is the scope of work that will be the basis for the repair plan. The activities can be handwritten on note paper with freehand sketches, if necessary, or if time permits engineering drawings can be generated.

Once the repair plan is developed the inspector needs to be involved for the acceptance of the plan prior to the start of the work. In some cases this will be accomplished over the telephone. Without his concurrence any work that is carried out in a jurisdiction that has a law covering repairs or alterations will be with risk that it might be rejected. Evidence should be available to show that the effort was made to contact the inspector, confirmation by TELEX or FAX is a recommended practice.

API 510 SUPPLEMENT 2 requires consideration of metal temperature similar to the requirements Section VIII-Division 1, except that API 510 will permit 10F above the minimum design metal temperature for thin wall vessels.

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Material, document, and process control, proper welding procedures, qualified welders, PWHT and NDE procedures(if required) and good inspection reports are a minimum whether or not a law exists. If a law exists, these items are most important if the work is started without the involvement of the Inspector.

Temporary Repairs

Temporary repairs are sometimes essential to restore a pressure vessel to service pending the completion of permanent repairs. Temporary repairs are frequently a consideration because of the lead time required for replacement material and fabricated parts. Reduced capacity may be a factor to consider in return to service. If safety of operating personnel is a consideration, many of the jurisdictional authorities will require that their office review and accept the repair plan. If the equipment is insured the insurance company will require their concurrence of the proposed repair. Failure to obtain their acceptance could result in a deductible or a cancellation of the coverage.

In making temporary repairs, the repair concern will in most cases invoke terms and conditions into the contract to provide financial protection for their organization. While the owner/user and insurance carrier may be willing to take a commercial risk regarding possible physical damage to the object and the surrounding property, the safety of people can never be put in jeopardy. Roped off areas, barriers, and/or explosion mats are often used to protect people required to be in the area for operating purposes.

REPAIRS AND ALTERATIONS OF BOILERS
AND PRESSURE VESSELS

NBIC

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

1995 NATIONAL BOARD INSPECTION CODE

The NBIC overview will overlap with the earlier course notes and with some of the material covered in the oral presentation. The course notes and oral presentation will offer comments on the following contents in the 1995 NBIC:

Foreword	
Introduction	
Part RA	Administrative Requirements
Part RB	Inservice inspection of Pressure Retaining Items
Part RC	Repairs and Alterations of Pressure Retaining Items
Part RD	Repair Methods
Appendix 1	Preparation of Technical inquiries to the NBIC Committee
Appendix 2	Stamping and Nameplate information
Appendix 4	Glossary of Terms
Appendix 5	National Board Forms
Appendix 6	Examples of Repairs and Alterations to Pressure Retaining items
Appendix A	Standard Welding Procedures
Appendix B	Recommended Preheat Temperatures

The course notes and the oral presentation will not offer comments on the following:

Appendix 3	Steam Locomotive Firetube Boiler inspection and Repair
Appendix 7	Procedures to Extend the "VR" Certificate of Authorization Stamp to ASME 'NV' Stamped Pressure Relief Devices
Appendix D	Recommended Guide for the Design of a Test System for Pressure Relief Devices in Compressible Fluid Service
Appendix E	Recommended Procedures for Repairing Pressure Relief Valves
Appendix F	Pressure Differential Between Safety or Safety Relief Valve Setting and Boiler or Pressure Vessel Operating Pressure
Appendix G	Safety Valves on the Low Pressure Side of Steam Pressure Reducing Valves

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

FOREWORD

Reading the FOREWORD is recommended. The expression "pressure retaining items" is used, permitting NBIC to be used for repairs/alterations to pressure vessels, boilers, non-code pressure vessels and piping systems. Previous NBJC editions were interpreted to limit the Code to boilers and pressure vessels and further to boilers and pressure vessel constructed to the ASME Boiler and Pressure Vessel Code. The expression 'pressure vessels" will continue to be used in the course notes except when a requirement is specific to a boiler, unfired pressure vessel, non-code pressure vessel, or piping.

It will be noted that the objective of the rules are to promote uniformity and address protection of life and property as well as achieve a long and safe period of usefulness. Reference is made to Mandatory Appendix 1 for guidance in seeking interpretations or offering requests for revisions.

It is recognized that NBIC does not contain rules to cover all details of repairs/alterations. Where complete details are not given, it is intended that the repair/alteration organization, subject to the approval of the owner/user and the acceptance of the Inspector, provide details for the repair or alteration which will be as safe as provided by the rules of the original code of construction. NBIC users are cautioned that the jurisdiction where pressure retaining items are installed may require approval. In some cases the jurisdiction will require approval by a registered professional engineer experienced in pressure vessel design.

While not stated, it is recognized that the edition of the Code covering the pressure vessel, or materials used in the original construction, may not be available. In such cases, it is the intent that the latest edition of the Code be used to the extent possible. Where it not possible or practical to meet the original construction Code, engineering principles that will provide safety and serviceability equivalent to the original construction may be used. Jurisdiction approval may be required. See RC-1020 and APPENDIX B-17.

The latest edition of the NBIC is not to be made retroactive.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

INTRODUCTION

Reading of the INTRODUCTION is recommended.

NBIC provides guidance to jurisdictional authorities, inspectors, users and organizations for the purpose of encouraging uniform administration of rules regarding pressure retaining items.

NBIC does not provide details for all conditions. Code users must seek technical guidance where details are not provided.

ADDENDA are issued annually. ADDENDA, as in the ASME Codes covering new construction, may be used when issued and become effective six months after issuance. The addenda is included in the purchase price of the Code and will be sent to NBIC holders.

The NBIC recognizes API 510 covering maintenance, inspection, repair, alteration, and rerating of pressure vessels, and API 570 covering piping. These Codes are used by the petroleum and chemical industries. NBIC covers installations not covered by the API Codes unless jurisdictions rule otherwise.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

PART RA - ADMINISTRATIVE REQUIREMENTS

RA-1000 GENERAL

RA-1010 SCOPE

This part describes requirements for the accreditation of repair organizations and for the accreditation of Owner/User Organizations. It is appropriate since the owner/user may be accredited and accepted by the jurisdiction to conduct repairs, alterations, or inservice inspections. An owner/user may also be accredited as a repair/alteration concern and obtain the "R" symbol stamp to fabricate parts, repair and/or alter pressure retaining items. Structured in the Owner/User Organization can be Inspectors with NB Owner/User Commissions.

Accreditation programs are available for the following:

"R" Symbol Stamp
"VR" Symbol Stamp
"NR" Symbol Stamp
Owner/user inspection Organization

RA-2000 ACCREDITATION OF REPAIR ORGANIZATIONS

The NBIC also includes accreditation for the 'VR' for over pressure protective devices and 'NR' for nuclear components; however, these programs are not covered herein.

RA-2100 "R" ADMINISTRATIVE RULES AND PROCEDURES

RA-2110 SCOPE

The Scope clarifies that issuance of the NB Certificate of Authorization and the "R" Symbol Stamp is not restricted to repair/alteration organizations. Owner/User Organizations, as covered in RA-1010, may obtain Certificates of Authorizations and the NBIC "R" Symbol Stamp and perform repair/alterations to their own pressure retaining items.

NBIC cautions that a jurisdiction may have their own rules and NBIC may not be acceptable. Most jurisdictions having laws covering repairs/alterations to pressure retaining items state the NBIC is acceptable, a few make NBIC mandatory.

Pressure vessel users and repair/alteration organizations should have copies and know the laws, rules, and regulations where they operate. The inspector will know the laws and should be brought into programs during the planning stages.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

The NBIC makes no reference to the ASME Code and clarifies the types of NB Certificates of Authorizations available. Six types of authorizations are offered for the "R" Symbol Stamp :

Repairs Only	Shop Only
Repairs Only	Field Only
Repairs Only	Shop & Field
Alterations & Repairs	Shop Only
Alterations & Repairs	Field Only
Alterations & Repairs	Shop & Field

The NB Certificate of Authorization will be noted accordingly.

RA-2120 PREREQUISITES FOR ISSUING A NATIONAL BOARD CERTIFICATE OF AUTHORIZATION

Before making application to NB, organizations need:

a. Inspection agreement with authorized inspection agency unless making application as Owner/User inspection Organization.

b. Written quality system addressing location scope of work (RA-2130).

c. A copy of the current NBIC.

d. Have, or have access to, construction Codes(s) covering the contemplated work, usually latest editions.

RA-2130 PROCEDURE FOR OBTAINING OR RENEWING A NATIONAL BOARD CERTIFICATE OF AUTHORIZATION

The following is required prior to the issuance or renewal of the NB Certificate of Authorization and "R" stamp:

a. Apply to NB for application forms. Applications for renewal shall be submitted at least six months prior to expiration.

b. Separate applications for each plant/shop. Application to designate location and repair only or repair/alteration.

c. Organization required to make arrangements for the review.

d. Review team as a minimum will have representatives from authorized inspection agency and jurisdiction. If jurisdiction is the inspection agency or declines, an NB Representative will lead the review. Jurisdiction may elect to participate as a team member when NB representative leads the review. It is normal for the inspector assigned to the shop to participate.

e. Review team will review quality control system and observe implementation of the quality control system on production work or mock-up.

f. There will be an exit interview and the team will recommend to issue or recommend withholding the Certificate of Authorization. If the recommendation is to withhold the applicant has the right to appeal to the NB Committee on Accreditation.

g. The jurisdiction may audit the quality system upon request by owner/user, inspection agency, or National Board.

h. Notify National Board of any changes of ownership or address changes and any changes in contracts with Authorized inspection Agencies.

i. Provides for holders of ASME Code Stamps to obtain NB "R" Stamp without review of the facility provided:

   1. The organization has a quality control system to cover the scope of the repairs/alterations to be made, subject to review by the jurisdiction.

   2. The ASME Certificate of Authorization was issued within one year of the application for the NB "R' Certificate of Authorization. The initial "R" certificate expires concurrently with the ASME Certificate. NB review is required for renewal.

j. The jurisdiction may audit the Quality System and activities of the organization holding a NB Certificate of Authorization at any time.

k. NBIC Committee may at any time change the rules for the issuance of the NB Certificate of authorization.

RA-2140 NATIONAL BOARD "R" SYMBOL STAMP

RA-2140 addresses use and control of the NB "R" Stamp. RA-2140 does not require comments in the course notes.

RA-2150 QUALITY SYSTEMS

RA-2150 is generic and does not require comments in the course notes.

RA-2151 OUTLINE OF REQUIREMENTS FOR A QUALITY SYSTEM FOR QUALIFICATION FOR THE NATIONAL BOARD "R" STAMP

RA-2151 is easy reading and does not require comments in the courses. It is suggested that the organization seeking NB Accreditation work with their Authorized Inspection Agency in developing the required quality control system.

RA-2200 "VR" ADMINISTRATIVE RULES AND PROCEDURES

Not covered in the course notes or oral presentation. Interested parties may want to attend courses presented by NB on over pressure protective devices. Course information, and date of presentations, can be obtained by writing the NB.

RA-2300 "NR" ACCREDITATION REQUIREMENTS

Not covered in the course notes or oral presentation.

RA-3000 ACCREDITATION OF OWNER/USER INSPECTION ORGANIZATION

RA-3020 PREREQUISITES FOR ACCREDITATION

The following is required to be accredited as an Owner/User inspection Organization:

a. Must be an organization that owns/operates pressure retaining items.

b. Must employ inspectors that have NB Owner/User Commissions.

c. Organization structure and procedures that support inspection process.

d. Must meet jurisdictional requirements at location.

e. Program and facilities reviewed and approved by jurisdiction.

f. Have current edition of NBIC.

RA-3030 PROCEDURE FOR OBTAINING A NB OWNER/USER CERTIFICATE

- Owner/user organization must apply to the NB.
- The review will be conducted by a jurisdictional representative.
- A review report, with recommendations is forwarded to the NB.
- A 3 year Owner/User certificate will be issued if fees are paid and requirements met.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

RA-3040 ADMINISTRATIVE PROCEDURES

Must have a written program that addresses:

1. Statement signed by a senior company official indicating the authority and responsibility of the Inspector and the conflict resolution process.

2. Organizational structure. Inspector line of authority must be independent of product production or operation.

3. Process used by Inspector if an unsafe conditions is encountered.

4. Written inspection procedures.

5. Continuing training program.

6. Process used to insure that those performing inspection services, such as NDE are qualified.

7. Inspection report preparation, maintenance and distribution process.

RA-3050 GENERAL CONDITIONS

The Owner/User can obtain the NB "R" stamp and the Owner/User Inspection Organization can perform inspections of in-house repairs and alterations.

It is also to be noted that the Owner/User can obtain the NB "R" Certification of Authorization and stamp to carry repairs or alterations and use an Authorized Inspection Agency without being accredited as an Owner/User Inspection Organization.

An NB Commissioned Owner/User Inspector working under the accepted Owner/User Inspection Organization quality and inspection programs can make inservice and repair/alteration inspections if permitted by the jurisdiction.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

PART RB INSERVICE INSPECTION

Only the parts of this section that pertains to Inservice inspection of Pressure Vessels will be covered.

RB-2000 PERSONNEL SAFETY

Requires no comments in course notes.

RB-3100 INSPECTION OF BOILERS

RB-3120 PRE-INSPECTION ACTIVITIES

NBIC states that the operating conditions should be considered when establishing inspection criteria and gives only general guidance. Pre-inspection should include a review of the boiler history and include such information as:

- Operating history
- Date of last inspection
- Current jurisdictional inspection certificate
- ASME or other code of construction markings
- National Board and/or jurisdiction registration number

The boiler should be cool and clean and the following parts removed as required:

- Manholes and handholes covers
- Washout plugs
- Inspection plugs in water column connectors
- Grates of internally fired boilers
- Insulation and brickwork when necessary
- Pressure gage should be removed and tested unless other information verifies its accuracy.

RB-3131 ASSESSMENT OF INSTALLATION

Conduct an external inspection to determine if boiler's condition is such that it can operate safely. The following should be checked:

- General cleanliness and accessibility of boiler, boiler area and auxiliary apparatus.
- Compliance of boiler fittings, valves and piping with ASME Code or other codes and standards.

RB-3132 CAUSES OF DETERIORATION

Boiler deterioration may be caused by:

- Improper or inadequate water treatment
- Excessive fluctuations in pressure and/or temperature or

- Improper or lack of maintenance.

RB-3133 TYPES OF DEFECTS

Boiler defects may include:

- Bulged plates or tubes
- Blistered plates or tubes
- Cracks or other weld or heat-affected zone defects
- Pinhole leaks
- Improper or inadequate safety devices
- Corroded or eroded material

RB-3140 EVIDENCE OF LEAKAGE

Insulating material, masonry or fixed parts of a boiler are not normally removed for inspection unless:

- Defects or deterioration are suspected or are commonly found in the type of boiler.
- Evidence of leakage is present.

Pressure test may be necessary to obtain additional information regarding a leak or other defect. If pressure tested, observe the following:

- Test pressure need not exceed the stamped MAWP if for tightness.
- During the test the relief device should be prepared per manufacturer's advice.
- The water temperature should not be less than 70°F. The maximum temperature during inspection should not exceed 120°F.
- Pressure test hold time shall be 10 minutes prior to examination.
- Hold time shall be as required for examination.

RB-3151 PIPING

Piping should be inspected to ensure:

- Provisions for expansion
- Provision for adequate support
- No evidence of leakage
- No evidence of excessive vibration
- Proper alignment of connections
- Proper rating for the service
- Freedom of the piping to expand and contract during blowdown
- That blowoff piping is properly secured and discharges at a safe location.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

RB-3152 WATERSIDE DEPOSITS

All accessible surfaces of exposed metal on the water side should be inspected for deposits caused by:

- Water treatment
- Scale. Any excessive scale or deposits should be removed.
- Oil
- Other substances

RB-3153 STAYS

- Examine all stays to determine if in even tension; if not,adjust or repair.
- Examine stayed plates for cracks at stay holes.
- Replace all broken stays.
- Examine for leakage if staybolts have telltale holes.
- Hammer test all firebox staybolts.

RB-3154 FLANGED AND OTHER CONNECTIONS

Examine for defects both internally and externally.

- manholes,
- reinforcing plates,
- nozzles,
- other flanged or bolted connections,

Examine and ensure that all openings leading to external attachments such as;

- water column connections,
- low water fuel cutoff devices,
- openings in dry pipes,
- openings to safety valves, etc.

are free from obstructions.

RB-3156 OVERHEATING

- Can cause bulges, blistering and damaged tubes.
- Pay particular attention to surfaces exposed to the fire.
- Repair bulges or blisters large enough to weaken plates or tubes or when evidence of leakage is present.
- When repairing, blistered areas should be removed and required thickness provided.
- Water tube bulges should always be repaired.
- Bulges on plates may be driven back or flush patches installed.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

RB-3157 CRACKS

- Cracks may result from:
  - + Existing material flaws
  - + Design details
  - + Operating conditions
  - + Fatigue due to continual flexing
  - + Thermal differential when cooling is inadequate
  - + A combination of the above.

- Cracks noted in shell plates should be repaired.

- Fire cracks that run from the edge of the plate into rivet holes of girth seams may be left in place.

- Thermal fatigue cracks determined by engineering analysis to be self arresting may be left in place.

- Examine areas where cracks are most likely to occur. This include:
  - + ligaments between the tube holes in watertube boiler drums
  - + between the tube holes on the tube sheet of firetube boilers
  - + from and between rivet holes
  - + any flange where there may be repeated flexing
  - + around welded pipe and tube connections.

- Lap joint boilers are subjected to cracking in the longitudinal seam. If there is evidence of leakage or other sign of distress, the inspector should examine the area for cracks. The plates may be notched or slotted if necessary.

- Repair of lap joint cracks on longitudinal seams are prohibited.

- NDE may be necessary to locate cracks.

RB-3158 CORROSION

- Most common cause is free oxygen and dissolved salts in feedwater.

- If found inspector should advise owner so that feedwater treatment is adjusted.

- Metal thickness may be determined by UT or drilling.

- Inspect for grooving
  - + adjacent to riveted joints
  - + flanged surfaces - use mirror when necessary

- Inspect for accelerated corrosion on the fireside surfaces of tubes in horizontal firetube boilers.

- Inspect the upper ends when exposed to heat of combustion of vertical tube boilers.

- The upper tube sheet in a vertical "dry top" boiler should be inspected for overheating.

- Inspect for pitting and corrosion on the waterside surface of tubes.

- Excessive corrosion and pitting is often found at and above the water line in vertical firetube boilers.

- Carefully examine tube surfaces for corrosion, erosion, bulges, cracks and defective welds.

- Inspect restricted fireside spaces, such as where nipples are used to join drums or header, for fuel and ash deposits which can cause corrosion.

RB-3159 MISCELLANEOUS

- Inspect piping to water column to ensure that water cannot accumulate in the steam connection.
- Check the position of the water column to verify compliance with the ASME or other code of construction.
- Inspect gas side baffling.
- Check location and condition of combustion arches. Look for evidence of flame impingement.
- Check for localization of heat cause by improper or defective firing equipment.
- Examine supports and settings to ensure that deposits of ash or soot will not bind or prevent proper operation.
- Inspect re-rolled or replaced tubes for proper workmanship.

RB-3161 GAUGES

- Ensure the accuracy of the indicated water level by testing the gage as follows:
  1) Close the lower gage glass valve, then open the drain cock and blow the glass clear.
  2) Close the drain cock and open the lower gage glass valve. Water should return to the gage glass immediately.
  3) Close the upper gage glass valve, then open the drain cock and allow the water to flow until it runs clean.

4) Close the drain cock and open the upper gage glass valve. Water should return to the gage glass immediately.

Repeat the above operation if the water return is sluggish. Correct any leakage.

- Each hot water boiler should be equipped with a temperature gage at or near the boiler outlet that will indicate the water temperature at all times.

- Remove and test all pressure gauges and compare reading to a standard test gage or a dead weight tester where required.

- Check the location of the steam pressure gage and make sure that provisions are made for blowing out the pipe leading to the gage.

RB-3163 CONTROLS

- All low water fuel cutoff and water feeding devices should be inspected to ensure proper installation and operation.
- Float chamber type control devices may be disassembled and examined for wear and sludge.
- Check that the following control are provided:
  1) Each automatically fired steam boiler is protected from overpressure by not less than two pressure operated controls.
  2) Each automatically fired hot water boiler is protected from over temperature by not less than two temperature operated controls.
  3) Each hot water boiler is fitted with a thermometer that will at all times indicate the water temperature at or near the boiler outlet.

RB-3170 RECORDS REVIEW

The inspector should review the:
- boiler log,
- records of maintenance and
- feed water treatment

to ensure that regular and adequate tests have been performed.

The Inspector should consult the owner or user regarding repairs and alterations made since the last inspection. All should be reviewed for compliance with the applicable requirements.

RB-3180 CONCLUSIONS

- The Inspector should note all operating and maintenance practices and determine their acceptability. All defects or deficiencies in the condition, operating and maintenance practices should be discussed with the owner/user and recommendations regarding corrections made.

RB-3200 INSPECTION OF PRESSURE VESSELS

RB-3220 PRE-INSPECTION ACTIVITIES

General guidance is given. Review information such as:

- Operating conditions
- Normal contents of the vessel
- Date of last inspection
- Current jurisdictional inspection certificate ASME Code Symbol Stamping or mark of code of construction
- National Board and/or jurisdictional registration number
- Records of wall thickness checks, especially on vessels where corrosion is a consideration

The following activities should be performed as required to support the inspection:

- Inspection plugs and covers should be removed.

- Vessel should be sufficiently cleaned to allow for visual inspection of internal and external surfaces.

- Pressure relief devices, see RB-3500.

RB-3230 ASSESSMENT OF INSTALLATION

NBIC states that inspections may be internal or external and that on-stream inspections may be used to satisfy inspection requirements provided that accuracy of the method can be demonstrated.

The purpose is that the inspection provide the necessary information the essential sections of the vessel are of a condition to continue to operate for the expected time interval.

RB-3231 EXTERNAL INSPECTION

Purpose is to provide information regarding the overall condition of the vessel.

The inspection shall as a minimum include;

1. Determination of condition of exterior insulation. Examine surfaces under damaged insulation and at nozzles and other areas where water can enter, particularly where below about 250°F.

2. Determination of the condition of the supports and support alignment,

3. Investigation of any signs of leakage.

4. Examination of vessel connections,

5. Examination of general condition of surface, that is dents, distortion, cuts gouges, etc.

6. Inspection of rivet heads, butt strap, shanks plate and caulked edge conditions on riveted vessels.

RB-3232 INTERNAL INSPECTION

NBIC states that all parts of the vessel should be inspected for corrosion, erosion, hydrogen blistering, deformation, cracking and laminations.

As a minimum the inspector should inspect the following;

- Vessel Connections
- Vessel Closures, especially quick actuating type
- Vessel internals
- Corrosion, especially at the bottom, liquid level lines, areas opposite inlet nozzles

RB-3233 NON-DESTRUCTIVE EXAMINATION (NDE)

NBIC list all applicable NDE methods and states that they should be implemented by experienced and qualified individuals. Methods listed are:

- Magnetic Particle
- Liquid Penetrant
- Ultrasonic
- Radiography
- Eddy Current
- Visual
- Metallographic examination
- Acoustic Emission

RB-3234

NBIC does not require a pressure test as part of inspection but one should be made when inspection discloses unusual or hard to evaluate forms of deterioration that may affect the vessel.

A pressure test to determine tightness need not be greater than the set pressure of the lowest pressure relief valve.

PressuKre test shall not be greater than 1 1/2 times the MAWP adjusted for temperature. It may be increased if the original test pressure was increased to test the corrosion allowance.

If pressure tested the following must be observed:

> THE METAL TEMPERATURE SHALL BE ≥ 60 °F UNLESS TOUGHNESS DATA INDICATES A LOWER TEMPERATURE IS ACCEPTABLE.
>
> THE METAL TEMPERATURE SHALL NOT EXCEED 120 °F DURING EXAMINATION UNLESS THE OWNER AND INSPECTOR AGREE OTHERWISE.

Test media other than water may be used.

RB-3235 REMAINING LIFE & INSPECTION INTERVALS

When the corrosion rate controls the life of a vessel the remaining life shall be calculated by:

$$RL = \frac{t_{actual} - t_{minimum}}{CR}$$

$$CR = \frac{t_{previous} - t_{actual}}{\text{years between}}$$

Where,

tactual = the thickness measured at the time of inspection

tminimum = the minimum allowable thickness

tprevious = the thickness at the same location measured during a previous inspection.

NBIC states that $t_{minimum}$ must be the greater of,

1. The calculated thickness, exclusive of corrosion allowance, required for the pressure relieving device set pressure or

2. The minimum thickness permitted by the provision of the applicable section of the original code of construction.

Corrosion rates for new vessels or change of service may be determined by:

- Data collected in the same or similar service.

- Inspector's estimate based on experience or knowledge in similar service.

- On-stream thickness measurements at approximately 1000 hour intervals.

RB-3237 INSPECTION INTERVAL

- The internal inspection or complete on-stream inspection shall not exceed one-half of the estimated remaining life of the vessel or ten years, whichever is less.

- If the remaining life is estimated to be less than four years, the inspection interval may be the full remaining life up to a maximum of 2 years.

RB-3238 CONDITIONS THAT AFFECT REMAINING LIFE EVALUATION

The inspection interval should be adjusted based on the following:

a) Deterioration other than corrosion.

b) An external inspection may be substituted for an internal inspection provided:

   1. Five year history with non-corrosive character of content and corrosion rate is zero.

   2. No questionable condition is disclosed by the external inspection,

   3. The operating temperature of the steel is below the creep range,

   4. The vessel is protected against inadvertent contamination.

c) Corrosion Resistant Lining

Vessel or vessel section inspection intervals may be based on the history of the lining but not greater than 10 years.

d) Two or More Zones

The inspection interval for a zone of a vessel may be based on data for that zone if a significant difference exists (corrosion rate, temperature)

e) Above Ground Vessels

- All above ground vessels shall be externally inspected every 5 years or at the quarter corrosion-rate life, whichever is less.

- The inspection shall as a minimum include;

    1. Determination of condition of exterior insulation,

    2. Determination of the condition of the supports and support alignment,

    3. Investigation of any signs of leakage.

    4. Examination of vessel Connections,

    5. Examination of general condition of surface, that is dents, distortion, cuts gouges, etc.

    6. Inspection of rivet heads, butt strap, shanks plate and caulked edge conditions on riveted vessels.

f) Interrupted Service

The stated inspection intervals assume continuous operation interrupted only by normal shutdowns. The effect of extended off-line intervals must be considered in establishing inspection intervals.

Vessels must be given an inspection before placing in service if the vessel is out of service for one year or more.

Actual thickness and deterioration shall be determined by:

- Any NDE method provided the following tolerances are met:

Wall thickness in. (mm)	Tolerance
≤5/16 in (8 mm)	0.10t
≥5/16	greater of 1/32 in (.8mm) or 0.05t

- Measurements through openings.

- Drilled holes.

g) Circumferential Stresses

When circumferential stresses governed the MAWP, the least thicknesses in the most critical plane may be averaged over a length "L" not exceeding:

1. The lesser of 1/2 the vessel diameter or 20 inches for diameters of 60 inches or less.

2. The lesser of 1/3 the vessel diameter or 40 inches for vessels with diameters greater than 60 inches.

If the area contains an opening then "L" is limited to the limits of reinforcement on either side of the opening.

h) Longitudinal Stresses

If longitudinal stresses govern, then "L" shall be a length of arc in the most critical plane.

I) Pitting

Widely scattered pits may be ignored when:

- No pit depth greater than 1/2 the required wall thickness exclusive of corrosion allowance.

- Total area of pits does not exceed 7 square in. in any 8 in. circle.

- The sum of their dimensions does not exceed 2 in. along a straight line within the 8 in. circle.

j) Weld Joint Efficiency Factor

Thicknesses shall be based on weld joint efficiency of the weld except if surfaces are remote to the weld then may use E=1.0. Weld surface is defined as the greater of one inch on either side of weld or two times the minimum thickness on either side of the weld.

k) Ellipsoidal and Torispherical Heads

When measuring corroded thickness of ellipsoidal or torispherical heads the governing thickness may be:

- Thickness of knuckle region - use appropriate head formula.
- Thickness of central portion - use spherical shell formula.

l) Adjustment of Corrosion Rate

The corrosion rate for determining the inspection frequency shall be adjusted if any inspection shows that an inaccurate corrosion rate was used.

m) Riveted Pressure Vessels

The effect of corrosion on the failure mode must be considered when determining the strength of riveted joints. Credit may be taken for excess thickness.

RB-3240 CAUSES OF DETERIORATION

List possible deterioration mechanisms:

Corrosion

Fatigue

Creep

Low Temperature

Temper Embrittlement

Hydrogen Attack

Hydrogen Embrittlement (<200°F, reversible)

Stress Corrosion Cracking

RB-3260 GAGES, SAFETY DEVICES AND CONTROLS

Gages, safety devices and controls should be checked to verify proper operation.

RB-3270 RECORDS REVIEW

A permanent record shall be maintained for each vessel. The following should be included.

a. An ASME Manufacturers' Data Report or, other equivalent specifications.

b. Form NB-5 Boiler or Pressure Vessel Data Report - First Internal Inspection

c. Complete pressure relieving device information including safety or safety relief valve data.

d. Progressive records that include as a minimum the following:

1. Location and thickness of monitor samples and other critical inspection locations.

2. Limiting metal temperature and location on the vessel when this is a factor in establishing the minimum allowable thickness.

3. Computed required metal thicknesses and maximum allowable working pressure for the design temperature and pressure relieving device opening pressure, static head and other loadings.

4. Test pressure if tested at the time of inspection.

5. Scheduled (approximate) date of next inspection

6. Date of installation and date of any significant change in service conditions temperature, character of content or rate of corrosion

7. Drawings showing sufficient details to permit calculations. Sketches may be used if drawings are not available.

RB3300 INSPECTION OF WATER HEATERS (For other than space heating))
RB3400 INSPECTION OF PIPING SYSTEMS Basic information.
RB3500 INSPECTION OF SAFETY DEVICES FOR OVERPRESSURE PROTECTION
RB3600 INSPECTION OF DEARATORS
RB3700 INSPECTION OF LIQUID AMMONIA VESSELS
RB3800 INSPECTION OF PRESSURE VESSELS WITH QUICK ACTUATING CLOSURES. Good basic warnings on these potentially dangerous devices.
RB4000 REPACEMENT OF STAMPED DATA Nameplate replacement.

b) Covers standard pressure retaining items that are preassembled by attachment welds.

c) Covers procuring replacement parts to the ASME original code of construction. An ASME Manufacturer's Partial Data Report is required.

d) Covers replacement parts when the original construction code is other than the ASME code. The organization furnishing the part shall be qualified under the original code of construction. When this is not possible the part can be fabricated under the NB "R"Symbol stamp and furnished with the R-3 Part Form.

RC-1060 AUTHORIZATION

This paragraph follows the 1992 NBIC requiring the inspector's authorization to perform a repair/alteration and for the inspector providing the authorization to follow the repair/alteration to completion, or as a minimum an inspector from the same authorized inspection agency.

RC-1070 INSPECTOR

This paragraph establishes the source of the inspector. The inspector can be from the jurisdiction, the inspection agency under contract with the "R' Symbol Stamp holder, the inspection agency insuring the pressure retaining item, or from the Owner/User Inspection Organization.

RC-1090 WELDING

References the original construction code.

RC-1091 WELDING PROCEDURE SPECIFICATIONS

References the original construction code. Provides for Section IX of the ASME Code when not possible or practical to meet original construction code.

RC-1092 STANDARD WELDING PROCEDURE SPECIFICATIONS

APPENDIX A recognizes twenty seven ANSI/AWS Standard Welding Procedures that may be used for repairs/alteration to pressure retaining items. The repair/alteration organization must assume full responsibility for the specifications.

RC-1093 PERFORMANCE QUALIFICATION

Requires the original construction code or Section IX of the ASME Code.

RC-1094 RECORDS

Does not require comments in course notes other than the Certificate holder must maintain records of WPQ results.

RC-1095 WELDERS' IDENTIFICATION

Standard wording requiring a system that will identify a welder with a weld.

RC-1096 WELDERS' CONTINUITY

Requires a system to establish that a welder has welded under the process within six months. Provides for requalification for cause.

RC-1100 HEAT TREATMENT

RC-1101 PREHEATING

Recommended preheat temperatures are provided in APPENDIX B. The welding procedure specification shall cover preheat temperatures used in carrying out the repair/alteration to the pressure retaining item.

RC-1103 ALTERNATIVE POSTWELD HEAT TREATMENT METHODS

Alternative methods require the acceptance of the inspector. He may insist on approval by a metallurgist and/or welding engineer. Details will be covered in PART RD.

RC-1110 NONDESTRUCTIVE EXAMINATION

Nondestructive examinations shall be as required by the original construction code. NDE technicians shall be qualified as required by the original construction code.

RC-1120 PRESSURE GAUGES, MEASUREMENT, EXAMINATION AND TEST EQUIPMENT

Follow original construction code.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

RC-1130 ACCEPTANCE INSPECTION

The inspector as a minimum is required to review drawings, assure welding to original construction code, witness pressure test if conducted, and assure required PWHT and NDE is carried out.

RC-1140 STAMPING

Follow APPENDIX 2.

RC-1141 REMOVAL OF ORIGINAL STAMPING OR NAMEPLATE

Requires jurisdictional approval, inspector to witness removal of stamping/nameplate and restamping or attachment of the name plate. Any relocation of the stamping or nameplate shall be described on the NBIC repair form.

RC-1150 REGISTRATION OF DOCUMENTATION

Registration of repairs/alterations is optional except repair/alteration data reports shall be furnished to the National Board for ASME pressure vessels registered with National Board. See RC-3052 that requires filing with the NB if the vessel was registered with NB.

It is to be noted that some jurisdictions require registration of all repairs/alterations.

RC-2000 ADDITIONAL REQUIREMENTS FOR REPAIRS

Review list of repairs in Appendix 6.

RC-2010

This section provides additional requirements for repairs to pressure retaining items.

RC-2020 DRAWINGS

Drawings that include sufficient information shall be prepared to cover repairs. While these drawings will be formal for planned outages they may take the form of sketches on note pads in the event of emergency outages.

RC-2030 AUTHORIZATION

As stated earlier repairs to pressure retaining items shall not be initiated without the acceptance of the inspector. Note that it is stated that the inspector accepts the repair plan before the work is started, he does not go on record as approving the plan.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

Subject to the acceptance of the jurisdiction, the inspector may give prior approval for routine repairs without his review and acceptance of each activity. Normally the procedures required for routine repairs will be covered in the quality system and will be reviewed and accepted at the time of the review for the National Board Certificate of Authorization. Typical routine repairs are covered in RC-2031.

RC-2031 ROUTINE REPAIRS

The categories of routine repairs are outlined in paragraphs a through d. The R-1 Report shall be completed by the organization making the repair and shall be noted "Routine Repair" on line 9 The control Of the R-1 Report shall be outlined in the quality control system. Some jurisdictions will require that all reports for repairs, including authorized routine repair reports, be filed with their office. The jurisdiction is that at the installation location.

RC-2050 EXAMINATION AND TEST
C-2051 METHODS

The "R" Certificate Holder is responsible for the examinations and tests. Repairs shall be verified by examination or test, subject to the acceptance by the inspector and, where required, by the jurisdiction. The test pressure shall not exceed 150 % of the MAWP stamped on the item, adjusted for temperature. The minimum pressure shall be that required to verify the leak tightness of the repair.

It is recommended to pressure test major repairs where there is a risk of brittle fracture. This can be established for pressure vessels by using the impact testing exemption curves of ASME Section VIII Division 1.

Care shall be taken to check the over pressure protective device, which might be set below the test pressure. If the test pressure will exceed the setting, the device shall be prepared as recommended by the device manufacturer.

The metal temperature shall be per the original code of construction but not less than 60°F. The test temperatures established in Section VIII-Division 1 are recommended.

A hold time of ten minutes at test pressure before the inspector is to make his inspection of the pressure retaining item is established.

Where the weight of the vessel will not support a liquid test, or where liquid contamination can not be tolerated, a pneumatic test can be conducted with the acceptance of the inspector. Proper safety precautions should be taken during the pneumatic test.

Any NDE methods used shall be suitable for providing meaningful results to verify the integrity of the repair.

Pressure testing following an alteration or rerating will be covered under RC-3030.

RC-2060 STAMPING

APPENDIX 2 covers stamping and nameplates. Stampings are required for repairs and alterations. APPENDIX 2 is self explanatory and will be covered later in the course notes.

RC-2070 DOCUMENTATION

APPENDIX 5 (National Board Forms) is self explanatory and will not be covered in the course notes.

RC-2071 PREPARATION OF FORM R-1

The NB 'R' Certificate Holder shall prepare, sign, and distribute the Form R-1, attaching to the Form R-1 copies of parts data reports that cover any fabricated part used in the repair.

The inspector shall countersign the Form R-1 indicating his concurrence to the best of his knowledge the NB 'R' Certificate Holder did satisfy NBIC requirements.

RC-2072 DISTRIBUTION

The repair concern is required to distribute completed Form R-1, with attachments to the owner/user, inspector, jurisdiction where required, and the Authorized inspection Agency responsible for inservice inspection of the item.

The National Board will accept registration of Form R-1 covering repairs to registered pressure vessels if desired by the owner/user or the repair concern.

RC-2080 REPAIR OF ASME CODE SECTION VIII-DIVISION 2 PRESSURE VESSELS

RC-2082 REPAIR PLAN

A repair plan is routine for any repair. However, a repair plan reviewed and certified by a registered professional engineer indicating compatibility to user's design specification and manufacturers design report is mandatory when conducting a repair to a Section VIII-Division 2 pressure vessel.

The repair plan shall then be submitted to the Authorized inspection Agency whose inspector will sign the R-1 report following the completion of the repair.

RC-3000 ADDITIONAL REQUIREMENTS FOR ALTERATIONS

Review list of alterations in Appendix 6.

RC-3010 SCOPE

RC-3000 paragraphs contain additional requirements for alterations to pressure retaining items. Re-rating is covered under RC-3000 and is classified under 1995 NBIC as an alteration. API 510 does not treat all rerates as alterations.

RC-3020 DESIGN

The NBIC provides for a "R" Certificate Holder to alter or re-rate pressure retaining items. The capabilities of the organization to alter or re-rate pressure retaining items is evaluated at the time of the review and the organization's NB Certificate of Authorization is endorsed to establish his scope of activities.

Alterations to pressure retaining items shall meet the original code of construction. The edition can be the edition that was used in the original construction or the latest edition of the applicable code. It is unfortunate that the NBIC does not make a clear statement regarding the applicable code.

If a Manufacturer's Data Report was required for the original construction, a copy shall be obtained for use in the design consideration of an alteration or rerating of a pressure retaining item. If the stamping includes NB registration, a copy of the Data Report, for a reasonable charge, can be obtained from NB headquarters. If the manufacturer's data report is not available, agreement on the design basis to be used shall be obtained from the inspector and the jurisdiction.

RC-3021 CALCULATIONS

Calculations covering the scope of the alteration or rerating shall be completed and made available for review by the inspector accepting the design before the start of work. The inspector will determine that complete calculations do exist and he may review them to the extent he deems necessary before his acceptance. The inspector is not required to approved the validity of the calculations.

RC-3022 RE-RATING

It will be noted that 1995 NBIC addresses rerating as increasing the internal/external working pressure or temperature or decreasing the minimum temperature. While NBIC address only upgrading pressure retaining items, it is common for jurisdictions to address down grading pressure retaining items as a result of corrosion or erosion. API 510 addresses upgrading and downgrading. Jurisdiction acceptance shall be obtained for rerating and the following information shall be submitted:

a) Calculations developed as set forth in "R" Certificate Holders quality control system. Normally, as a minimum, the quality control system will require an engineer experienced in design of pressure retaining items generate the calculations and a check by a second engineer. The quality control system could, as once was a NBIC requirement, require design calculations by a registered professional engineer.

b) Calculations shall be to the original construction code.

c) Current inservice inspection records shall be available to verify condition of the pressure retaining items to support the rerating.

d) Evidence that a pressure test for the new design condition as required for new construction was conducted.

RC-3024 ALLOWABLE STRESSES

For rerating at higher allowable stresses than were used originally, see RD-3000.

RC-3030 EXAMINATION AND TEST, RC-3031 METHODS

a) Test requirements in the original code for new construction shall be satisfied when the alteration involves rerating. This will normally be at 1.5 MAWP, temperature adjusted.

b) NDE may be substituted for pressure testing when pressure testing is not practical. However, this requires the concurrence of the owner, the Inspector and the jurisdiction where required. *It is recommended to pressure test major alterations where there is a risk of brittle fracture. This can be established for pressure vessels by using the impact testing exemption curves of ASME Section VIII Division 1.*

c) If the design includes corrosion allowance, the test can be calculated using the as built metal thickness.

d) The NB "R" Certificate Holder is responsible to determine that fabricated parts by others are tested at the plant where fabricated or at the pressure vessel site.

e) As an alternative to pressure testing connecting welds per the original code of construction, these welds may be tested or examined per the rules for repairs.

f) Over pressure devices need to be addressed by removing or blocking as recommended by the manufacturer of the item. Instrumentation and trim need to be evaluated to determine if suitable for the 1.5 MAWP test pressure.

g) Test temperature shall not be less than 60F unless characteristics of the materials used in the construction indicate the acceptability of a lower metal temperature for the test. The metal test temperature should not exceed 120F. The owner/user may, with the acceptance of the inspector, test at a higher temperature. Personnel safety is a prime consideration when testing at elevated temperatures.

h) Hold time for the pressure test shall be ten minutes prioe to examination by the inspector. The NBIC does not provide a caution; however, it is recommended, for personnel safety, that the test pressure be lowered to MAWP during inspections, as permitted in the ASME code.

i) When, because of contamination or the pressure retaining item not being able to support the weight of the liquid, a pneumatic test as established by the original construction code can be used.

RC-3040 STAMPING

Stamping is per APPENDIX 2 and is self explanatory and will not be covered in the course notes.

RC-3050 DOCUMENTATION

Alterations performed in accordance with 1995 NBIC shall be documented on Form R-2 as shown in APPENDIX 5. Form R-4 shall be used to record additional information when space is insufficient on Form R-2.

RC-3051 PREPARATION

Form R-2 and the new Form R-3 are required for alterations and rerating. APPENDIX 5 is self explanatory and will not be covered in the course notes. A copy of the original data report shall be attached to the Form R-2

RC-3052 DISTRIBUTION

The NB 'R' Certificate Holder shall develop and sign Form R-2, Form R-3, and Form R-4 as applicable to cover the alteration or rerating. The applicable forms need to accepted and signed by the inspector. Distribution by 'R' Certificate Holder shall be to the inspector, Owner/User, Authorized inspection Agency, and NB if the pressure retaining item has a NB registration. If not registered, a copy of the applicable form(s) to the NB is optional.

RD-1000 WELDING METHODS AS ALTERNATIVES TO POSTWELD HEAT TREATMENT

RD-1010 SCOPE

RD-1010, RD-1020, RD-1030, RD-1040, and RD-1050 require careful reading and evaluation by knowledgeable an experienced welding engineer and/or metallurgist when using welding methods as alternatives to postweld heat treatment. NBIC recognizes that PWHT as required by the applicable new construction code may be inadvisable or not possible in making a repair/alteration to existing pressure retaining items. There are three alternative welding methods that may be used. The alterrative welding methods should only be implemented with the approval of an experienced welding engineer or metallurgist. Other methods may be used if supported by an experienced welding engineer or metallurgist and acceptable to the inspector and jurisdiction.

RD-1020 NONDESTRUCTIVE EXAMINATION OF WELDS

- PT or MT areas to be welded
- PT or MT finished repair weld
- If repair depth greater than 3/8" and originally RT required by original Code, then RT. If RT not possible then MT or PT all accessible surfaces and reevaluate MAWP.

RD-1030 WELDING METHOD 1

- Use when notch toughness testing not required.
- P-No 1, Group 1, 2 and 3 and P-No 3, Group 1 and 2 materials only.
- Limited to shielded metal-arc, gas metal-arc, fluxcored arc and gas tungsten-arc welding processes.
- Welders and WPS must be qualified.
- Use 300°F minimum preheat and 450°F maximum interpass temperature.

RD-1040 WELDING METHOD 2

- Use when notch toughness testing required.
- Limited to carbon and low alloy steels, with limitations.

- Limited to shielded metal-arc, gas metal-arc, fluxcored arc and gas tungsten-arc welding processes.
- Welders and WPS must be qualified. Test plate thickness and repair grooves must be per Table 1.
- Depth of repair groove not limited provided test material satisfies required thickness.
- Test material must be same material specification.
- Organization making repair and qualifying the WPS must demonstrate sufficient toughness in weld metal and HAZ.
- Charpy impact tests required.
- Additional variables listed for the WPS.

RD-1050 WELDING METHOD 3

- Use when notch toughness testing not required by original code.
- Limited to P-No 1 and P-No 3 materials.
- Limited to shielded metal-arc and gas tungsten-arc welding processes.
- Welders and WPS must be qualified. Test plate thickness and test grooves must be per Table 1.
- Depth of repair groove not limited provided test material satisfy required thickness.
- Test material must be same P-No and Group No. as repair material.
- Additional variables listed for the WPS.
- The welding technique shall be the controlled deposition, temper bead or half bead technique.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

RD-2000 REPAIR METHODS

RD-2010 SCOPE

RD-2000 paragraphs provide repair rules and illustrations for unstayed boiler furnace cracks, rivet or stay bolt hole cracks, wasted areas, access openings, flanges, seal welding of tubes, re-ending or piercing tubes, flush patches, tube patches, and replacement of threaded stays. RD-2000 paragraphs are self explanatory and should be reviewed.

Minor Cracks: Should be examined to determine extent of the defect or whether repair by welding is required.

Flanges: Machined in place to thickness of original flange or thickness supported by calculations. May weld build-up if needed.

RD-3000 ALTERATIONS BASED ON ALLWABLE STRESS VALUES

The following apply for rerating by using a later edition of the original code which permits higher allowable stress values than in the original construction;

- The "R" organization shall verify operation at the new service conditions by calculations
- Lethal service and high-cycle/fatigue controlling service are not permitted.
- The item shall have been built to the 1968 Edition or later.
- The higher stress edition of the code shall apply.
- Operating history shall be satisfactory and damage shall be repaired.
- Rerating shall be acceptable to the inspector and the jurisdiction (if required).
- Part RC applies.
- Document this on form R-2.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

MANDATORY APPENDICES

APPENDIX 1 PREPARATION OF TECHNICAL INQUIRIES TO THE NATIONAL BOARD INSPECTION CODE COMMITTEE

The NBIC Committee meets regularly to consider written request for interpretations and revisions and to develop now rules as dictated by technological development. It is expected, and encouraged, with the new edition and new format, that questions will be generated.

It is important that the format in Appendix 1 be followed in making an inquiry for interpretations or revisions. The cover letter can provide background and should make precise reference to the NBIC paragraph in question. When requesting an interpretation the question and proposed reply should be in the format that NBIC uses to publishes inquiries. The question should be condensed and precise and worded so that the reply can be a simple yes or no.

APPENDIX 2 STAMPING AND NAME PLATES

The single stamping in the 1992 NBIC repair/alteration is replaced in the 1995 NBIC with separate stamping for repair and separate stamping for alteration.

A new stamping for re-rating that includes the Certificate Holder's NB "R" Symbol Stamp and certificate number is required.

The Certificate of Authorization and stamping for replacement parts is new in the 1995 NBIC. This provides for repair/alteration concerns to furnish replacement parts and not have an ASME Certificate of Authorization and stamp.

APPENDIX 3 STEAM LOCOMOTIVE FIRETUBE BOILER INSPECTION AND REPAIR

Not covered in the course notes or oral presentation.

APPENDIX 4 GLOSSARY OF TERMS

APPENDIX 4 should be reviewed. It is self explanatory and is not covered by the course notes. Appropriate references will be made to APPENDIX 4 in the oral presentation.

APPENDIX 5 NATIONAL BOARD FORMS

APPENDIX 5 should be reviewed in particular the part covering Form R-1, Form R-2, Form R-3, and Form R-4. It will be noted that Form R-1 no longer covers repairs or alterations and separate form now exists for each activity. Form R-3 for replacement parts by NB 'R' Certificate Holder and Form R-4 supplementary are new.

APPENDIX 6 EXAMPLES OF REPAIRS AND ALTERATIONS

APPENDIX 6 is self explanatory.

APPENDIX 7 PROCEDURES TO EXTEND THE "VR" CERTIFICATE OF AUTHORIZATION AND STAMP TO ASME "NV" STAMPED PRESSURE RELIEF DEVICES

Not covered by the course.

APPENDIX 8 INSPECTION, REPAIR AND ALTERATION OF GRAPHITE PRESSURE EQUIPMENT

Not covered by the course.

APPENDIX A STANDARD WELDING PROCEDURES

APPENDIX A is self explanatory. Standard welding procedures are reference under course notes covering repairs and alterations and will be covered in more detail in the oral presentation.

APPENDIX B RECOMMENDED PREHEAT TEMPERATURES

APPENDIX B provides minimum temperatures for preheat as a general guide. It cautions that individual materials may have preheat requirements more or less restrictive than the general guide. The preheat used in conducting the repair/alteration shall be included in the welding procedure specification. APPENDIX B should be reviewed.

APPENDIX C HISTORICAL BOILERS (These are riveted)
APPENDIX D RECOMMENDED GUIDE FOR THE DESIGN OF A TEST SYSTEM FOR PRESSURE RELIEF DEVICES IN COMPRESSIBLE FLUID SERVICE
APPENDIX E RECOMMENDED PROCEDURES FOR REPAIRING PRESSURE RELIEF VALVES
APPENDIX F PRESSURE DIFFERENTIAL BETWEEN SAFETY OR SAFETY RELIEF VALVE SETTING AND BOILER OR PRESSURE VESSEL OPERATING PRESSURE
APPENDIX G SAFETY VALVES ON THE LOW PRESSURE SIDE OF STEAM PRESSURE REDUCING VALVES
APPENDIX H RECOMMENDED GUIDE FOR THE INSPECTION OF PRESSURE VESSELS INLP GAS SERVICE

Not covered in the course notes or the oral presentation.

REPAIRS AND ALTERATIONS OF BOILERS
AND PRESSURE VESSELS

API 510

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

API 510 PRESSURE VESSEL INSPECTION CODE

FOREWORD

Reading the FOREWORD is recommended. It gives the history of API 510 starting with the 1934 Edition of API/ASME Code for Unfired Pressure Vessels for Petroleum Liquids and Gases.

SECTION 1 - SCOPE

Covers maintenance inspection, repair, alteration, and rerating procedures for pressure vessels used by petroleum and chemical process industries .

Application is restricted to organizations that employ or have access to an authorized inspection agency as defined in 3.4,

Use is restricted to organizations that employ or have access to engineering and inspection personnel or organizations that are technically qualified to maintain, inspect, repair, alter, or rerate pressure vessels.

API 510 inspection code applies to vessels constructed in accordance with:

- API/ASME Code for Unfired Pressure Vessels for Petroleum Liquids and Gases,
- ASME Code Section VIII,
- Other recognized pressure vessel codes,
- Nonstandard vessels,
- Jurisdictional specials.

The following vessels are excluded from API 510:

- Vessels on movable structures
- All containers listed as exemptions in ASME Code Section VIII, Div. 1.
- UM vessels as defined in Section VIII, Division 1.

REPAIRS AND ALTERATIONS OF BOILERS AND PRESSURE VESSELS

SECTION 2 - REFERENCES

API

RP 572 Inspection of Pressure Vessels
RP 574 Inspection of Piping, Tubing, Valves and Fittings
RP 576 Insperction of Pressure-Relieving Devices
RP 579 Fitness-For-Service
Publ 2201 Procedures for Welding or Hot Tapping on Equipment in Service
API 510 Inspector CertificationExamination Body of Knowledge
Guide for Inspection of Refinery Equipment, Chapter II, "Conditions Causing Deterioration or Failures" (out of print)

ASME

Boiler and Pressure Vessel Code Section V, VI, VII, VIII, IX and XI

NACE

RP0472 Methods and Controls to Prevent In-Service Environmental Cracking of CS Weldments in Corrosive Petroleum Environments
MR0175 Standard Materials Requirements, Sulfide Stress Cortrosion Cracking Resistant Metallic Materials for Oilfield Equipment

National Board

National Board Inspection Code

WRC Bulletin 412 Challenges and Solutions in Repair Welding for Power and Processing Plants

ASNT

CP-189 Standard for Qualification and Certification of Nondestructive Testing Personnel
Recommended Practice SNT-TC-1A

SECTION 3 - DEFINITIONS

Alteration
ASME Code
Authorized Pressure Vessel Inspector (AI)
Authorized inspection Agency
Construction Code
Inspection Code
Jurisdiction
Maximum Allowable Working Pressure
Minimum Allowable Shell Thickness
On-Stream Inspection
Pressure Vessel
Pressure Vessel Engineer
Quality Assurance
Repair
Repair Organization
Rerating
Examiner
Controlled Deposition Welding

SECTION 4 - OWNER-USER INSPECTION ORGANIZATION

4.1 GENERAL

Places the responsibility of an authorized inspection agency on owner/user of vessel who controls the frequency of inspection or maintenance of his vessels. Control of maintenance inspection, ratinkg, repairs and alterations is optional for the owner/user inspection organization.

4.2 API AUTHORIZED PRESSURE VESSEL INSPECTOR QUALIFICATION AND CERTIFICATION

Combined education and experience at least equal to:

a) Degree in engineering plus 1 year experience.

b) A 2 year certificate in engineering or technology from a technical college plus 2 years experience.

c) Equivalent of high school education plus 3 years experience.

d) Five years experience in the inspection of boilers or pressure vessels.

Additionally, certified by an agency per this code.

4.3 OWNER-USER ORGANIZATION RESPONSIBILITIES

Responsible for developing, documenting, implementing, executing, and assessing pressure vessel inspection systems meeting requirements of this code. Quality Assurance Inspection Manual includes:

a) Organization & reporting structure
b) Documentation/maintenance of inspection and QA procedures
c) Documentation and reports of inspection and test results
d) Corrective action
e) Internal audits
f) Review & approval of drawings, calculations & specifications
g) All jurisdictional requirements met
h) Reporting changes to API Authorized Inspector

i) Training requirements
j) Welding controls
k) NDE controls
l) Material controls
m) Calibration
n) Controls by owner-User for inspection & repair organizations
o) Auditing of pressure-relieving device QC system

4.4 API AUTHORIZED PRESSURE VESSEL INSPECTOR DUTY

When inspections, repairs or alterations are being conducted:

- Inspector responsible to owner-user for meeting API 510 requirements
- Directly involved in inspection activities
- May be assisted by other trained and qualified individuals
- Must evaluate and accept all examination results

SECTION 5 - INSPECTION PRACTICES

5.1 PREPARATORY WORK

OSHA confined space regulations.

Company confined space entry procedure.

Basic steps prior to confined space entry:

Isolate
Drain
Purge
Clean
Ventilate
Gas Test
Obtain entry permit
Have person standby at entrance
Check tools and personal safety equipment

5.2 MODES OF DETERIORATION AND FAILURE

- Contaminants in fluids may react with vessel material and cause corrosion
- Stress reversals due to temperature and pressure changes (frequent, at high secondary stress points)
- Welds of materials with different thermal coefficients of expansion
- Excessive temperatures - creep. Consider the following in developing an inspection plan and assessing remaining life:
  a. Creep deformation and stress rupture
  b. Creep crack growth
  c. Effect of hydrogen on creep
  d. Interaction of creep and fatigue
  e. Metallurgical effects, including ductility reduction
- Water and some chemicals at subfreezing temperatures
- Brittle failure at ambient temperature for some carbon and low alloy steels
- Low alloy steel temper embrittlement
- Stress corrosion cracking
- Hydrogen attack
- Carburization or graphitization
- Erosion

5.3 CORROSION-RATE DETERMINATION

For new vessel or a service condition change use one of following methods to determine the rate:

Calculated from data collected by owner or user from same or similar service.

Estimated using previous experience or published data.

On stream determinations after 1000 hours service using NDE thickness or corrosion monitoring devices.

If an inaccurate rate has been assumed, the rate used for the next period shall be adjusted to agree with the actual rate.

5.4 MAXIMUM ALLOWABLE WORKING PRESSURE DETERMINATION

Established using latest ASME Code edition or original Code edition to which vessel constructed. Rerating is required if new MAWP is greater than original.

Computations can be performed only if following details are known and comply with the applicable code:

- Head
- Shell
- Nozzle reinforcement design
- Material specifications
- Allowable stresses
- Weld efficiencies
- Inspection acceptance criteria
- Cyclical service requirements

In corrosive service the wall thickness used shall be actual thickness determined by inspection (average thickness) minus twice the estimated corrosion loss before the next inspection, except as modified in 6.4.If the measured thickness is greater than that in the material test or manufacturers data reports, it must be confirmed by multiple measurements and the procedure approved by the AI.

Allowances shall be made for UG-22 loadings other than pressure.

5.5 DEFECT INSPECTION

Examine for visual indications of distortion.

Visual inspection method is the most important.

Other NDE methods used to supplement visual.

Adequate surface preparation important. There is no one method for accomplishing this.

Not necessary to remove external or internal coverings such as insulation, refractory, corrosion-resistant lining, etc. if they are in good shape and there is no reason to suspect that an unsafe condition exists. However, it may be advisable to remove small portions to check effectiveness.

Removal of internals not necessary if reasonable assurance exists that deterioration is not occurring to an extent beyond that in more accessible parts.

5.6 INSPECTION OF PARTS

Examine surfaces of shells and heads for cracks, blisters, bulges and other signs of deterioration.

Examine welded joints and adjacent heat affected zones for service induced cracks and other defects.

Examine riveted joints.

Examine surfaces of manways, nozzles and other openings for distortion, cracks and other defects.

Examine flange faces for distortion and condition of gasket-seating surfaces .

5.7 CORROSION AND MINIMUM THICKNESS EVALUATION

Corrosion may cause uniform loss or have a pitted appearance.

Minimum actual thickness and maximum corrosion rate may be determined by:

- NDE thickness testing;
- Drilled test holes;
- Measurements through openings;
- Gaging from uncorroded surfaces.

Averaging shall be used for large areas;

- For inside diameters less than or equal to 60 inches:

  Lesser of 20 in. or 1/2 vessel diameter.

- For inside diameters greater than 60 inches:

  Lesser of 40 in. or 1/3 vessel diameter.

When circumferential stress governs critical element is along longitudinal axis.

When longitudinal stress governs critical element is a circumferential arc.

Widely scattered pits may be ignored when:

- No pit depth greater than 1/2 (original design) wall thickness exclusive of corrosion allowance.
- Total area of pits does not exceed 7 square in. in any 8 in. circle.
- The sum of their dimensions does not exceed 2 in. along a straight line within the 8 in. circle.

When corrosion of weld surfaces with joint factor other than 1.0 and surfaces remote from weld occurs, two calculations are required to determine which governs the MAWP.

Surface at a weld includes the greater of 1 in. or twice the minimum thickness on either side of weld.

When thinning of walls occurs that is below the minimum wall thickness requirement, a design by analysis evaluation using Section VIII, Division 2 may be employed. Resultant wall thickness may be less than that determined by calculation using section VIII, Division 1. When this approach is used, consulting with a pressure vessel engineer is required

When measuring corroded thickness of ellipsoidal or torispherical heads the governing thickness may be:

Thickness of knuckle region - use appropriate head formula.

Thickness of central portion - use spherical shell formula.

SECTION 6 - INSPECTION AND TESTING OF PRESSURE VESSELS AND PRESSURE - RELIEVING DEVICES

6.1 GENERAL

Internal inspection is preferred. On-stream inspections may be acceptable in lieu of internal inspection for vessels under the specific circumstances defined in 6.4.

The inspection and inspection techniques must provide the information necessary to determine that all of the essential sections or components can safely operate until the next scheduled shutdown.

6.2 RISK - BASED INSPECTION (RBI)

A risk - based assessment includes evaluation of the liklihood and consequenses of failure. The liklihood assessment must be based on reasonable forms of degradation in the particular service. The inspection must be suitable for finding these forms of degradation. The liklihood of failure assessment should be repeated each time equipment or process changes are made that could affect degradation.

Other factors that should be considered in a RBI assessment include:

- Appropriatness of the materials of construction
- Design conditions relative to operating conditions
- Design codes and standards used
- Effectiveness of corrosion monitoring
- Quality of maintenance, and inspection QA/QC programs
- Equipment failure data
- Potential incidents such as fire, explosion, toxic exposure, environmental impact and health effects of a failure.

After a RBI assessment is conducted and documented, the results are used to establish a vessel inspection strategy defining the following:

- The most appropriate inspection methods, scope, tools and techniques based on the expected forms of degradation
- Appropriate frequencyfrequecy for internal, external and onstream inspections
- Need for pressure testing after damage, repairs or modifications
- Prevention and mitigation steps to reduce the liklihood and consequences of a failure

An RBI assessment may be used to increase or decrease the 10 year inspection limit of 6.4. When the limit is increased, The RBI assessment shall be reviewed and approved by a pressure vessel engineer and authorized inspector at intervals not greated than 10 years, or more often if warranted by changes.

6.3 EXTERNAL INSPECTION

Above ground vessels inspected, preferably in operation, at the lesser of 5 years or the same interval as the required internal inspection.

Inspection for corrosion under insulation considered for vessels operating between 25°F and 250°F or in intermittent service. Alternatively, internal shell thickness measurements at typical problem areas such as stiffening rings, around nozzles, etc., may be performed during internal inspection.

Buried vessels inspected based on corrosion-rate information from:

- Maintenance activity on adjacent piping of similar material.
- Buried corrosion test coupons.
- Representative portions of actual vessel or a vessel in similar circumstances .

Vessels with over 10 years of life remaining or that are protected against external corrosion do not need to have external insulation removed. Examples are vessels insulated to preclude the entrance of water (not the average insulation system!), jacketed cryogenic vessels, cold box vessels (surrounded by an inert gas), and vessels with sufficiently high or low temperatures to preclude the presence of water. However, the condition of the insulation system. Jacketing or cold box should be observed every 5 years.

6.4 INTERNAL AND ON-STREAM INSPECTION

Period between internal or on-stream inspections not to exceed lesser of:

1/2 remaining corrosion-rate life, or
10 years

When remaining safe operating life is less than 4 years, inspection interval may be the full life up to a maximum of 2 years
For vessels in non-continuous service and isolated from corrosive environments, the 10 years are actual service exposed life.

Internal inspection is the preferred method of inspection and shall be conducted on vessels subject to significant localized corrosion and other types of damage.

At the discretion of the authorized pressure vessel inspector, on-stream inspection may be substituted for internal inspection in the following situations:

a) When vessel entry is physically impossible;
b) When the general corrosion rate is known to be less than 0.005 in. per year and the remaining life is greater than 10 years and the following conditions are met:

1) Corrosive character of contents, including effect of trace components established for a five year period of same or comparable service;
2) No questionable condition is disclosed by external inspection specified in 4.2;
3) Operating temperature does not exceed lower temperature limits for material creep-rupture range;
4) Vessel not considered subject to environmental cracking or hydrogen damage from fluid handled;
5) Vessel is not strip-lined or plate-lined.

When on-stream inspection used it shall be performed using UT, RT, or other appropriate NDE to measure metal thickness.

A representative number of thickness measurements must be conducted on each vessel such as:

a) All major components (shells, heads, cone sections);
b) Representative sample of nozzles.

Number and location of measurements should consider previous inspection results and potential consequences of loss of containment.

Measurements intended to establish general and local corrosion rates in different vessel sections. Minimum number when the rate of corrosion is low and not localized. For localized corrosion, consult an expert in the mechanism and use methods that will reveal the scope and extent.

The remaining life of the vessel or vessel component shall be calculated per the following:

$$RL = \frac{t_{actual} - t_{minimum}}{CR}$$

$$CR = \frac{t_{previous} - t_{actual}}{\text{years between}}$$

Where,

tactual = the thickness measured at the time of inspection
tminimum = the minimum allowable thickness
tprevious = the thickness at the same location measured during a previous inspection.

Statistical analysis may be used in corrosion rate and remaining life calculations to extend the internal inspection interval.

For large vessels with zones having differing corrosion rates, each zone may be treated independently regarding inspection intervals.

As an alternative to calculating remaining life, projected maximum allowable working pressure of each component may be calculated.

If service conditions change, the period of operation until the next inspection shall be established for the new conditions.

When both ownership and location change, inspect before reuse and set new interval.

6.5 PRESSURE TEST

API authorized inspector determines the need for pressure test. Normally required after alterations (7.2.10). The test pressure shall be in accordance with the construction code used for determining the MAWP.

Two categories of test:

- Hydrostatic
- Pneumatic when hydrostatic testing is impracticable. Consider risks of compressed gas release and precautions required, including inspection precautions in ASME Code.

To minimize risk of brittle fracture temperatures should be maintained:

a) At least 30°F above MDMT for vessels more than 2 inches thick.

b) At least 10°F above MDMT for vessels 2 inches or less. Use the minimum acceptable operating temperature if a vessel does not have a MDMT.

Special handling of safety relief valves and other appurtenances is required .

6.6 PRESSURE-RELIEVING DEVICES

Tested and repaired by organizations that:

- Are experienced in valve maintenance.

- Have a fully documented quality control program having at least the 16 features listed and a fully documented training program.

Device Interval of inspection/testing determined by:

Performance in the particular service.

Should not exceed five years but may be increased to a maximum of 10 years for clean, non-corrosive service or for devices that have demonstrated satisfactory performance.

Where relieving device is found to be heavily fouled or inoperative, conduct a review of interval and cause.

6.7 RECORDS

Maintain permanent records throughout service life of vessel.

Progressive records are updated regularly to include new information pertinent to operation, inspection and maintenance history of vessel.

These records contain three types of vessel information pertinent to mechanical integrity:

1) Construction and design information such as Manufacturer's Data Reports and design calculations.

2) Operating and inspection history such as inspection reports and thickness measurements.

3) Repair, alteration, and rerating information such as repair and alteration forms, rerating documentation.

For pressure vessels that have no nameplate and minimal or no documentation;

1) Perform inspection and make any necessary repairs
2) Define design parameters and prepare drawings and calculations
3) Base calculations on applicable codes and vessel condition. Do not use a design factor of 3.5. See Section VIII Division 1 UG-10c for evaluation of unidentified materials. If UG-10c is not followed, use SA-283 Grade C allowable stresses for carbon steel and for alloy and nonferrous, use xray fluorescence to determine material type.
4) When radiography extent is not known, use a joint factor of 0.7 for butt welds or consider performing radiography if a higher joint factor is required.
5) Attach a nameplate or stamping showing the MAWP, maximum and minimum allowable temperatures and date.
6) Perform a pressure test as soon as practical as required by the design code.

SECTION 7 - REPAIRS, ALTERATIONS, AND RERATING OF PRESSURE VESSELS

7.1 GENERAL

Covers alterations and repairs by **welding.**

Repairs and alterations must follow the applicable requirements of the ASME Code, the code to which the vessel was built or other specific pressure vessel rating codes.

All proposed methods of execution, materials, and welding procedures for alterations or repairs to be approved by inspector and if necessary a pressure vessel engineer experienced in pressure vessel design, fabrication, or inspection.

7.1.1 Authorization

All alterations or repairs to be authorized by an inspector prior to work starting.

Alterations to all pressure vessels and repairs to Section VIII, Division 2 vessels must be approved by a pressure vessel engineer

The inspector will designate required fabrication approvals.

The inspector may give prior authorization for certain limited or routine repairs provided no pressure test will be required.

7.1.2 Approval

Inspector must approve all completed work to assure that it was satisfactorily completed.

7.1.3 Defect Repairs

Cracks may be repaired with authorization of inspector. For cracks at high stress areas, Inspector must consult with a *pressure vessel* engineer.

Repair procedure shall include appropriate NDE and inspection criteria.

7.2 WELDING

All welding must be in accordance with ASME Code Section IX and applicable requirements of Section VIII.

7.2.1 Procedures and Qualifications

Qualified procedures and welders shall meet requirements of Section IX.

7.2.2 Qualification Records

Repair organization shall maintain records of qualified procedures and welders and the records shall be available to the inspector prior to start of welding.

7.2.3 Preheat or Controlled Deposition Welding Methods as Alternatives to Postweld Heat Treatment

Preheating and controlled deposition welding may be used as alternative to PWHT where PWHT is inadvisable or mechanically unnecessary. Prior to using any alternative method, a metallurgical review conducted by a pressure vessel engineer shall be performed. This should consider factors such as the reason for the original PWHT, service susceptibility to stress corrosion cracking, hydrogen attack, creep, etc. and stresses in the weld. Materials are limited to those listed in 7.2.3.1 and 7.2.3.2.

7.2.3.1 Preheating Method (Notch Toughness Testing Not Required)

Materials are limited to P-No. 1 Group 1,2 and 3 and to P-No. 3 Group 1 and 2 (excluding Mn-Mo steels in Group 2).

Welding is limited to SMAW, GMAW and GTAW.

Preheat shall be 300°F and maximum interpass temperature shall be 600°F with requirements on distances from the weld.

7.2.3.2 Controlled-Deposition Welding Method (Notch Toughness Testing Required)

Materials are limited to P-Nos. 1, 3 and 4 and welding porocesses are limited as in 7.2.3.1.

A weld procedure specification shall be developed and qualified for each application. There are numerous welding requirements listed in 7.2.3.2.

The welding shall be a controlled-deposition temper-bead or half-bead technique.

7.2.4 Nondestructive Examination of Welds

Prior to and after welding, the area shall be MT or PT examined. If the vessel was originally required to be radiographed, new welds shall be radiographed. If this is not practical, perform UT.

7.2 5 Local Postweld Heat Treatment

Local PWHT may be substituted for 360 degree banding when specified precautions are taken and requirements met.

Application is reviewed and a procedure developed by pressure vessel engineers.

7.2.6 Repairs to Stainless Steel weld overlay and cladding

Repair procedure reviewed and endorsed by a pressure vessel engineer and authorized by the inspector.

Consideration given to factors that may modify the repair sequence including equipment in hydrogen service at elevated temperature.

Repairs monitored by inspector.

PT per Section VIII, Division 1, Appendix 8 after cooling to ambient temperature.

Vessels constructed of P-3, P-4 or P-5 base materials should be examined for cracking by UT per ASME Code Section V, Article 5, Paragraph T-543. This inspection conducted following a delay of 24 hours after completion of repair for equipment in hydrogen service, especially for chromium-molybdenum alloys.

7.2.7 Design

Butt joints shall have complete penetration and fusion.

Replacement Parts shall be fabricated per the requirements of the appropriate code.

New connections must be per the requirements of the appropriate code.

Fillet welded patches must be designed using the appropriate joint efficiency and this may not allow the use of a patch. Fillet welded patches are only for temporary repairs and then subject to acceptance by jurisdiction.

Fillet welded patches subject to approval by the authorized inspector and a pressure vessel engineer.

Overlay and flush patches shall have rounded corners.

Full encirclement lap bands and non-penetrating nozzles may be considered long term repairs if the requirements are followed.

7.2.8 Material

Must conform to Section VIII requirements, be weldable and have carbon content not exceeding 0.35 %.

7.2.9 Inspection

Acceptance criteria should include NDE techniques per original code or, when not practical, alternative NDE methods.

7. 2.10 Testing

Inspector determines need for pressure test.

Pressure test is normally required after an alteration.

Substituting NDE methods for pressure testing may be done only after jurisdictional approval, when required.

Substituting NDE methods for pressure testing after an alteration requires consultation with inspector and *pressure vessel* engineer.

Need for pressure test based on:

> Nature and extent of repairs or alterations.
>
> NDE examinations performed.
>
> *Level of brittle fracture risk during operation.*

7.2.11 Filler Metal

Filler metal for weld repairs should have minimum specified tensile strength ≥ that of base metal. If not:

> Consider compatibility of filler metal with base metal chemistry
>
> Repair thickness ≤ 50% base metal thickness, excluding CA
>
> Increase thickness of repair weld by ratio of tensile strengths base metal/filler metal
>
> Increased thickness of repair to have rounded corners and blended into base metal using 3:1 taper
>
> Repair made with minimum of two passes

7.3 RERATING

Change in temperature rating or MAWP.

Five basic requirements must be met.

Calculations by Manufacturer or owner-User pressure vessel engineer.

Established per construction code or latest ASME Code.

Current inspections records verify vessel is satisfactory for proposed service and corrosion allowance is appropriate.

Vessel has at some time been pressure tested per ASME Code to new service conditions or integrity maintained by special NDE. Otherwise, pressure test for rerated conditions.

Inspection and rerating acceptable to inspector

If the vessel was not designed to the higher allowable stresses of the Code 1999 Addenda and later, these higher stresses may be used in rerating if Figure7-1 is followed.

Nameplate stamping consists of :

Rerated by

MAWP______________________________psi

at________________°F. Date

SECTION 8- ALTERNATIVE RULES FOR NATURAL RESOURCE VESSELS

This section is not covered.

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

SHORT COURSE

ASME BOILER AND PRESSURE VESSEL CODE

SECTION VIII, DIVISION 1

PRESSURE VESSELS

GENERAL REQUIREMENTS AND FABRICATION

John L. Mooney, P.E.
Member ASME

ABSTRACT

This part of the presentation will concentrate on general requirements and fabrication and partly cover materials. Also included is the jurisdictional status of the Code and the responsibilities of Manufacturers, users and the Authorized Inspectors. The other part of the course covers design, organization of the ASME Boiler and Pressure Vessel Code Committee and materials properties related to design. The time available does not permit an in depth presentation of any part of the Code and can only cover the items considered to be of most interest to the average user. Specific areas of interest to those attending and questions will be handled in the open discussion periods. The course is directed at technicians, engineers and users with limited exposure to the Code and will be useful in designing, manufacturing, inspecting, operating and maintaining pressure vessels.

NOTE:
The written material and the comments of the speaker represent his personal opinions and interpretations of the Code.

COURSE INTRODUCTION

Code rules are intended to provide serviceable pressure vessels which protect life and property and to keep up with technical innovations to meet the needs of Manufacturers, users and regulatory bodies. To accomplish these objectives at a reasonable cost is a challenge, particularly in today's competitive environment.

The Code cannot include rules for all types of vessels or services. Section VIII, Division 1 covers a wide scope of pressure vessels from simple carbon steel water tanks and air receivers to those of nonferrous or high alloy or high strength materials and heavy wall or layered construction with no limit on pressure and high temperatures.

ASME Code construction is accepted in many countries. World-wide use of the Code became more prevalent with the issuance of Certificates of Authorization and Code Stamps in 1970 to foreign Manufacturers. The ASME and the National Board of Boiler and Pressure Vessel Inspectors are required to provide foreign Manufacturers the same consideration as US Manufacturers, with no restraint of trade. A requirement, until recently, was that foreign Manufacturers who use the ASME Stamp had to file the Manufacturer's Data Reports with the National Board. This provided evidence that a National Board commissioned Authorized Inspector had performed the required inspections.

The Code holds users, designers and Manufacturers responsible for meeting Code rules as well as meeting service needs. Users or their designated agents are responsible for establishing design requirements taking into consideration startup, normal operation, upset and shutdown conditions. User's agents may be their staff engineers, engineering organizations, consultants, manufacturers of systems for a specific service, organizations which sell or lease vessels for a specific service or engineers of the vessel manufacturer. The engineer acting as the designated agent does not have to be a registered professional engineer. What is essential is that he has experience in the design of pressure vessels and be familiar with the intended service. Design considerations shall include, but are not limited to the following;

- Maximum and minimum pressure and temperature
- Cyclic conditions
- Defining the contents, noting toxic contents for Code lethal contents requirements
- Materials and corrosion allowance
- Special materials requirements such as postweld heat treatment for corrosion resistance
- The installed position and the type of supports
- Nozzle locations plus external piping reactions
- The number and size of openings for pressure relief devices
- Nondestructive testing when not adequately covered by the Code
- External loads such as wind or seismic and those from vibrating machinery

The Manufacturer that holds the Code Certificate of Authorization and signs the U-1 Manufacturers Data Report assumes final responsibility for Code compliance. It is also the responsibility of the Manufacturer to obtain the design requirements from the owner/user.

The ASME Boiler and Pressure Vessel Code Stamp permits legal operation of the vessel in a jurisdiction that has a pressure vessel law. The stamp also provides useful information on the Manufacturer and construction of the vessel should operational changes, repairs or alteration be required.

ASME Code Section VIII, Division 2, Alternative Rules, may be used for pressure vessels where the Division 2 higher allowable stresses reduce costs, despite the more severe design, fabrication, inspection and testing requirements relative to Division 1. Division 2 requires that the User or his agent provide a Design Specification to the Manufacturer, certified by a professional engineer experienced in pressure vessel design.

ASME Code Section VIII, Division 3, Alternative Rules for High Pressure Vessels, was published for the first time in 1997. Division 3 is required for pressure vessels with design pressures above 10,000 psi but may be used for lower pressures.

SECTION VIII, DIVISION 1 HISTORY AND BACKGROUND

Section VIII of the Code was published in 1925 and was based on riveted construction and a theoretical safety factor of five based on material ultimate tensile strength. Some of the high alloy materials now in the Code have stress values based on yield, but these will be ignored in this discussion.

Welded vessels were permitted in the late 1930's. After their introduction and the requirements for radiographing welds, the safety factor varied from four to five, depending on the degree of radiography.

Prior to the 1950 edition of Section VIII, vessels were constructed and stamped U-68, U-69, U-70, U-200 or U-201. Allowable stresses were based on safety factors of five (U-68, 69, 70) or four (U-200 , 201).

U-68 vessels were the minimum for lethal service, service below -20F or design pressures over 600 psi. All welds were required to be radiographed butt welds. If the vessel was carbon steel, postweld heat treatment (PWHT) was mandatory. The basic joint efficiency was 90 percent but could be increased to 95 percent if the weld reinforcement was ground substantially flush.

U-69 vessels were limited to 600 psi and required main longitudinal and circumferential welds to be non-radiographed butt joints. The joint efficiency was 80 percent and this increased to 85 percent with PWHT. The 80 percent was revised to 85 percent without PWHT due to detrimental effects of PWHT on nonferrous and high alloy materials.

U-70 vessels were limited to 200 psi and 250F and constructed of low carbon steel. The stress values varied for butt, single lap welds, double lap welds or lap brazed joints.

U-68 vessels could be upgraded to U-200 vessels if butt welds were fully radiographed. U-69 vessels could be upgraded to U-201 vessels with butt welds spot radiographed.

Prior to the 1950 Code, an API-ASME Code also existed under the control of the American Petroleum Institute. This Code was not recognized by jurisdictions having pressure vessel laws and was used mainly by the petroleum, petrochemical and chemical industries located in jurisdictions that not have an unfired pressure vessel law. The API-ASME Code had a safety factor of four and sectioning or spot radiography of butt welds was a minimum requirement. The Codes were combined with the publishing of the ASME Section VIII 1950 edition.

The 1950 edition of the Code increased the allowable stress values in the carbon and low alloy steel tables to one-fourth of the ultimate tensile strength, which still applies.

Many other revisions have occurred but one of the revisions that affected the most vessels was the 1987 Addenda toughness revisions. Prior to this, carbon steel and low alloy vessels could be used to temperatures down to -20F with few restrictions. Since most brittle fractures occurred at warmer temperatures, the Code rules were revised. Brittle fractures may initiate at defects in areas of inadequate toughness and with tensile stresses exceeding 6,000 to 8,000 psi.

FOREWORD

The Code Foreword discusses Code objectives, general contents, the time limit when Code revisions become mandatory, Code Committee functions, The National Board of Boiler and Pressure Vessel Inspectors, the use of ASTM and ASME materials and other topics of general interest.

The Foreword also states what the Code is not or does not contain;

- The likelihood or consequences of deterioration in service
- A design handbook
- A replacement for education, experience and engineering judgment
- All tolerances
- Endorsement of proprietary or specific designs
- Rules for all types of construction. Those types of construction that are not addressed are not to be considered prohibited.

However, the Code has a wide variety of construction rules that, with proper selection, will provide a margin for deterioration in service and a useful life.

INTRODUCTION

U-1 SCOPE

The Scope outlines the kinds of pressure vessels that can be constructed under the rules and defines items that are not intended to be covered. However, the Scope states that Section VIII, Division 1 may be applied to vessels exempt from the Scope provided that the vessel meets all the requirements of the Division. It is the laws and regulations that exist at the point of installation that will dictate the need for Code construction and the appropriate Section of the Code. Some jurisdictions that have pressure vessel laws use the Code Scope to establish the vessels covered by the laws, while others vary from the Code Scope. Some jurisdictions only cover power boilers and there is one state that has no law. The Code Committee will not answer inquiries that describe a vessel or a service and ask whether Code construction is a requirement.

It is essential that users be familiar with the pressure vessel laws at the point of installation. Even if no pressure vessel law applies, it is recommended that the vessels within the Code scope be Code constructed due to safety, liability and the possibility that the vessel may be relocated.

U-1(c) allows the following items to be exempt from the scope of Division 1:

- Those within the scope of other sections such as boilers
- Fired process tubular heaters
- Machinery such as pumps, compressors, turbines, generators and engines
- Piping systems
- Piping components such as mixers, separators, snubbers, meters and strainers, provided that their pressure containing parts are generally regarded as piping components
- Vessels containing water limited to 300 psi and 210F
- Hot water supply storage tanks limited to 120 gal, 210F and a heat input of 200,000 Btu/h
- Vessels having an internal or external pressure not exceeding 15 psi
- Vessels having an inside diameter or diagonal not exceeding six inches
- Vessels for human occupancy

U-1(d) cautions the user and the designer that for pressure exceeding 3000 psi, deviations from and additions to the rules may be necessary. The rules do not limit pressures to 3000 psi.

When deviation from the rules is necessary and a pressure vessel law exists at the point of installation, the user or his design agent must petition the jurisdiction for a State Special Stamping. Examples of this could include high pressure vessels, use of materials not yet accepted, higher allowable stress values, design temperatures above the stress table limits and special shapes.

U-1(e)(1) defines the Code scope as terminating at the face of nozzle flanges, welding end connections, or sealing surfaces, and includes any pressure retaining opening covers and nonpressure parts welded to pressure parts. Nozzle extensions are not prohibited provided that the Code rules are satisfied. For example, a 90 degree piping elbow and a straight pipe section may be part of the Code nozzle providing that the Manufacturer's Data Report describes this as a nozzle extension.

U-1(e)(2) Where nonpressure parts are attached to pressure parts, the Code scope includes the nonpressure part, which may be a pad to which a non Code part could be attached.

U-1(g) permits the construction of unfired steam boilers (such as waste heat boilers) to Division 1. However, some jurisdictions require these to be constructed under Section 1, Power Boilers. This same comment applies to U-1(h) direct fired pressure vessels and U-1(i) gas fired jacketed steam vessels.

U-1(j) provides for "UM" stamping of vessels of sizes and/or pressures that are excluded by most jurisdictions. UM vessels may be used provided that radiography of welds is not a service requirement and that the vessel does not exceed 5 cubic ft at 250 psi, 3 cubic ft at 350 psi, or 1-1/2 cubic ft at 600 psi. UM vessels are typically mass produced vessels. They are not inspected during construction by Authorized Inspectors and a Manufacturer's Certificate of Compliance Form U-3 will be provided.

U-2 GENERAL

U-2(a) applies to engineered pressure vessels for process industries and not the common service vessels such as air receivers, LPG storage vessels, water tanks, etc. The user or his designated agent must establish the design requirements (see the Course Introduction).

U-2(b) states that the Manufacturer has the responsibility of complying with all the applicable parts of this Division. The Manufacturer may subcontract any part or service with parts fabricated by another ASME stamp holder being the responsibility of that manufacturer. ASME stamped parts with partial data reports will be the vessel Manufacturer's authorization to use the parts. The vessel Manufacturer's control of suppliers and subcontractors is important and will be detailed in the Manufacturer's Code approved quality control system

U-2(c) provides for design and construction using any combination of fabrication and material providing that the rules for each method and material are satisfied.

U-2(d) and (g) note that the rules do not cover all details of design and construction and permit tests (such as proof tests) to be used as well as details subject to the acceptance of the Inspector.

U-2(e) states that the Authorized Inspector shall make the inspections required by the rules plus any other inspections he deems necessary to permit him to certify that Code requirements were met. For the engineered vessel, the Inspector will be looking for input from the user or his design agent. The Inspector is not required to review design calculations but only to check that complete calculations prepared by qualified people exist.

U-2(h) applies to field assembly of vessels. Code responsibility cannot be divided except for procured stamped parts. The three types of field assembly are as follows:

- The Manufacturer extends his quality control system into the field and completes the vessel without subcontracting as noted below.
- The Manufacturer subcontracts the field assembly to another qualified stamp holder and obtains partial data reports for the work completed by that stamp holder. The partial data reports and the parts stamping on the vessel are the authorization for the prime Manufacturer to stamp the vessel.
- A qualified stamp holder in the field receives parts with stamping and partial data reports from the producing shop. The field stamp holder is the Manufacturer of the vessel.

U-3 and Table U-3 provide a ready reference to the standards mentioned in the Code and incorporate the standard's year. Each standard update has to be approved by Committee action and updates are not automatically accepted. Every effort is made to accept the updates and in some cases the standard has been revised to address Code comments.

Note: In the following discussion, not all Code paragraphs are discussed and many omitted paragraphs are covered in the other part of the presentation.

SUBSECTION A, GENERAL REQUIREMENTS

PART UG
GENERAL REQUIREMENTS FOR ALL METHODS OF CONSTRUCTION AND ALL MATERIALS

UG-1 SCOPE

The general requirements shall be used with the specific requirements of Subsections B (Fabrication) and C (Materials). The rules in these other subsections often supplement or override the general requirements.

MATERIALS

UG-4 GENERAL (Also covered in the other part of the course)
UG-5 TO 9 PLATE, FORGINGS, CASTINGS, PIPE AND TUBES, WELDING MATERIALS

Materials outside the limits of size and/or thickness in a material specification may be used but not those which exceed the limit in the Code stress tables or the applicable Code Case. Requirements to obtain approval of a new material are outlined in Section II Part D Appendix 5.

Materials may be dual marked with the material specifications and/or grades which the material meets. An example of this would be a plate material dual marked to grades 65 and 70 where the minimum ultimate tensile strengths are 65,000 and 70,000 psi. The material could be used at the higher strength levels of the 70,000 psi material but the maximum tensile strength will be limited by the range allowed by the 65,000 psi grade. Limiting the maximum tensile strength could be beneficial in avoiding cracking in certain services.

UG-8 states that pipes may be used for vessel shells and has materials, design, and test requirements for finned tubes.

Material test reports are only required by the Code for plates and all other forms are acceptable with material markings. For welding materials, trade name filler metals and weld materials covered by test reports are acceptable in addition to those covered in Section II, Part C providing that the welding procedure is qualified with the same materials. ASME materials begin with an SA or SB while ASTM materials begin with an A or B.

UG-19 SPECIAL CONSTRUCTIONS

This covers vessels with one or more pressure chambers. Each chamber must be considered as an independent vessel for design, fabrication and testing. The most restrictive design condition applies to parts that are a pressure barrier between two or more chambers, such as a tubesheet and tubes in a shell and tube heat exchanger. Each chamber must be tested independently without pressure in the other chamber. UG-99(e)(2) allows designing and testing the common barrier to only the differential pressure where the service is such that pressure must exist in both chambers. This design must be included in the Manufacturer's data report and should be included in the vessel stamping as a reminder to the installer and user.

Some additional types of multi-chambered vessels are jacketed vessels (Code Appendix 9), dimpled and embossed assemblies (Code Appendix 17), and vessels with half-pipe jackets (Code Appendix EE).

UG-20 DESIGN TEMPERATURE (Also covered in the other part of the course)

UG-20(f) allows impact test exemptions for low to medium strength carbon steels up to ½ inch or 1 inch thick, depending on toughness, at design temperatures down to -20F. A number of requirements must be satisfied for the exemption, including the standard Code hydrotest at 1.5 times maximum allowable working pressure (MAWP). Code Case 2046 for duplicate vessels and Code Case 2055 for engineered vessels permit pneumatic tests at 1.5 times the MAWP for a MAWP not exceeding 500 psi and thickness not exceeding ½ inch.

UG-24 CASTINGS

This paragraph and Code Appendix 7 cover the degree and types of examinations required and permissible defects. Quality factors of 80 to 100 percent are applied to the allowable stress values. Material purchase orders should provide the necessary examination and marking instructions. The quality factor applied to castings with welded joints shall be based on the lesser of the casting quality factor or the weld joint efficiency.

UG-25 CORROSION

General guidelines are included in Code Appendix E. The user must provide information to the Manufacturer of engineered pressure vessels.

Telltale holes, drilled partly through the vessel thickness from the outside, may be used to indicate when the thickness has been corroded to a predetermined thickness. These holes shall not be used with toxic or flammable contents.

UG-26 THROUGH UG-55 These are mostly design paragraphs covered in the other part of the course except for the following;

UG-35 OTHER TYPES OF CLOSURES

Quick-actuating closures must have interlocks, locking mechanisms or locking devices designed so that the failure of a locking element cannot result in the release of the contents under pressure. The design must prevent opening the closure while under pressure and the user should review the device and provide input. They may be used for handholes, manholes or full end closures for loading bulk products. Poor designs of closures and interlock devices, inadequate maintenance and improperly trained operators have been the causes of serious failures.

UG-43 METHODS OF ATTACHMENT OF PIPE AND NOZZLE NECKS TO VESSEL WALLS

These include brazing, welding, studding, threading and expanding. Studding connections shall have a flat surface machined on the shell, a built-up pad, a plate or a fitting. Drilled holes shall not penetrate to within one-fourth of the vessel wall thickness. Expanded connections are accomplished by inserting a pipe or tube into a vessel opening and expanding it into the shell. There are limitations on and requirements for threaded and expanded connections.

UG-45 NOZZLE NECK THICKNESS

This shall not be less than the thickness computed from the applicable loadings plus corrosion allowance or the minimum thickness (87.5 % of nominal thickness) of standard wall pipe plus corrosion allowance.

UG-46 INSPECTION OPENINGS

The types and size requirements for inspection openings are determined by the vessel diameter. Certain vessels do not require inspection openings. The openings may be fitted with threaded fittings, handhole and manhole closures and/or removable covers. Removable parts or piping components on nozzles may substitute for dedicated inspection openings.

FABRICATION

UG-76 CUTTING PLATES AND OTHER STOCK

Material may be separated by machining, shearing, grinding or by oxygen or arc cutting. If gas or arc cut, the slag and discoloration must be removed by machining, grinding or chipping. Gas or arc cutting are not normally detrimental to most materials but care must be used on high alloy or high strength materials.

UG-77 MATERIAL IDENTIFICATION

Material marking must be such that all material in the completed vessel may be identified (also see UG-93 and UG-94).

UG-78 REPAIR OF DEFECTS IN MATERIALS

Repairs to materials by the material manufacturer or the vessel Manufacturer are permitted with the acceptance of the Authorized Inspector. The material manufacturer should consult with the vessel Manufacturer before making repairs and the user may require that his approval be obtained for any or major material repairs. The repair should be mapped and retained for reference by the vessel Manufacturer and the user.

UG-79 FORMING SHELL SECTIONS AND HEADS
UG-80 PERMISSABLE OUT-OF-ROUNDNESS OF CYLINDRICAL, CONICAL AND SPHERICAL HEADS
UG-81 TOLERANCE FOR FORMED HEADS

These are the basic Code tolerances in addition to those at welded joints. Users should consider specifying other tolerances such as for nozzle locations, etc. The tolerances for external pressure are more severe than for internal pressure since buckling strength may be reduced by irregularities while tensile stresses from internal pressure tend to round out components. An important step to avoid a flat area in rolled shells is preliminary forming of the longitudinal plate edges.

UG-84 CHARPY IMPACT TESTS

Toughness requirements for base materials, weld materials and heat affected zones are listed. Fig. UG-84.1 specifies the impact values for carbon and low allow steels with specified minimum tensile strengths of less than 95,000 psi. Higher strength and thicker materials require additional toughness to avoid brittle fracture and this is indicated in the Figure. Table UG-84.2 provides for impact test temperature reduction for subsize specimens when the material is too thin for a standard 0.4 x 0.4 inch charpy specimen . Table UG-84.3 lists the material specifications for impact tests for the various product forms. Table UG-84.4 allows impact tests at warmer temperatures than the minimum design metal temperature for lower strength steels. UG-84(i) covers vessel (production) impact test plates which are made from one of the heats of steel used for the vessel. These test plates are intended to represent as nearly as practicable the quality and type of welding in the vessel joints. Exemptions from impact testing for carbon and low alloy steels are in UG-20 and UCS-66.

UG-85 HEAT TREATMENT

Material heat treatment, such as normalizing to improve material properties, may be performed by other than the material manufacturer. The user should be informed.

INSPECTION AND TEST

UG-90 GENERAL

Documentation and test requirements for the Manufacturer and verification and inspection requirements are summarized for the Inspector. Users may use this list to select what they or their inspectors will review or inspect.

The Authorized Inspector need only inspect the vessel for requirements such as material or dimensional defects, marking, stamping and hydrotest. However, he must verify a list of items indicating that the manufacturer's quality control system is in Code compliance.

UG-91 THE INSPECTOR

The Authorized Inspector shall be from the inspection organization of a state, municipality or Canadian province, an insurance company writing boiler/vessel insurance or the owner/user.

UG-93 INSPECTION OF MATERIALS
UG-94 MARKING ON MATERIAL
UG-95 EXAMINATION OF SURFACES DURING FABRICATION
UG-96 DIMENSIONAL CHECK OF COMPONENT PARTS
UG-97 INSPECTION DURING FABRICATION

The Manufacturer is responsible to make all dimensional checks. The Authorized Inspector may make selected checks and satisfy himself that the Manufacturer made and documented the dimensional checks. The AI will probably not make detailed dimensional checks.

Material inspections are not required on receipt but are required prior to and during fabrication. These are covered in the quality control system.

UG-93(d)(3) requires pressure corner joints to have the weld preparation and final weld in flat plates thicker than ½ inch magnetic particle or dye penetrant examined.

UG-98 MAXIMUM ALLOWABLE WORKING PRESSURE

The terms maximum allowable working pressure (MAWP), design pressure, operating pressure and working pressure are all used in the Code. Operating pressure and working pressure are the same and they relate to the pressure in service and will be lower than design pressure or MAWP. MAWP is that pressure permissible at the top of the vessel in its normal position at the operating temperature specified for that pressure. There must be a margin between operating pressure and MAWP for practical reasons such as to prevent the pressure relief valves from leaking or from opening during small upsets. Component design pressure may vary from MAWP due to liquid head. For example, in a partly liquid filled vessel the design pressure and MAWP may be similar at the top of the vessel while the design pressure for the components in the liquid area will be higher than the MAWP. The MAWP will be stamped on the vessel and be used to establish the setting for the pressure relief device.

When the design pressure is specified by a vessel purchaser it denotes the pressure at the top of the vessel. The Manufacturer will incorporate any liquid head in the calculations and choose vessel component thickness, which may be thicker than required due to ordering a standard thickness. If the Manufacturer then recalculates the MAWP for each component, the lowest of these MAWPs establishes the vessel MAWP (which may then exceed the originally specified design pressure).

UG-99 STANDARD HYDROSTATIC TEST (Also covered in the other part of the course)

Since the pressure test is the first time the vessel will be pressurized, the risk of brittle fracture must be considered. The Code recommends that the metal temperature during test be 30F above the minimum design metal temperature. Only the people necessary for the operation should be in the area and they should remain in protected areas until the pressure is reduced to that for the visual inspection (a minimum of 2/3 of the test pressure). Pressure vessels have failed in a brittle manner before the maximum hydrotest pressure was reached. The benefits of the pressure test include a tendency to blunt notches, cracks and stress raisers.

If it is uneconomical to design a tall vertical vessel for hydrotest, it may be possible to place it horizontally for the test. Test water purity is a consideration for materials such as stainless steel, which may be attacked by chlorides.

UG-100 PNEUMATIC TEST

Comments similar to those for hydrotest apply to personnel safety and brittle fracture during a pneumatic test. The explosive force of the compressed air can do even more damage than a failure during a hydrotest. Therefore, the Code makes it mandatory to add 30F to the MDMT for the test temperature. Except for a few Code Cases, pneumatic testing may be substituted for hydrostatic testing only if traces of moisture cannot be tolerated in the completed vessel or the vessel is not designed and supported to be filled with water.

UG-101 PROOF TESTS TO ESTABLISH A MAWP

This test is usually conducted where there is no design formula or the shape of the part makes analysis difficult. With today's computer programs, proof testing is becoming less common. The tests are limited to those based on yield or burst and four types are permitted:

- Brittle coating
- Test to Burst
- Strain measurement
- Displacement measurement

A test procedure should be accepted by the Authorized Inspector before proceeding and he must certify that he witnessed the test and accepted the results. The documentation must be retained for proof tested parts.

UG-103 NONDESTRUCTIVE TESTING

Where magnetic particle and liquid penetrant examinations are required, they shall be performed per Appendices 6 and 8. Division 1 does not require formal certification of technicians for these inspections, but many manufacturer's technicians will be certified.

UG-116 REQUIRED MARKINGS
UG-118 METHODS OF MARKING
UG-119 NAMEPLATES

The marking on the vessel includes the official Code U Symbol (for Unfired) or UM Symbol (mass produced), the Manufacturer and Manufacturer's serial number, The MAWP at its temperature, the MDMT at its pressure, and the year built. The type of construction used (welded, etc.), special service (lethal, etc.), and the degree of radiography and whether the vessel was PWHT are also indicated..

Many jurisdictions with pressure vessel laws require that the vessel be registered with the National Board and that stamping include a National Board registration number, usually located across the top of the stamping. Nameplates are encouraged rather than die stamping the vessel wall.

UG-117 CERTIFICATES OF AUTHORIZATION AND CODE SYMBOL STAMPS

Prior to a Manufacturer receiving an ASME Certificate of Authorization and Code stamp, he must have a written quality control system and be able to demonstrate the capability to implement the system in his plant or at a field construction site. The quality control system will be reviewed and recommended for acceptance by an Authorized Inspector and possibly others if the jurisdiction has a pressure vessel law. The minimum quality control system elements are in Code Appendix 10.

UG-120 DATA REPORTS

If the vessel is registered with the National Board, the Manufacturer's data report will be sent to the National Board. This is a good source of information if vessel data is lost. Refer to Code Appendix W for data reports and other forms.

UG-125 THROUGH 137 PRESSURE RELIEF VALVES

The Code includes requirements for materials, design, lift and flow certification and stamping. Vessel pressure is permitted to rise from 10 to 21 percent above the MAWP, depending on whether there are single or multiple pressure relieving devices or vessel exposure to fire. The Manufacturer must provide the proper opening for the device or note on the Manufacturer's data report that the device is installed external to the vessel, as on piping. It is the user's responsibility to determine the proper size of the device and install and maintain it. Elimination of pressure relief devices because the service or source of pressure cannot exceed the vessel MAWP is permitted by Code Case 2211. Code Appendix 11 covers Capacity Conversions for Safety Valves.

SUBSECTION B, REQUIREMENTS PERTAINING TO METHODS OF FABRICATION

PART UW
REQUIREMENTS FOR PRESSURE VESSELS FABRICATED BY WELDING

GENERAL

UW-3 WELDED JOINT CATEGORY

UW-3 and UW-12 (Joint Efficiencies) will be discussed before UW-2 (Service Restrictions) since the latter involves joint categories and types of joints. Joint categories are assigned by the joint locations in the vessel. The detailed description of categories in the Code should be understood. However, for a simplified discussion, consider the weld categories as: A longitudinal joints and joints between hemispherical heads and shells; B circumferential joints; C joints at flanges, flat heads and tubesheets; D nozzle-to-shell joints. Refer to the illustrations in Course Appendix II. The categories are used in Division 1 to specify joint types (butt, etc.) and degrees of inspection.

DESIGN

UW-11 RADIOGRAPHIC AND ULTRASONIC EXAMINATION
UW-12 JOINT EFFICIENCIES

These paragraphs establish the requirements for weld examination and the joint efficiencies (E) based on the type and degree of examination. Table UW-12 has three degrees of examination: full radiography, spot radiography and none, with corresponding joint efficiencies of 1, 0.85 and 0.7 for type number 1 butt joints. Full radiography is required for certain thicknesses of various materials and this is covered later. Nineteen examples of joint efficiencies are included in the Course Appendix VII with examples 5, 9, 12 and 19 as exercises. Code Appendix L provides flow charts and sample problems on joint efficiencies. The Code permits different joint efficiencies to be applied to different parts of the same vessel.

UW-11(a)(5)(b) requires that vessels designed under Table UW-12 column (a), full radiography, have a minimum of spot radiography of category B or C butt welds.

For seamless vessel sections, ellipsoidal and torispherical heads or pipe, UW-12 allows E=1 with spot radiography and E=0.85 with no radiography of the connecting welds. For seamless hemispherical heads, the joint between the head and the shell is a category A joint and the joint efficiency from Table UW-12 must be used in calculating the head thickness

UW-2 SERVICE RESTRICTIONS

The services that require special design, fabrication, heat treatment and inspection are:

- Lethal contents
- Low temperature
- Unfired steam boilers
- Fired pressure vessels

For many of the various joint categories in these services, joint types per Table UW-12 and special inspection are specified as well as full penetration welds for Category D (nozzle-to-shell) joints. Refer to Fig. UW-16.1 for full and partial penetration nozzle welds. A user may also specify similar restrictions for fatigue service, high pressure, etc. There are also joint type restrictions in Part UHT.

UW-5 GENERAL

Nonpressure part material which cannot be fully identified may be used if it is proven to be of weldable quality.

UW-9 DESIGN OF WELDED JOINTS

Welding grooves shall be such as to permit complete penetration and fusion. Tapered transitions shall have a length not less than three times the offset between abutting sections of different thicknesses. Longitudinal joints of adjacent courses shall be staggered except when radiographed for 4 inches each side of the circumferential weld intersections.

UW-13 ATTACHMENT DETAILS

The Figures referenced by this paragraph are useful in understanding acceptable head to shell and corner joints and should be consulted when any unusual joints are proposed.

UW-14 OPENINGS IN OR ADJACENT TO WELDS

Adequately reinforced openings may be located in welded joints. Small unreinforced openings may be located in certain welded joints provided these joints are radiographed. If the joint is not radiographed, there is a minimum spacing from the weld to the opening.

UW-15 WELDED CONNECTIONS

External nozzle reinforcing plates and saddles shall be provided with a telltale hole that may be used for a compressed air and soapsuds tightness test of welds underneath the reinforcement.

UW-16 MINIMUM REQUIREMENTS FOR ATTACHMENT WELDS AT OPENINGS

These category D welds may be full or partial penetration or fillet welds as limited by other parts of the Code. Figs. UW-16.1 and 16.2 show many acceptable attachment welds. Fittings not exceeding NPS 3 have some exemptions from weld size requirements.

UW-17 PLUG WELDS

Plug welds are welds made in holes in a part thus joining it to another overlapping part. They may be used in lap joints, in reinforcement around openings and in nonpressure structural attachments. Design criteria are given, including a limit of 30 percent of the load to be transmitted.

UW-20 TUBE-TO TUBESHEET WELDS

Design of strength welds, which develop the full strength of the tube, is specified. Seal welds are also permitted to ensure leak tightness of expanded tube joints.

FABRICATION

UW-26 GENERAL
UW-27 WELDING PROCESSES

UW-26 through UW-49 cover welding requirements. An accepted welding process must be used and the welding procedure and the welder or welding operator must be qualified to the requirements of Section IX, Welding and Brazing Qualifications. The types of welding processes used most frequently in pressure vessel welding are shielded metal arc, submerged arc, gas metal arc, gas tungsten arc, electrogas and, for heavy walls, electroslag. General guidelines on preheating are found in Code Appendix R

UW-28 QUALIFICATION OF WELDING PROCEDURE
UW-29 TESTS OF WELDERS AND WELDING OPERATORS

Qualification is required for pressure part welds, welds joining pressure parts to load bearing nonpressure parts, nonautomatic, nonloaded attachment welds, and tack welds. Welding of test plates to qualify a procedure must be by the manufacturer's welder.

UW-30 LOWEST PERMISSABLE TEMPERATURES FOR WELDING

Do not weld at base metal temperatures lower than 0°F. Preheat at temperatures between 0 and 32°F. Shielding is required for inclement weather.

UW-31 CUTTING, FITTING, AND ALIGNMENT
UW-33 ALIGNMENT TOLERANCES

Table UW-33 has joint alignment tolerances. If the joint is not fit up right, cut it apart and reweld it. Repeated requests of the Code Committee to permit a Section III discontinuity analysis have been denied. Category A joints have higher stresses and therefore tighter tolerances than for other joints.

UW-32 CLEANING OF SURFACES TO BE WELDED

Proper cleaning is essential with most welding procedures but may be waived where welding processes produce quality welds without surface cleaning.

UW-35 FINISHED LONGITUDINAL AND CIRCUMFERENTIAL JOINTS

The Code words prohibiting "coarse ripples, grooves overlaps, and abrupt ridges and valleys" are open to interpretation and may result in disagreements during and after fabrication. An inspector that is overly severe can insist on repairs that do not increase the safety or serviceability of the vessel. However, rough welds can mask defects which would otherwise be clear on the radiographs. If surface appearance is substandard, correct this before radiography. If the Manufacturer has workmanship standards (photographs or wax molds), they may be accepted by the Authorized Inspector and the user.

The word "undercut" has been replaced in Division 1 by "reduction in thickness adjacent to the weld." Some degree of this is inherent at weld edges. The reduction in thickness cannot reduce the thickness to below design thickness, and this may require an increase in the thickness ordered, particularly for minimum thickness vessels. The reduction in thickness is also limited to 1/32 inch or 10 percent of the base metal, whichever is less.

The table in UW-35 specifies maximum weld reinforcement (additional weld metal above the base plate thickness), mainly to minimize the thickness difference between weld and base plate for radiography. The reinforcement permitted for circumferential joints is higher than that for other joints since circumferential joints have lower stresses and are therefore somewhat less critical than longitudinal joints.

UW-38 REPAIR OF WELD DEFECTS

Cracks, pinholes and incomplete fusion require removal and repair.

UW-39 PEENING

Peening means hammering the weld and this is infrequently used to control distortion, relieve residual stresses or improve the quality of the weld. There are restrictions on peening the initial or final weld passes.

UW-40 PROCEDURES FOR POSTWELD HEAT TREATMENT

Postweld heat treatment (PWHT) is beneficial in reducing welding residual stresses and hardness in the weld and adjacent base metal. The most common methods of PWHT are:

- Heating the entire vessel in a furnace
- Heating circumferential welds or circumferential bands including nozzles by electrical resistance coils. This is also used for welds between sections of vessels that are too long for the furnace.
- For field constructed vessels, placing a burner in the vessel or ducting burner exhaust to the vessel.

Control of the vessel temperature by means of thermocouples is important and adequate thermal insulation is required to limit thermal gradients in the latter two above methods. Nominal thickness for PWHT is defined. For butt welds, nominal thickness is the thickness of the weld and for parts of unequal thickness, the thinner of the parts joined. Code requirements should be reviewed.

UW-42 SURFACE WELD METAL BUILDUP

Weld buildup may be used to restore metal thickness and to obtain the required transition between thick and thin sections. Tapering requires magnetic particle or dye penetrant examination and attention to both radiographic technique and penetrameters due to the thickness change. Restoring metal thickness is generally classified as a repair and the inspectors should be notified.

INSPECTION AND TEST

UW-50 NDE OF WELDS ON PNEUMATICALLY TESTED VESSELS

Dye penetrant or magnetic particle examination is required of welds around openings and attachment welds having a throat thickness > ¼ inch.

UW-51 RADIOGRAPHIC AND RADIOSCOPIC EXAMINATION OF WELDED JOINTS
UW-53 TECHNIQUE FOR ULTRASONIC EXAMINATION OF JOINTS

Defect acceptance criteria are presented for elongated and aligned indications and Code Appendix 4 is referenced for rounded indications. Cracks or lack of penetration are unacceptable. Section V, Article 2, is referenced as a guide for technique. Written procedures are not mandatory but procedures for the guidance of the radiographer are common and certification is required. The acceptance of a radiograph is based on meeting the density requirements by showing the designated hole or wire of the penetrameter. The manufacturer is not required to retain the radiographs and the user should consider obtaining them for heavy wall vessels.

Real time radioscopic examination shall be per Appendix II of Article 2 of Section V. This is used by some manufacturers of multiple duplicate vessels.

Ultrasonic examination per Division 1 Appendix 12 may be used as a substitute for radiography where meaningful radiographs cannot be obtained (some nozzle-to-shell welds) or for repaired welds (UW-51(b)). Ultrasonic examination is also required for electroslag and electron beam welding. Technician qualification is required.

UW-52 SPOT EXAMINATION OF WELDED JOINTS

Spot radiography of approximately one radiograph per 50 ft of weld provides a quality level in between full radiography and visual inspection. Users need to review the Note to UW-52 which discusses the philosophy of spot radiography. Spot radiography will indicate any serious welding procedure problems and makes the welder aware that serious mistakes will show up. For each spot radiograph where the welding is rejected, two additional spot radiographs are required in the same weld. It is desirable for the inspector to choose the radiograph locations as he may pick more severe locations than the Manufacturer.

PART UF
REQUIREMENTS FOR PRESSURE VESSELS FABRICATED BY FORGING

Most of Part UF follows Part UW and the other applicable parts of the Code. Part UF should be reviewed in detail for vessels using forgings for shell sections.

PART UB
REQUIREMENTS FOR PRESSURE VESSELS FABRICATED BY BRAZING

This is metal joining by heating to above 800F and filling between closely fitted joint surfaces with a nonferrous filler metal.

SUBSECTION C, REQUIREMENTS PERTAINING TO CLASSES OF MATERIALS

PART UCS
REQUIREMENTS FOR PRESSURE VESSELS
CONSTRUCTED OF CARBON AND LOW ALLOY STEEL

UCS-1 SCOPE

Except when specifically prohibited, Part UCS materials may be used with other Code materials in a vessel.

MATERIALS

UCS-6 STEEL PLATES

Structural steel plates may be used but are limited to SA-36 and SA-283 under the following conditions:

- Vessels are not used to contain lethal substances
- Not permitted for unfired steam boilers
- Shell, head and nozzle thickness is limited to 5/8 inch

DESIGN

UCS-27 SHELLS MADE FROM PIPE

Seamless and electrical resistance welded pipe (maximum diameter 30 inches) may be used for shells and nozzles. Refer to UW-12(d) and (e) for joint efficiencies.

UCS-56 REQUIREMENTS FOR POSTWELD HEAT TREATMENT

Manufacturers must be familiar with the rules for PWHT in each material section of the Code to avoid the many problems that may occur. Section II Part D provides the P numbers which categorize the steels for PWHT and other criteria. The notes in Table UCS-56 for each P number establish the thickness above which PWHT must be performed. For example, P-No. 1 carbon steel does not require PWHT up to 1-1/4 inch thick and this may be extended to 1-1/2 inch with a weld preheat of 200F minimum. UW-2 and UCS-79 may require PWHT in all thicknesses because of service environment or forming operations. The user or designer may specify PWHT due to the vessel contents or environment. Table UCS-56.1 allows reductions in PWHT temperatures with increased PWHT time. These reductions may result in inadequate PWHT and must be carefully considered before application, which usually involves a reason why the normal PWHT temperature would be detrimental.

Preheat (see Appendix R) is also frequently beneficial and is required for welded repairs in order to waive PWHT, as is MT or PT examination of the groove after defect removal. The temper bead technique is mentioned for P-No. 3 repairs. This consists of depositing thin weld layers (1/8 to 5/32 in. maximum diameter electrodes) and grinding off ½ the weld layer thickness.

UCS-57 RADIOGRAPHIC EXAMINATION

Table UCS-57 has thickness limits above which radiography of butt welds is mandatory. The limit varies from 1-1/4 inch for P-No. 1 mainly carbon steel materials to zero for P-No. 5 materials (e.g. 2-1/4, 5 and 9 Cr). Where UCS-57 is mandatory, UCS-19 requires Category A & B butt joints to be Type 1 (double butt) or Type 2 (single butt with backing strip). Table UCS-57 takes precedence over UW-11 and UW-12 except for the exemptions for Category B and C joints in nozzles and communicating chambers that neither exceed 10 inch nominal pipe size nor 1-1/8 inch wall thickness.

LOW TEMPERATURE OPERATION

UCS-66 MATERIALS

Figure UCS-66 establishes impact test exemptions using minimum design metal temperatures (MDMT's) and governing thickness (Par. UCS-66). Per (UG-20(b)), MDMT must consider low temperatures during operation, upsets, autorefrigeration and pressure test, including the effect of low ambient temperatures. For example, if the vessel is fully pressurized during startup at ambient temperature and other conditions are at higher temperatures, the MDMT could be set at the lowest one day mean ambient temperature or lower. If the vessel contents are volatile and subject to autorefrigeration (chilling during depressuring), this must be considered. Fig. UCS-66.1 allows lowering of impact test exemption temperature at lower than Code allowable stresses. For example, if the vessel metal reached -45F at a depressured level of 40 percent of allowable stress, the UCS-66.1 temperature credit might be such that -45F could be the MDMT and lower autorefrigeration temperatures would not have to be considered. This is provided that the vessel be warmed to above -45F before repressuring. Fig. UCS-66.2 is a flow diagram of the rules and Code Appendix L (Par. L-9) and the last pages of this Course Appendix VII provide examples.

UCS-67 IMPACT TESTS OF WELDING PROCEDURES

Welds and heat affected zones shall be impact tested when the base metal is required to be impact tested and as noted where the base metal is exempt. Testing when the base metal is exempt is intended to insure that the welding procedure will not reduce toughness to unacceptable levels.

UCS-68 DESIGN

When the MDMT is < -50F and the stresses are substantial (above the 0.4 ratio of Fig. UCS 66.1);

- Category A and B joint types are specified and Category C and D joints must be full penetration
- PWHT is required

If PWHT is performed when it is not a requirement of Division 1, a 30F reduction in impact test exemption temperature may be taken for P-No, 1 materials. The Code recognizes the benefits of PWHT in lowering welding residual stresses and hardnesses and thus reducing the risk of brittle fracture.

FABRICATION

UCS-79 FORMING SHELL SECTIONS AND HEADS

Cold or hot forming is permitted except cold forming by blows is not permitted. Heat treatment after cold forming is required if extreme fiber elongation is more than 5 percent from the as-rolled condition and any of the following conditions exist:

- Lethal service
- Material impact testing is required
- Thickness exceeds 5/8 inch before forming
- Forming reduction in thickness exceeds 10 percent
- Material temperature during forming is 250 to 900F

P-No. 1 Groups 1 and 2 materials do not require heat treatment with up to a 40 percent extreme fiber elongation, determined by the formulas, if none of the above conditions exist. Suppliers of cold formed parts shall provide certification if they performed the required heat treatment.

UCS-85 HEAT TREATMENT OF TEST SPECIMENS

The vessel Manufacturer must provide the material manufacturer data on the heat treatment to be performed during fabrication, with the Code noted exceptions. This permits the material manufacturer to simulate the material time at temperature so that the material specification tests will represent the fabricated vessel. The specimen time at temperature must be a minimum of 80 percent of that for the vessel and users frequently specify additional time to allow for field repairs and alterations.

NONMANDATORY APPENDIX CS

Properties such as creep or rupture strength are influenced by steel processing variables such as grain size, heat treatment, deoxidation and residual elements.

PART UNF
REQUIREMENTS FOR PRESSURE VESSELS
CONSTRUCTED OF NONFERROUS MATERIALS

GENERAL

Nonferrous materials are used to resist corrosion, for easy cleaning (e. g. the food industry) for strength and scaling resistance at high temperatures and for toughness at low temperatures. Specific chemical compositions, heat treatments, fabrication requirements and supplementary tests may be required. Typical vessel or part materials are aluminum, copper, brass, bronze, copper nickel (Monel), and nickel alloys. Titanium and zirconium are also covered.

UNF-23 MAXIMUM ALLOWABLE STRESS VALUES

Allowable stress in the annealed condition must be used for welded or brazed joints of material having increased tensile strength due to cold working or heat treatment, with noted exceptions.

This can apply to aluminum, copper and its alloys, nickel-chromium iron and nickel-iron chromium.

UNF-56 POSTWELD HEAT TREATMENT

This is not normally necessary or desirable for nonferrous materials (see the exceptions) and usually involves an agreement between the user and Manufacturer.

UNF-57 RADIOGRAPHIC EXAMINATION

Because of the nonferrous materials costs, thicknesses above the minimum will usually involve full radiography of butt welds. Subparagraphs (b) and (c) specify radiography for certain materials.

UNF-58 LIQUID PENETRANT EXAMINATION

Liquid penetrant examination of certain welds or welds that are not fully radiographed is required.

UNF-65 LOW TEMPERATURE OPERATION

Exemptions from impact tests are permitted when welds are of the same range of composition as the base material. Wrought aluminum is exempt down to -452F; copper and nickel and their alloys and cast aluminum down to -325F; and titanium down to -75F. Lower temperatures are permitted if the user has suitable test data.

APPENDIX NF CHARACTERISTICS OF THE NONFERROUS MATERIALS

This is a general discussion of properties and fabrication and should be read if you haven't worked with these materials. Special comments on aluminum include a warning on seizing of aluminum and aluminum alloy threaded connections. Strain hardened aluminum alloy bolting is has more resistance to seizing.

PART UHA
REQUIREMENTS FOR PRESSURE VESSELS CONSTRUCTED OF HIGH ALLOY STEEL

GENERAL

The use of high alloy materials such as stainless steel involves the same factors as those for nonferrous materials and the user and designer should interface to review past operating experience.

UHA-32 REQUIREMENTS FOR POSTWELD HEAT TREATMENT

Postweld heat treatment is usually not required for austenitic stainless steels (300 series) but may be required for some of the ferritic stainless steels (400 series).

UHA-33 RADIOGRAPHIC REQUIREMENTS

Because of material cost, thicknesses above the minimum will usually involve full radiography. Some ferritic stainless steels require radiography while austenitic stainless steels follow the general rules. In most cases, the radiography has to be performed after any PWHT.

UHA-34 LIQUID PENETRANT EXAMINATION

This is mandatory for all austenitic chromium-nickel alloy steel welds and all austenitic/ferritic duplex butt and fillet welds when the shell thickness exceeds ¾ inch. Examination shall be made after any PWHT.

UHA-42 WELD METAL COMPOSITION

It is recommended that the weld metal properties match the base material. However, a dissimilar weld deposit may be specified, (particularly for corrosion resistance), if the procedure is qualified and the manufacturer and user are satisfied that the weld is suitable for the service.

UHA-51 IMPACT TESTS

High alloy toughness criteria is "lateral expansion opposite the notch," which refers to the deformation opposite the notch of the charpy impact test specimen. Impact tests are required when certain thermal treatments are performed as brittleness may result. Exemption from impact testing for some common stainless steels and their heat affected zones (304/304L, 316/316L, 321 and 347) extends to -425°F. Other austenitic stainlesses with carbon content less than 0.1 percent are exempt down to -320°F and those with higher carbon are exempt to -55°F. Even if the base metal is exempt, impact tests may be required for the weld metal and HAZ. Most austenitic stainless steel weld metal is exempt to -155°F.

UHA-52 WELD TEST PLATES

For type 405 material (12 chrome) which is not PWHT, welded test plates shall be made from each melt of plate steel used in the vessel

APPENDIX HA, SUGGESTIONS ON THE SELECTION AND TRATMENT OF AUSTENITIC CHROMIUM-NICKEL AND FERRITIC AND MARTENSITIC HIGH-CHROMIUM STEELS

This contains general information on stainless steel attack mechanisms, fabrication and some of the factors that the user and designer should consider in selecting materials. The type of structure and the thermal and mechanical treatment are determining factors in the material's resistance to intergranular, stress corrosion, and other cracking and its ductility and toughness.
Austenitic steels held for a sufficient time between 800 F and 1600F may be sensitized to intergranular corrosion. Methods of combating this include annealing/rapid cooling, stabilizing with columbium, titanium or tantalum and use of steels with low carbon contents. Austenitic chromium-nickel steels that are highly stressed in tension may crack when exposed to certain corrosives (stress corrosion cracking). A sigma phase (formed between 1050F and 1700F) may significantly reduce the ductility and toughness of stainless steels. With chrome exceeding 12%, maximum embrittlement occurs at 885F. As may be seen, this is a complex subject.

PART UCI
REQUIREMENTS FOR PRESSURE VESSELS CONSTRUCTED OF CAST IRON
GENERAL

This does not apply to all cast irons, as Part UCD covers ductile cast iron. The requirements in Subsection A of the Code covering castings also apply. Cast iron is used for some channel sections and heads in heat exchangers and for pressure parts of odd shapes. The shape may require an analysis per U-2 or a proof test per UG-101. Table UCL-23 incorporates a theoretical safety factor of 10 on allowable tensile stress. However, a value of 1-1/2 times the Table value is permitted for bending stresses.

UCI-2 SERVICE RESTRICTIONS

Cast iron shall not be used for the following services:

- Lethal or flammable substances, liquid or gaseous
- Unfired steam boilers
- Direct fired vessels

UCI-3 PRESSURE-TEMPERATURE LIMITATIONS

The limits on cast iron vessels or parts are:

- 160 psi and 450F for gas, steam or vapor contents
- 160 psi and 375F for liquid contents
- 250 psi and 120F for liquids less than their boiling point
- 250 psi and 650F if stress relieved and annealed SA 278 Classes 40, through 60

There are also flange and fitting limits. There are no low temperature limits.

UCI-37 CORNERS AND FILLETS

Liberal radii are required at corners to avoid high stress concentrations and potential cracking.

UCI-78 REPAIRS OF CAST IRON

No welding is permitted on cast iron and repairs must be with threaded plugs not exceeding NPS 2.

UCI-99 STANDARD HYDROSTATIC TEST

The test pressure is 2 times the MAWP for MAWPs > 30 psi and 2-1/2 times MAWP for lower pressures.

PART UCL
REQUIREMENTS FOR WELDED PRESSURE VESSELS WITH CORROSION RESISTANT INTEGRAL CLADDING, WELD METAL OVERLAY, OR WITH APPLIED LININGS

GENERAL

The base plate and other material used under Part UCL shall meet the material requirements of Parts UCS, UHA or UNF. General guidelines are provided in Code Appendix F.

UCL-3 CONDITIONS OF SERVICE

The user should interface with the designer to review past operating experience and select the material of the lining for the particular service. Weld materials must be selected to avoid embrittlement (see Note to UCL-3 and UCL-34) and minimize corrosion.

UCL-11 INTEGRAL AND WELD OVERLAY CLAD MATERIAL

The total thickness of clad materials SA-263, 264 and 265 or the base metal and weld metal overlay may be used in the design calculations. If the total thickness of integral clad is used in design, a 20,000 psi cladding shear test is required. For weld overlay, a procedure should exist to assure uniformity of the overlay thickness.

UCL-25 CORROSION OF CLADDING OR LINING MATERIAL

Telltale holes may extend to the cladding or lining. Open telltale holes should not be used for toxic or flammable contents.

UCL-34 POST WELD HEAT TREATMENT

PWHT shall be performed to the requirements of the base metal and is performed after the weld metal overlay or installation of the applied lining. PWHT must be carefully specified to avoid sensitizing the material.

UCL-35 RADIOGRAPHIC REQUIREMENTS

These are to the base metal requirements. When the total thickness including a weld overlay is used in design, the total thickness shall be radiographed.

UCL-36 EXAMINATION OF CHROMIUM STAINLESS STEEL CLADDING OR LINING

Applied lining welds must be examined for cracks. Many owner/users specify dye penetrant examination for applied lining welds and for weld overlay and joints of integral cladding.

UCL-42 ALLOY WELDS IN BASE METAL

This is permitted with the proper welding procedure qualification. A typical application is welding small alloy nozzles but these should be limited in service temperature due to differential thermal expansion. Small nozzles with internal alloy overlay are available even if they are too small for the vessel Manufacturer to overlay.

UCL-44 ATTACHMENT OF APPLIED LININGS
UCL-51 TIGHTNESS OF APPLIED LININGS

The attachment of linings and the tightness test are a matter of agreement between the user and the Manufacturer. Telltale holes may be used as leak detection but must not be left open for toxic or flammable contents.

PART UCD
REQUIREMENTS FOR PRESSURE VESSELS CONSTRUCTED OF DUCTILE CAST IRON

This material has many requirements that are similar to cast iron (Part UCI) but has higher ductility, allowable design pressures of 1000 psi or higher and higher allowable tensile stresses (theoretical factor of safety of five on tensile stress). The lowest MDMT is -20F. Inspection and test are similar to Part UCI vessels (e.g., hydrotest at two times MAWP).

PART UHT
REQUIREMENTS FOR PRESSURE VESSELS CONSTRUCTED OF FERRITIC STEELS WITH TENSILE PROPERTIES ENHANCED BY HEAT TREATMENT

GENERAL

This part is to be used in conjunction with the other applicable rules in the Code. It does not apply to Part UCS steels whose properties are enhanced by heat treatment nor to nonwelded integrally forged vessels. UHT materials are usually quenched and tempered high strength (over 100,000 psi minimum specified tensile strength) low alloy steels with vanadium being a common addition.

MATERIALS

UHT-5 GENERAL
UHT-6 TEST REQUIREMENTS

UHT materials may be joined to UCS or UHA materials when permitted by Part UHT. Materials shall be impact tested using specimens that have the equivalent heat treatment to that in the vessel. Impact tests shall be at the design temperature or lower but not above 32F. Testing includes Charpy V-notch tests and, for certain materials and temperatures, drop-weight tests.

DESIGN

UHT-17 WELDED JOINTS
UHT-18 NOZZLES

Nozzles in walls 2 inches thick and less must be of the readily radiographable type per Figure UHT-18.1 to satisfy the requirement that Category D joints be radiographed butt welds. Other nozzle joints shall be full penetration welds with radiography on a best effort basis. Relaxation exists for some materials and nozzles of materials that are not quenched and tempered may be used.

UHT-20 JOINT ALIGNMENT

Table UHT-20 has tighter tolerances than UW-33. UHT vessels will normally be heavy wall and if tight shell and head tolerances are not held, the alignment tolerances will not be satisfied and rework will be required.

UHT-28 STRUCTURAL ATTACHMENTS AND STIFFENING RINGS

Attachments must have a yield strength within plus or minus 20 percent of the base material or stainless steel may be used on certain base materials.

UHT-56 POSTWELD HEAT TREATMENT

A common thickness breakpoint above which PWHT is required is 0.58 in.

UHT-57 EXAMINATION

Mandatory surface examinations include magnetic particle and, for weld overlays, dye penetrant.

FABRICATION

UHT-80 HEAT TREATMENT
UHT 81 HEAT TREATMENT VERIFICATION

Metal temperatures must be held to a very close tolerance of plus or minus 25F and calibrated recording indicators are required. Requirements on immersion and spray cooling are given and should be reviewed by the owner/user, particularly if the vessel manufacturer will upgrade the material. Test coupons must be heat treated with the vessel and additional coupon material must be ordered unless adequate size cutouts will be available.

UHT-82 WELDING

This supplements Code Section IX and indicates the importance of welding materials control and welding control.

UHT-83 METHODS OF METAL REMOVAL

Methods involving melting, such as arc air gouging and gas cutting require removal of the last 1/16 inch by grinding or chipping, followed by magnetic particle or dye penetrant examination.

UHT-85 STRUCTURAL AND TEMPORARY WELDS

Qualified procedures and welders shall be used for all attachment welds. After removal of temporary welds, the area shall be ground and magnetic particle or dye penetrant examined.

PART ULW
REQUIREMENTS FOR PRESSURE VESSELS FABRICATED BY LAYERED CONSTRUCTION

These are normally large, thick wall vessels designed for high pressures. Only a few manufacturers in the world are qualified. Because of limited use, course coverage is minimal. Particular attention should be paid to; ULW-17 for design of welded joints, ULW-26 for PWHT, ULW-51 and ULW-52 for welded joint examination, ULW-53 for step welding of category B joints, ULW-54 for full wall thickness butt joints, ULW-55 for flat heads and tubesheets, ULW-56 for Category D nozzle joints, ULW-57 for ultrasonic and magnetic particle examination, ULW- 76 for vent holes, ULW-77 and ULW-78 for contact between the layers, and ULW-115 for stamping and data reports.

PART ULT
ALTERNATIVE RULES FOR PRESURE VESSELS CONSTRUCTED OF MATERIALS HAVING HIGHER ALLOWABLE STRESSES AT LOW TEMPERATURE

The yield and tensile strengths of many metals increase as temperature is lowered from ambient. This part takes advantage of this but has limited application. It can save costs by weight reduction, particularly for cryogenic road tankers. Materials are limited to aluminum and the 5, 8 and 9 percent nickel steels between 3/16 and 2 inches thick. Particular attention should be paid to ULT-2, 5, 17, 18, 56, 57, 82, 99, and 115.

APPENDICES

Most of the Code Appendices (mandatory and non-mandatory) have been mentioned in this course but whenever there seems to be a lack of information in the body of the Code, check the Appendices.

APPENDIX I-1

TABLE UW-12
MAXIMUM ALLOWABLE JOINT EFFICIENCIES[1,5] FOR ARC AND GAS WELDED JOINTS

Type No.	Joint Description	Limitations	Joint Category	Degree of Radiographic Examination (a) Full[2]	(b) Spot[3]	(c) None
(1)	Butt joints as attained by double-welding or by other means which will obtain the same quality of deposited weld metal on the inside and outside weld surfaces to agree with the requirements of UW-35. Welds using metal backing strips which remain in place are excluded.	None	A, B, C, & D	1.00	0.85	0.70
(2)	Single-welded butt joint with backing strip other than those included under (1)	*(a)* None except as in (b) below	A, B, C, & D	0.90	0.80	0.65
		(b) Circumferential butt joints with one plate offset; see UW-13(b)(4) and Fig. UW-13.1, sketch (k)	A, B, & C	0.90	0.80	0.65
(3)	Single-welded butt joint without use of backing strip	Circumferential butt joints only, not over ⅝ in. thick and not over 24 in. outside diameter	A, B, & C	NA	NA	0.60
(4)	Double full fillet lap joint	*(a)* Longitudinal joints not over ⅜ in. thick	A	NA	NA	0.55
		(b) Circumferential joints not over ⅝ in. thick	B & C	NA	NA	0.55
(5)	Single full fillet lap joints with plug welds conforming to UW-17	*(a)* Circumferential joints[4] for attachment of heads not over 24 in. outside diameter to shells not over ½ in. thick	B	NA	NA	0.50
		(b) Circumferential joints for the attachment to shells of jackets not over ⅝ in. in nominal thickness where the distance from the center of the plug weld to the edge of the plate is not less than 1½ times the diameter of the hole for the plug.	C	NA	NA	0.50

APPENDIX 1-2

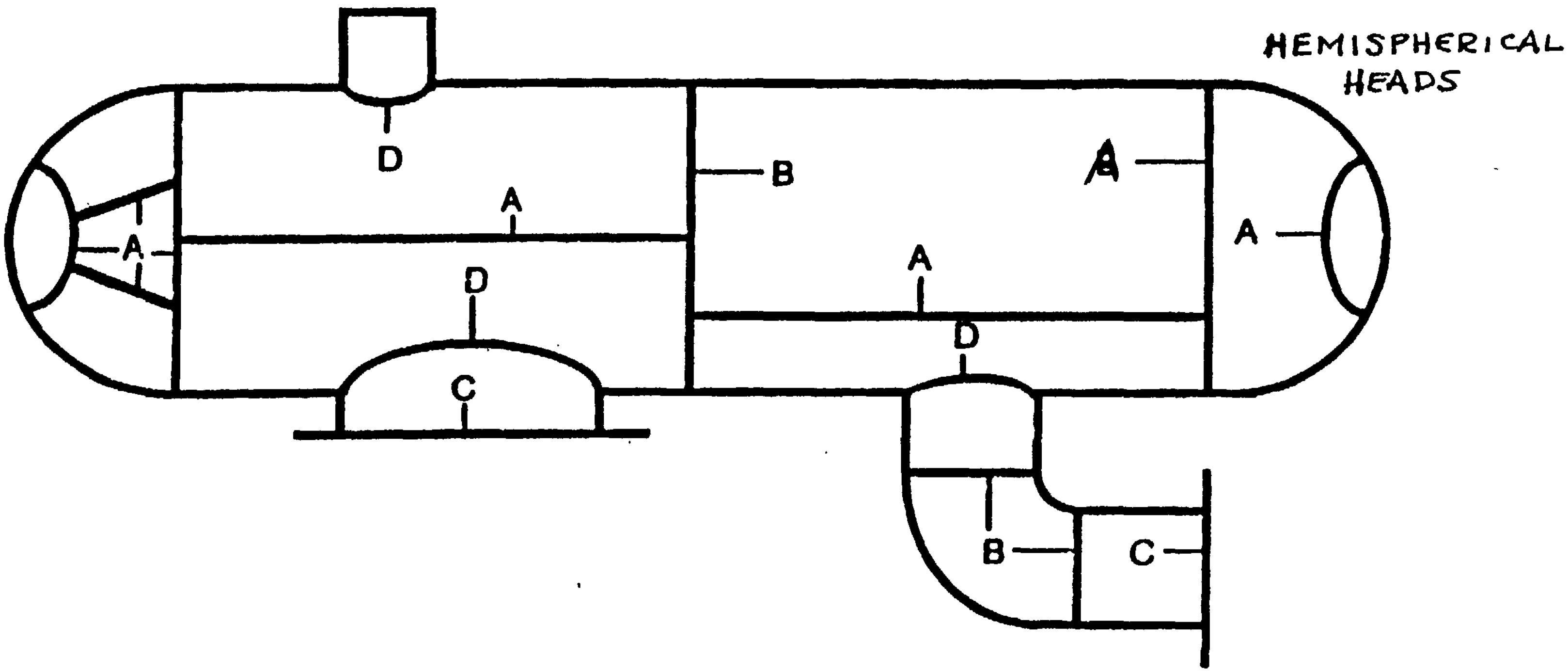
(6)	Single full fillet lap joints without plug welds	*(a)* For the attachment of heads convex to pressure to shells not over 5/8 in. required thickness, only with use of fillet weld on inside of shell; or	A & B	NA	NA	0.45
		(b) for attachment of heads having pressure on either side, to shells not over 24 in. inside diameter and not over 1/4 in. required thickness with fillet weld on outside of head flange only	A & B	NA	NA	0.45

NOTES:
(1) The single factor shown for each combination of joint category and degree of radiographic examination replaces both the stress reduction factor and the joint efficiency factor considerations previously used in this Division.
(2) See UW-12(a) and UW-51.
(3) See UW-12(b) and UW-52.
(4) Joints attaching hemispherical heads to shells are excluded.
(5) $E = 1.0$ for butt joints in compression.

APPENDIX II

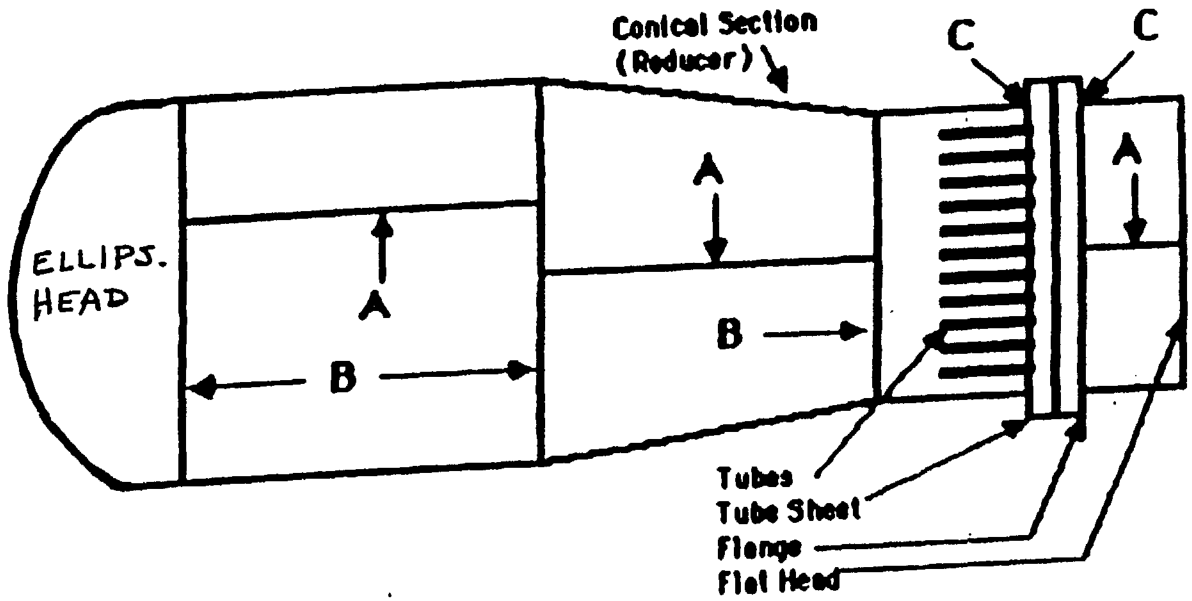
SECTION VIII-DIVISION 1

JOINT CAT.



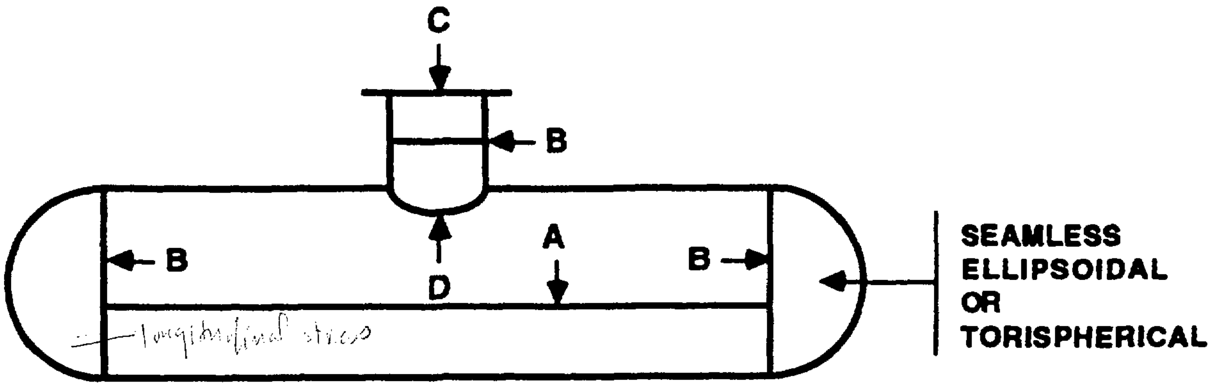
SECTION VIII - DIVISION I

APPENDIX II



ILLUSTRATIONS OF WELDED JOINT CATEGORIES

APPENDIX-VII-1
SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



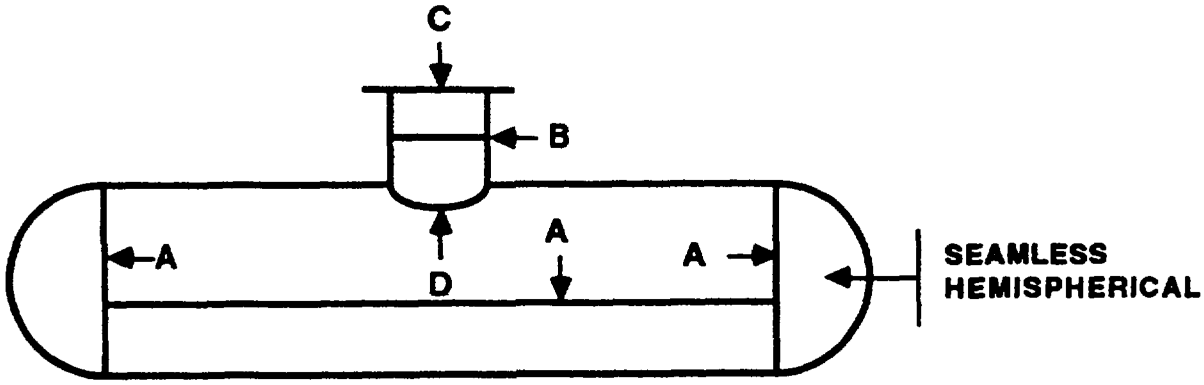
A TYPE 1 BUTT WELD FULL RT
B TYPE 1 BUTT WELD FULL RT
SEE UW-11(a)(5)(a)/UW-12(a)/TABLE UW-12(a)
E=1.00 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=1.00 SEAMLESS HEAD CALCULATIONS (QUALITY FACTOR)
E=1.00 LONGITUDINAL STRESS CALCULATIONS

A TYPE 1 BUTT WELD FULL RT
B TYPE 1 BUTT WELD SPOT RT
SEE UW-11(a)(5)(a)&(b)/UW-12(a),(b)&(d)/TABLE UW-12(a)&(b)
E=1.00 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=1.00 SEAMLESS HEAD CALCULATIONS (QUALITY FACTOR)
E=0.85 LONGITUDINAL STRESS CALCULATIONS

(Figure 1)

APPENDIX-VII-2

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA

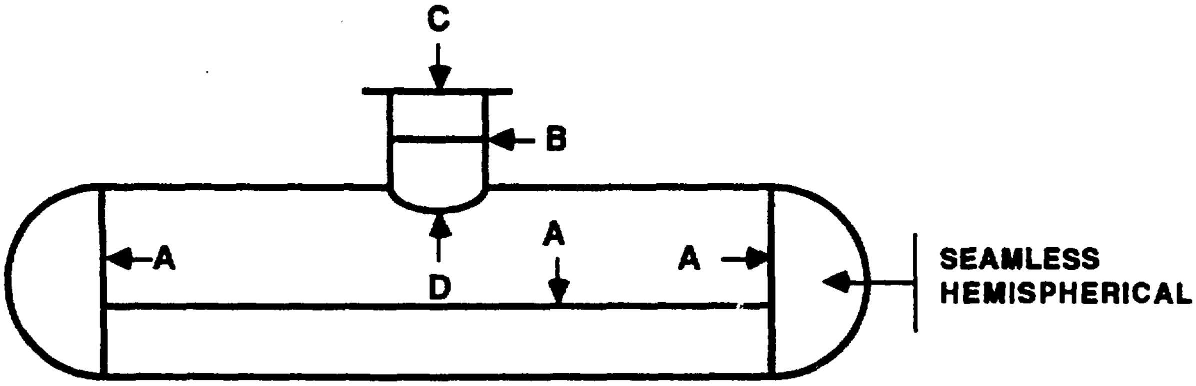


A TYPE 1 BUTT WELD FULL RT
SEE UW-11(a)(5)(a)/UW-12(a)/TABLE UW-12(a)
E=1.00 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=1.00 HEAD CALCULATIONS (JOINT EFFICIENCY)
E=1.00 LONGITUDINAL STRESS CALCULATIONS

A TYPE 1 BUTT WELD IN SHELL FULL RT
A TYPE 2 BUTT WELD HEAD TO SHELL FULL RT
SEE UW-11(a)(5)(a)/UW-12(a)/TABLE UW-12(a)
E=1.00 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.90 HEAD CALCULATIONS (JOINT EFFICIENCY)
E=0.90 LONGITUDINAL STRESS CALCULATIONS

(Figure 2)

APPENDIX-VII-3
SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



A TYPE 1 BUTT WELD IN SHELL FULL RT
A TYPE 2 BUTT WELD HEAD TO SHELL SPOT RT
SEE UW-11(a)(5)(a)&(b)/UW-12(a),(b)&(d)/TABLE UW-12(a)&(b)
E=1.00 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.80 HEAD CALCULATIONS (JOINT EFFICIENCY)
E=0.80 LONGITUDINAL STRESS CALCULATIONS

A TYPE 1 BUTT WELD IN SHELL NO RT
A TYPE 2 BUTT WELD HEAD TO SHELL NO RT
SEE UW-12(c) & (d)/TABLE UW-12(c)
E=0.70 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.65 HEAD CALCULATIONS (JOINT EFFICIENCY)
E=0.65 LONGITUDINAL STRESS CALCULATIONS

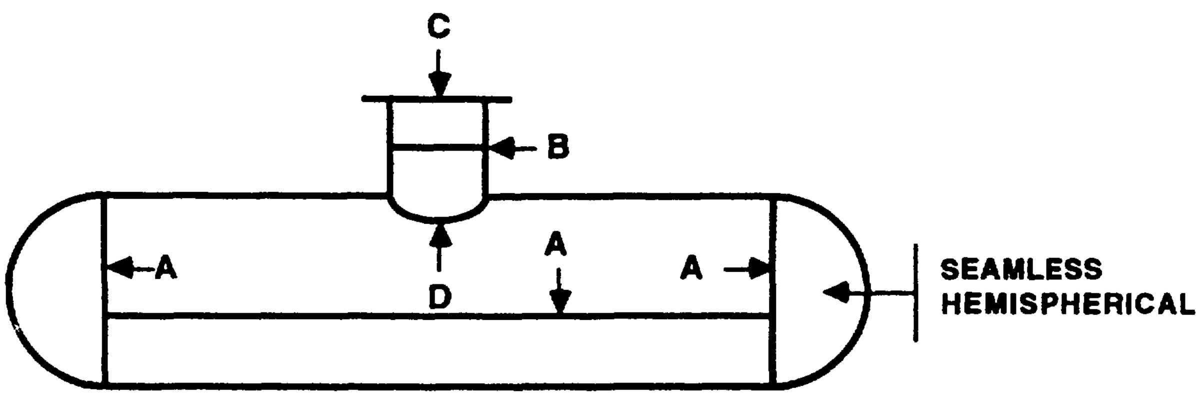
(Figure 3)

APPENDIX-VII-4

SECTION VIII-DIVISION 1

STRESS MULTIPLIERS

1986 ADDENDA



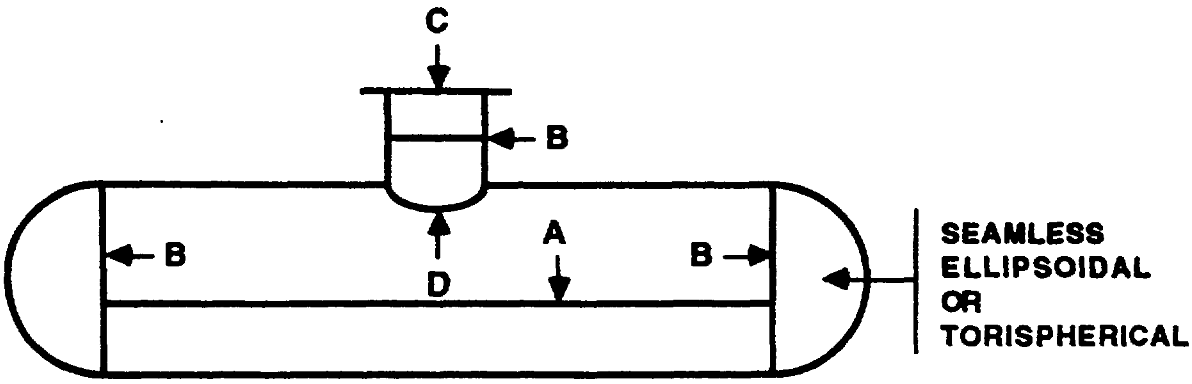
A TYPE 1 BUTT WELD IN SHELL SPOT RT
A TYPE 2 BUTT WELD HEAD TO SHELL SPOT RT
SEE UW-11(a)(5)(a)&(b)/UW-12(b)&(d)/TABLE UW-12(b)
E=0.85 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.80 HEAD CALCULATIONS (JOINT EFFICIENCY)
E=0.80 LONGITUDINAL STRESS CALCULATIONS

A TYPE 1 BUTT WELD IN SHELL SPOT RT
A TYPE 2 BUTT WELD HEAD TO SHELL NO RT
SEE UW-11(a)(5)(a)&(b)/UW-12(b),(c)&(d)/TABLE UW-12(b)&(c)
E=0.85 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.65 HEAD CALCULATIONS (JOINT EFFICIENCY)
E=0.65 LONGITUDINAL STRESS CALCULATIONS

(Figure 4)

APPENDIX-VII-5

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



A TYPE 1 BUTT WELD SPOT RT
B TYPE 1 BUTT WELD SPOT RT
SEE UW-11(a)(5)(a)&(b)/UW-12(b)&(d)/TABLE UW-12(b)
E=0.85 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=1.0 HEAD CALCULATIONS (QUALITY FACTOR)
E=0.85 LONGITUDINAL STRESS CALCULATIONS

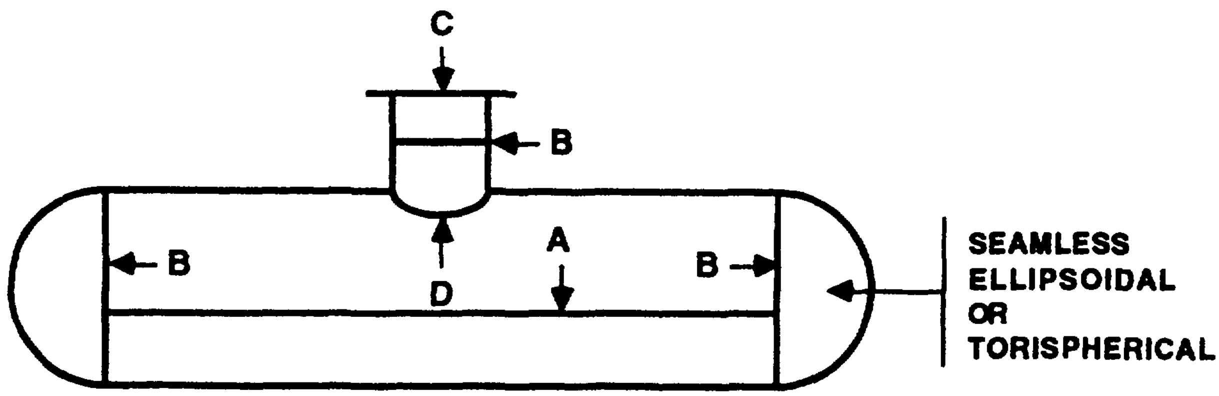
A TYPE 1 BUTT WELD SPOT RT
B TYPE 1 BUTT WELD NO RT
SEE UW-11(a)(5)(a)&(b)/UW-12(b),(c)&(d)/TABLE UW-12(b)&(c)
E=0.85 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=1.0 (.85) HEAD CALCULATIONS (QUALITY FACTOR)
E=0.85 LONGITUDINAL STRESS CALCULATIONS

JP RULE

(Figure 5)

APPENDIX-VII-6

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



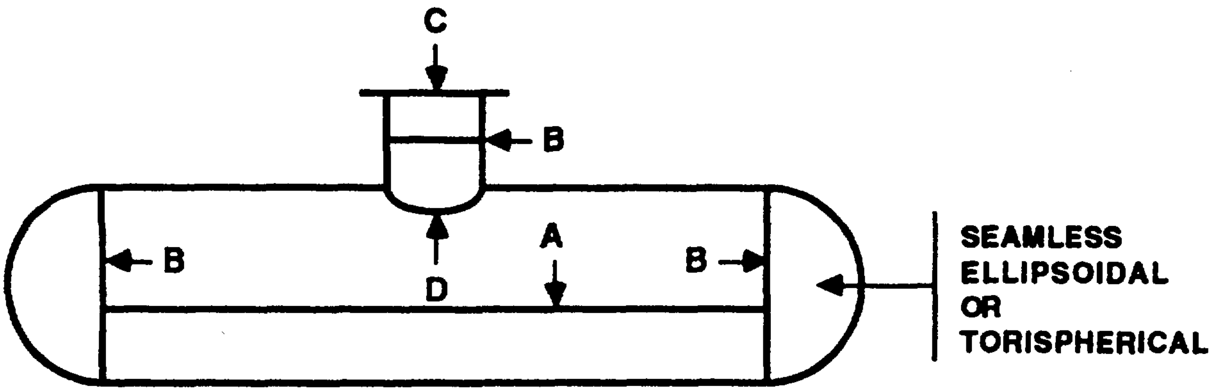
A TYPE 1 BUTT WELD SPOT RT
B TYPE 2 BUTT WELD SPOT RT
SEE UW-11(a)(5)(a)&(b)/UW-12(b)&(d)/TABLE UW-12(b)
E=0.85 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=1.00 HEAD CALCULATIONS (QUALITY FACTOR)
E=0.80 LONGITUDINAL STRESS CALCULATIONS

A TYPE 1 BUTT WELD SPOT RT
B TYPE 2 BUTT WELD NOT RT
SEE UW-11(a)(5)(a)&(b)/UW-12(b)(c)&(d)/TABLE UW-12(b)&(c)
E=0.85 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.85 HEAD CALCULATIONS (QUALITY FACTOR)
E=0.65 LONGITUDINAL STRESS CALCULATIONS

(Figure 6)

APPENDIX-VII-7

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



A TYPE 2 BUTT WELD NO RT
B TYPE 3 BUTT WELD NO RT
SEE UW-11(a)(5)(a)&(b)/UW-12(c)&(d)/TABLE UW-12(c)
E=0.65 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.85 SEAMLESS HEAD CALCULATIONS (QUALITY FACTOR)
E=0.60 THE LONGITUDINAL STRESS CALCULATIONS

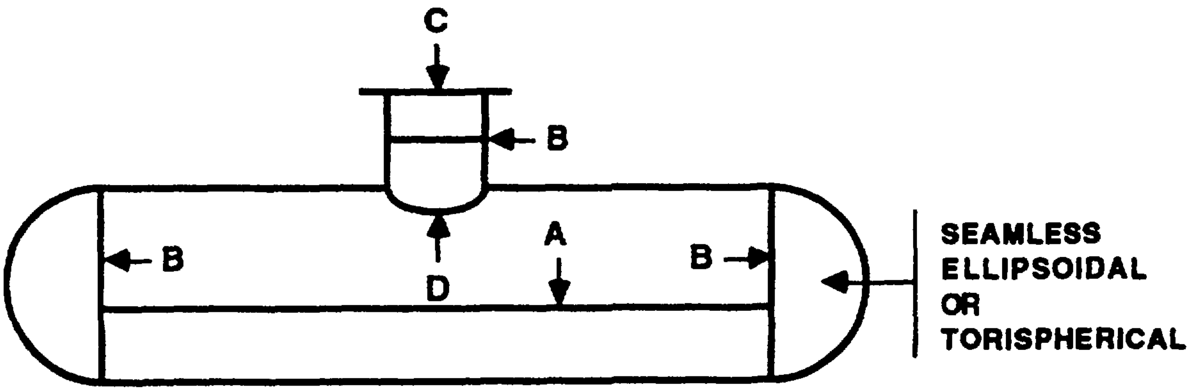
A TYPE 2 BUTT WELD SPOT RT
B TYPE 3 BUTT WELD FULL OR SPOT RT
SEE UW-11(a)(5)(a)&(b)/UW-12(a),(b)&(d)/TABLE UW-12(a)&(b)
E=0.80 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.85 SEAMLESS HEAD CALCULATIONS (QUALITY FACTOR)*
E=0.60 LONGITUDINAL STRESS CALCULATIONS

NOTE: *INTERPRETATIONS-VOLUME 20-JULY, 1987 CLARIFY INTENT
REVISIONS APPEARED 12/31/87 ADDENDA

(Figure 7)

APPENDIX-VII-8

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



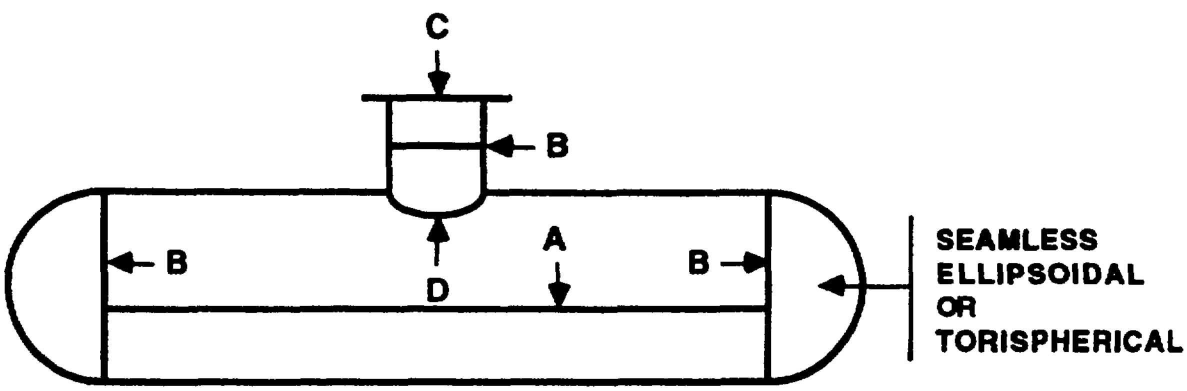
A TYPE 1 BUTT WELD NO RT
B TYPE 1 BUTT WELD NO RT
SEE UW-12(c)&(d)/TABLE UW-12(c)
E=0.70 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.85 HEAD CALCULATIONS (QUALITY FACTOR)
E=0.70 LONGITUDINAL STRESS CALCULATIONS

A TYPE 1 BUTT WELD NO RT
B TYPE 1 BUTT WELD SPOT RT
SEE UW-11(a)(5)(a)&(b)/UW-12(b),(c)&(d)/TABLE UW-12(b)&(c)
E=0.70 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=1.00 HEAD CALCULATIONS (QUALITY FACTOR)
E=0.85 LONGITUDINAL STRESS CALCULATIONS

(Figure 8)

APPENDIX-VII-9

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



A TYPE 1 BUTT WELD NO RT
B TYPE 6 SINGLE FILLET WELD
SEE UW-11(a)(5)(a)&(b)/UW-12(d)/TABLE UW-12(c)
E= 0.70 SHELL CALCULATIONS (JOINT EFFICIENCY)
E= 0.85 HEAD CALCULATIONS (QUALITY FACTOR)
E= 0.45 LONGITUDINAL STRESS CALCULATIONS

A TYPE 1 BUTT WELD SPOT RT
B TYPE 6 SINGLE FILLET WELD
SEE UW-11(a)(5)(a)&(b)/UW-12(b)&(d)/Table UW-12(b)&(c)
E= 0.85 SHELL CALCULATIONS (JOINT EFFICIENCY)
E= 0.85 HEAD CALCULATIONS (QUALITY FACTOR)
E= 0.45 LONGITUDINAL STRESS CALCULATIONS

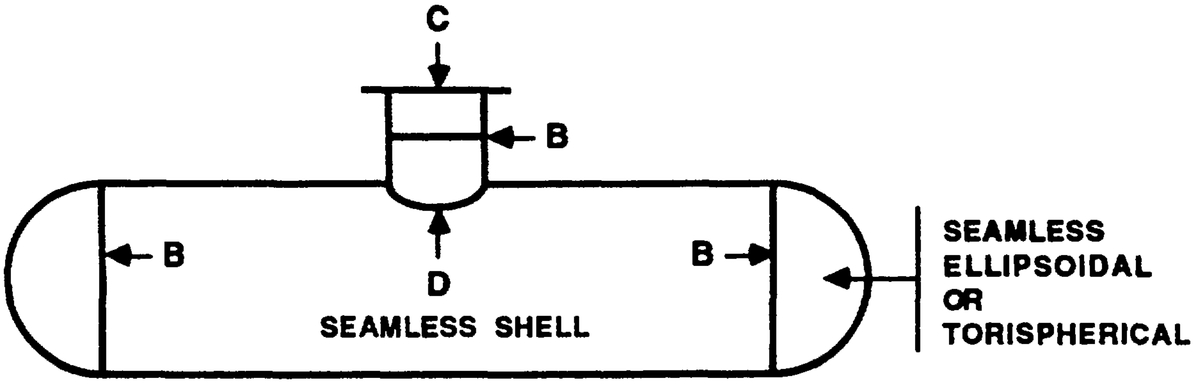
B TYPE 4 DOUBLE FILLET WELD SAME AS B TYPE 6 EXCEPT
E=0.55 LONGITUDINAL STRESS CALCULATIONS

NOTE: INTERPRETATIONS-VOLUME 20-JULY,1987 CLARIFY INTENT
REVISIONS APPEARED 12/31/87 ADDENDA

(Figure 9)

APPENDIX-VII-10

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



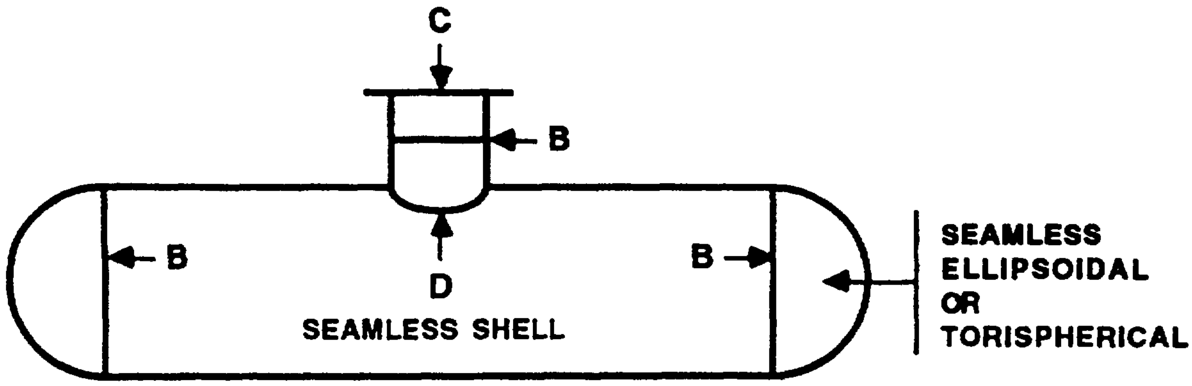
SEAMLESS SHELL
B TYPE 1 BUTT WELD FULL RT
SEE UW-11(a)(5)(a)/UW-12(a)&(d)/TABLE UW-12(a)
E=1.00 SHELL CALCULATIONS (QUALITY FACTOR)
E=1.00 HEAD CALCULATIONS (QUALITY FACTOR)
E=1.00 LONGITUDINAL STRESS CALCULATIONS

SEAMLESS SHELL
B TYPE 1 BUTT WELD SPOT RT
SEE UW-11(a)(5)(a)&(b)/UW-12(b)&(d)/TABLE UW-12(c)
E=1.00 SHELL CALCULATIONS (QUALITY FACTOR)
E=1.00 HEAD CALCULATIONS (QUALITY FACTOR)
E=0.85 LONGITUDINAL STRESS CALCULATIONS

(Figure 10)

APPENDIX-VII-11

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



SEAMLESS SHELL
B TYPE 6 SINGLE FILLET WELD
SEE UW-11(a)(5)(a)/UW-12(d)/TABLE UW-12(c)
E=0.85 SHELL CALCULATIONS (QUALITY FACTOR)
E=0.85 HEAD CALCULATIONS (QUALITY FACTOR)
E=0.45 LONGITUDINAL STRESS CALCULATIONS

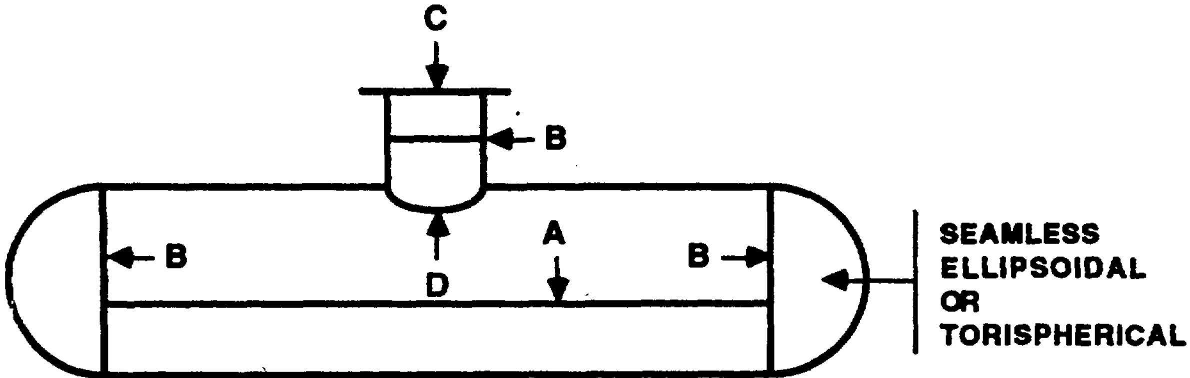
B TYPE 4 DOUBLE FILLET WELD SAME AS B TYPE 6 EXCEPT
E=0.55 LONGITUDINAL STRESS CALCULATIONS

NOTE: INTENT IN INTERPRETATIONS-VOLUME 20-JULY,1987
REVISIONS APPEAR IN THE 12/31/87 ADDENDA

(Figure 11)

APPENDIX-VII-12

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



electric resistance weld

A ERW PIPE or seamless
B TYPE 1 BUTT WELD SPOT RT
SEE UW-11(a)(5)(a)&(b)/UW-12(b)&(d)/TABLE UW-12(b)
E= 1.0 SHELL CALCULATIONS (QUALITY FACTOR), NOTE 1 spiral
E= 1.0 HEAD CALCULATIONS (QUALITY FACTOR)
E= ~~1.0~~ 0.85 LONGITUDINAL STRESS CALCULATIONS, NOTE 2

A ERW PIPE
B TYPE 1 BUTT WELD NO RT
SEE UW-11(a)(5)/UW-12(c)&(d)/TABLE UW-12(c)
E= 0.85 SHELL CALCULATIONS (QUALITY FACTOR), NOTE 1
E= 0.85 HEAD CALCULATIONS (QUALITY FACTOR)
E=___ LONGITUDINAL STRESS CALCULATIONS, NOTE 2

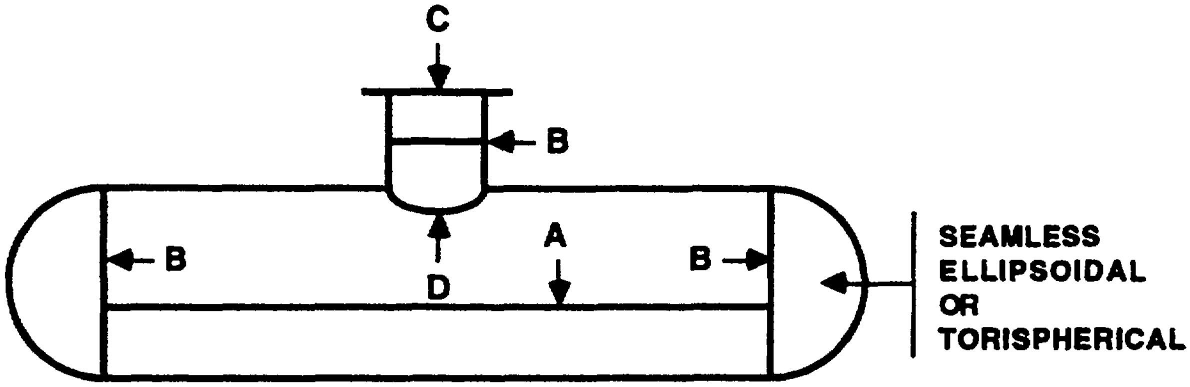
NOTE 1: THE QUALITY FACTOR SHOWN IN THE SHELL CALCULATIONS IS IN ADDITION TO THE CATEGORY A JOINT FACTOR (E=0.85) INCLUDED IN THE ALLOWABLE STRESS VALUE OBTAINED FROM THE APPLICABLE STRESS TABLE.

NOTE 2: DIVIDE THE ALLOWABLE STRESS VALUE OBTAINED FROM THE APPLICABLE STRESS TABLE BY 0.85 BEFORE APPLYING A JOINT EFFICIENCY IN THE LONGITUDINAL STRESS CALCULATION (eg, SEE NOTE 26, TABLE UCS-23)

(Figure 12)

APPENDIX-VII-13

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



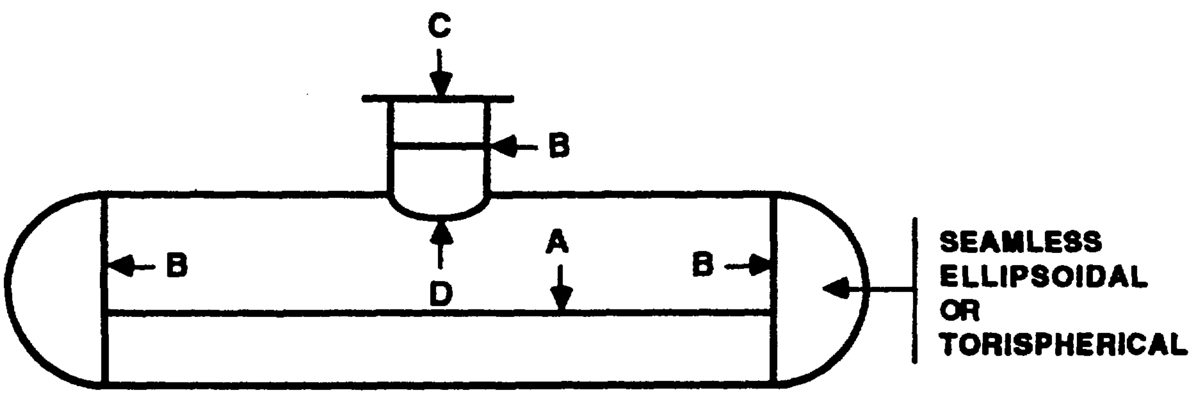
D TYPE 1 BUTT WELD
SEE UW-11(a)(5) FULL RT IF PART IS DESIGNED E=1.00
RT NOT REQUIRED FOR tr or trn CALCULATIONS E=1.00
E=EFFICIENCY IF OPENING THROUGH CATEGORY A WELD

D FULL OR PARTIAL PENETRATION CORNER WELD
RT NOT REQUIRED E=1.00
RT NOT REQUIRED FOR tr or trn CALCULATIONS E=1.00
E=EFFICIENCY OF THE BUTT WELD PENETRATED

(Figure 13)

APPENDIX-VII-14

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



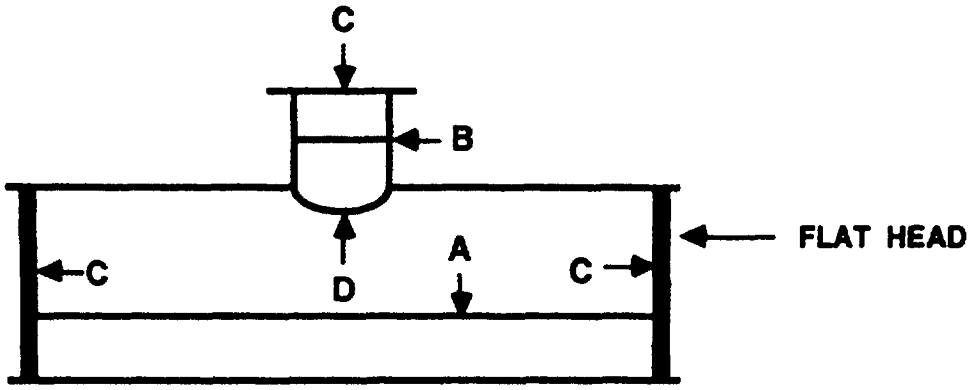
B&C TYPES I OR 2 BUTT WELD IN NOZZLES/COMMUNICATING CHAMBERS
RT NOT REQUIRED TO CALCULATE trn
RT MAY BE REQUIRED FOR:
- **SERVICE RESTRICTION**
- **MATERIAL THICKNESS**
- **USER/DESIGNATED AGENT**

NOTE: THIS SUBJECT IS UNDER CONSIDERATION FOR REVISIONS

(Figure 14)

APPENDIX-VII-15

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



C IS PARTIAL OR FULL PENETRATION CORNER JOINT FOR ALL CASES

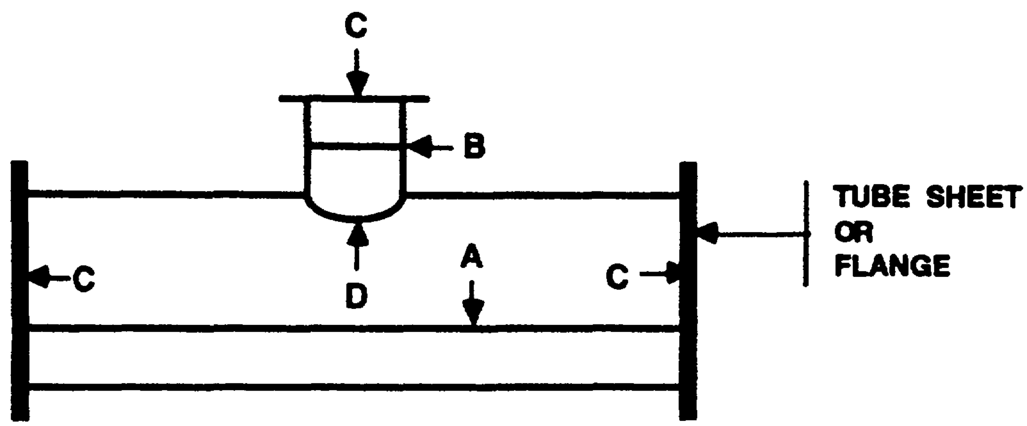
E FOR C IS NOT ESTABLISHED BY CODE RULES

A TYPE 1 BUTT WELD FULL RT - E=1.00
A TYPE 1 BUTT WELD SPOT RT - E=0.85
A TYPE 1 BUTT WELD NO RT - E=0.70
A ERW BUTT WELD-E=1.00 (JOINT EFFICIENCY IN STRESS VALUE)
A SEAMLESS - E=1.00

NOTE: FLAT HEAD FORMULA HAS BUILT-IN STRESS MULTIPILER REGARDLESS OF TYPE OF JOINT OR EXAMINATION. A FACTOR FOR E WILL APPLY IN THE FLAT HEAD FORMULA IF A CATEGORY A JOINT DOES EXIST IN THE HEAD

(Figure 15)

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



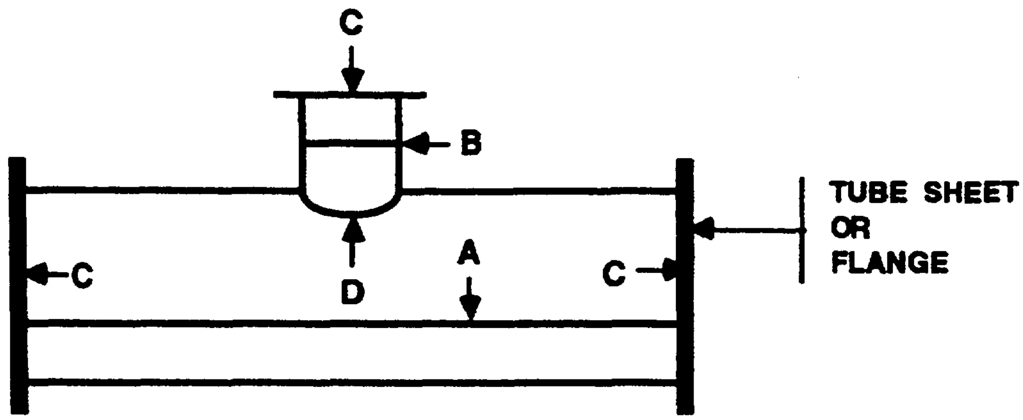
A TYPE 1 BUTT WELD FULL RT
C TYPE 1 BUTT WELD FULL RT
E=1.00 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=1.00 LONGITUDINAL STRESS CALCULATIONS

A TYPE 1 BUTT WELD FULL RT
C TYPE 1 BUTT WELD SPOT RT
E=1.00 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.85 LONGITUDINAL STRESS CALCULATIONS

(Figure 16)

APPENDIX-VII-17

SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA

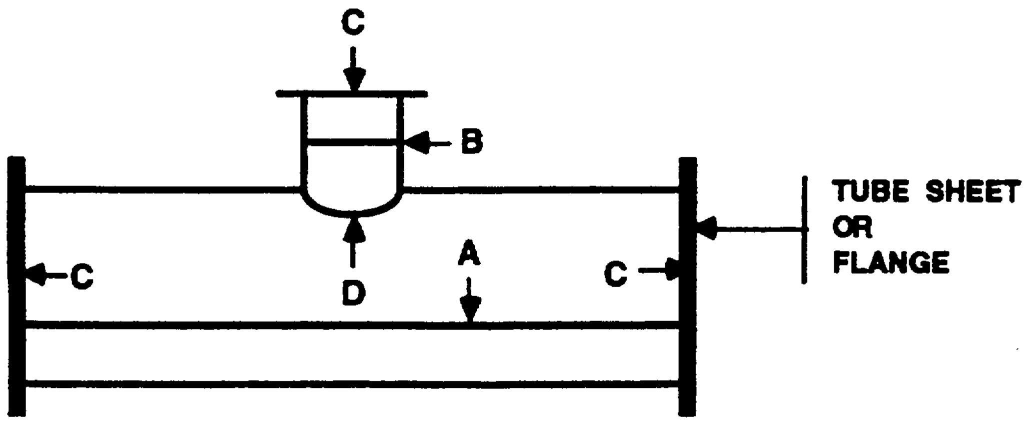


A TYPE 1 BUTT WELD FULL RT
C TYPE 1 BUTT WELD NO RT
E=0.85 SHELL CALCULATIONS (QUALITY FACTOR)
E=0.70 LONGITUDINAL STRESS CALCULATIONS

A TYPE 1 BUTT WELD SPOT RT
C TYPE 1 BUTT WELD NO RT
E=0.85 SHELL CALCULATIONS (JOINT EFFICIENCY)
E=0.70 LONGITUDINAL STRESS CALCULATIONS

(Figure 17)

APPENDIX-VII-18
SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



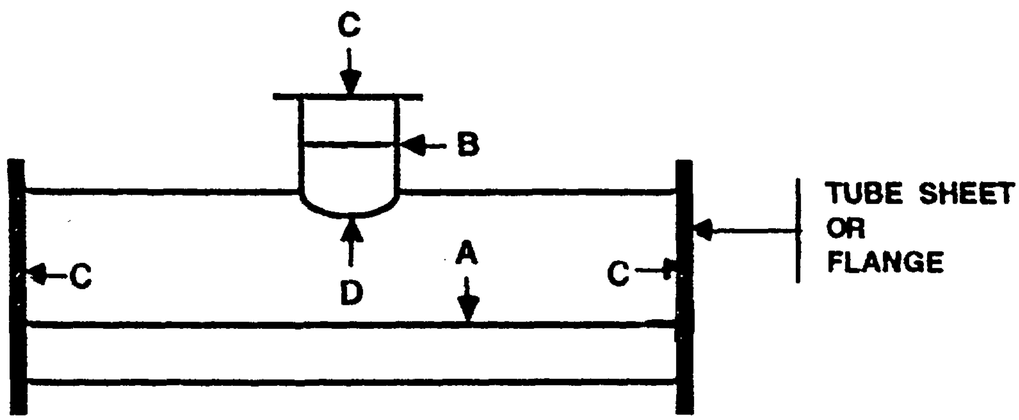
A TYPE 1 BUTT WELD FULL RT
C WELD FULL OR PARTIAL PENETRATION CORNER JOINT
E=1.00 SHELL CALCULATIONS (JOINT EFFICIENCY)
E FOR C IS NOT ESTABLISHED BY CODE RULES

A TYPE 1 BUTT WELD SPOT RT
C WELD FULL OR PARTIAL PENETRATION CORNER JOINT
E=0.85 SHELL CALCULATIONS (JOINT EFFICIENCY)
E FOR C IS NOT ESTABLISHED BY CODE RULES

A TYPE 1 BUTT WELD NO RT
C WELD FULL OR PARTIAL PENETRATION CORNER JOINT
E=0.70 SHELL CALCULATIONS (JOINT EFFICIENCY)
E FOR C IS NOT ESTABLISHED BY CODE RULES

(Figure 18)

APPENDIX-VII-19
SECTION VIII-DIVISION 1
STRESS MULTIPLIERS
1986 ADDENDA



A ERW PIPE seamless
C TYPE 1 BUTT WELD SPOT RT
E=1.0 SHELL CALCULATIONS (QUALITY FACTOR), NOTE 1
E=0.85 LONGITUDINAL STRESS CALCULATIONS, NOTE 2

A ERW PIPE
C TYPE 1 BUTT WELD NO RT
E=0.85 SHELL CALCULATIONS (QUALITY FACTOR), NOTE 1
E=0.7 LONGITUDINAL STRESS CALCULATIONS, NOTE 2

A ERW PIPE
C TYPE 2 BUTT WELD NO RT
E=0.85 SHELL CALCULATIONS (QUALITY FACTOR), NOTE 1
E=0.65 LONGITUDINAL STRESS CALCULATIONS, NOTE 2

A ERW PIPE
C FULL OR PARTIAL PENETRATION CORNER JOINT
E=1.0 SHELL CALCULATIONS (QUALITY FACTOR), NOTE 1
E FOR C IS NOT ESTABLISHED BY CODE RULES, NOTE 2

NOTE 1: THE QUALITY FACTOR SHOWN IN THE SHELL CALCULATIONS IS IN ADDITION TO THE CATEGORY A JOINT FACTOR (E=0.85) INCLUDED IN THE ALLOWABLE STRESS VALUE OBTAINED FROM THE APPLICABLE STRESS TALBLE

NOTE 2: DIVIDE THE ALLOWABLE STRESS VALUE OBTAINED FROM THE APPLICABLE STRESS TABLE BY 0.85 BEFORE APPLYING A JOINT EFFICIENCY IN THE LONGITUDINAL STRESS CALCULATIONS (eg. SEE NOTE 26, TABLE UCS-23)

(Figure 19)

TABLE UCS-57
THICKNESS ABOVE WHICH FULL RADIOGRAPHIC EXAMINATION OF BUTT WELDED JOINTS IS MANDATORY

P-No. & Gr. No. Classification of Material	Nominal Thickness Above Which Butt Welded Joints Shall Be Fully Radiographed, in.
1 Gr. 1, 2, 3	$1\frac{1}{4}$
3 Gr. 1, 2, 3	$\frac{3}{4}$
4 Gr. 1, 2	$\frac{5}{8}$
5A, 5B Gr. 1	0
9A Gr. 1	$\frac{5}{8}$
9B Gr. 1	$\frac{5}{8}$
10A Gr. 1	$\frac{3}{4}$
10B Gr. 2	$\frac{5}{8}$
10C Gr. 1	$\frac{5}{8}$
10F Gr. 6	$\frac{3}{4}$

LOW TEMPERATURE OPERATION

UCS-65 SCOPE

The following paragraphs contain requirements for vessels and vessel parts constructed of carbon and low alloy steels with respect to minimum design metal temperatures.

UCS-66 MATERIALS

(a) Figure UCS-66 shall be used to establish impact testing exemptions for steels listed in Part UCS. Unless otherwise exempted by the rules of this Division, impact testing is required for a combination of minimum design metal temperature (see UG-20) and thickness (as defined below) which is below the curve assigned to the subject material. If a minimum design metal temperature and thickness combination is on or above the curve, impact testing is not required by the rules of this Division, except as required by UCS-67(a)(2) for weld metal.

Components, such as shells, heads, nozzles, manways, reinforcing pads, flanges, tubesheets, flat cover plates, and attachments which are essential to the structural integrity of the vessel when welded to pressure retaining components, shall be treated as separate components. Each component shall be evaluated for impact test requirements based on its individual material classification, thickness as defined in (1), (2), or (3) below, and the minimum design metal temperature.

The following thickness limitations apply when using Fig. UCS-66.

(1) Excluding castings, the governing thickness t_g of a welded part is as follows:

(a) for butt joints except those in flat heads and tubesheets, the nominal thickness of the thickest welded joint [see Fig. UCS-66.3 sketch (a)];

(b) for corner, fillet, or lap welded joints, including attachments as defined above, the thinner of the two parts joined;

(c) for flat heads or tubesheets, the larger of (b) above or the flat component thickness divided by 4;

(d) for welded assemblies comprised of more than two components (e.g., nozzle-to-shell joint with reinforcing pad), the governing thickness and permissible minimum design metal temperature of each of the individual welded joints of the assembly shall be determined, and the warmest of the minimum design metal temperatures shall be used as the permissible minimum design metal temperature of the welded assembly. [See Fig. UCS-66.3 sketch (b), L-9.3.1, and L-9.5.2.]

If the governing thickness at any welded joint exceeds 4 in. and the minimum design metal temperature is colder than 120°F, impact tested material shall be used.

(2) The governing thickness of a casting shall be its largest nominal thickness.

(3) The governing thickness of flat nonwelded parts, such as bolted flanges, tubesheets, and flat heads, is the flat component thickness divided by 4. **A96**

(4) The governing thickness of a nonwelded dished head [see Fig. 1-6 sketch (c)] is the greater of the flat flange thickness divided by 4 or the minimum thickness of the dished portion. **A96**

(5) If the governing thickness of the nonwelded part exceeds 6 in. and the minimum design metal temperature is colder than 120°F, impact tested material shall be used. **A96**

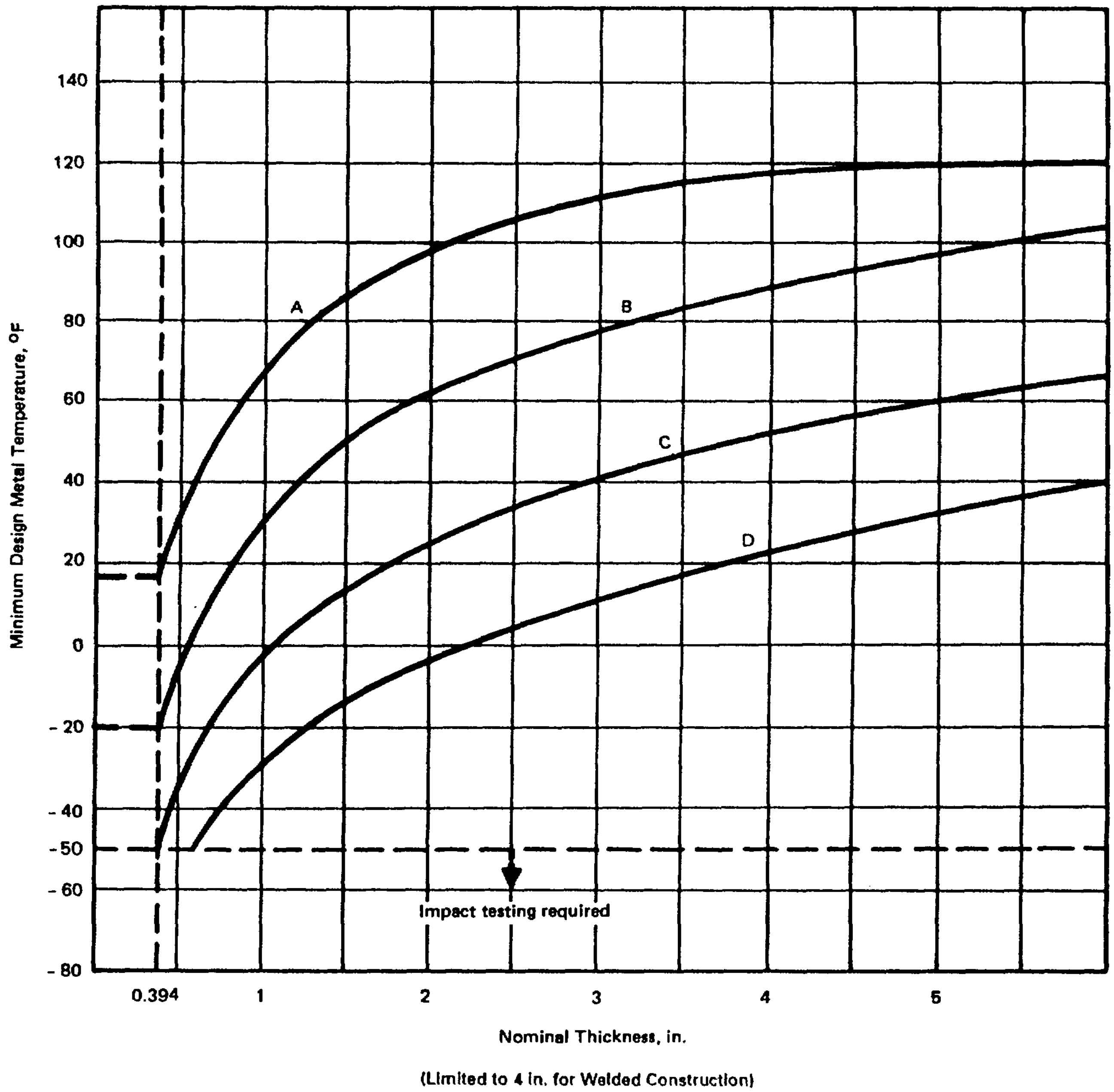
Examples of the governing thickness for some typical vessel details are shown in Fig. UCS-66.3.

NOTE: The use of provisions in UCS-66 which waive the requirements for impact testing does not provide assurance that all test results for these materials would satisfy the impact energy requirements of UG-84 if tested.

(b) When the coincident Ratio defined in Fig. UCS-66.1 is less than one, this Figure provides a further basis for the use of Part UCS material without impact testing. **A96**

(1) For such vessels, and for minimum design metal temperatures of −50°F and warmer, the minimum design metal temperature without impact testing determined in (a) above for the given material and thickness may be reduced as determined from Fig. UCS-66.2. If the resulting temperature is colder than the required minimum design metal temperature, impact testing of the material is not required.

(2) For minimum design temperatures colder than −50°F, impact testing is required for all materials, except as allowed in (b)(3) below and in UCS-68(c).



General Notes and Notes follow on next page

A96 FIG. UCS-66 IMPACT TEST EXEMPTION CURVES [SEE NOTES (1) AND (2)] [SEE UCS-66(a)]

FIG. UCS-66 (CONT'D)

A96 GENERAL NOTES ON ASSIGNMENT OF MATERIALS TO CURVES:

(a) Curve A applies to:
  (1) all carbon and all low alloy steel plates, structural shapes, and bars not listed in Curves B, C, and D below;
  (2) SA-216 Grades WCB and WCC if normalized and tempered or water-quenched and tempered; SA-217 Grade WC6 if normalized and tempered or water-quenched and tempered.

(b) Curve B applies to:
  (1) SA-216 Grade WCA if normalized and tempered or water-quenched and tempered
  SA-216 Grades WCB and WCC for thicknesses not exceeding 2 in., if produced to fine grain practice and water-quenched and tempered
  SA-217 Grade WC9 if normalized and tempered
  SA-285 Grades A and B
  SA-414 Grade A
  X SA-515 Grade 60
  SA-516 Grades 65 and 70 if not normalized
  SA-612 if not normalized
  SA-662 Grade B if not normalized;
  (2) except for cast steels, all materials of Curve A if produced to fine grain practice and normalized which are not listed in Curves C and D below;
  (3) all pipe, fittings, forgings and tubing not listed for Curves C and D below;
  (4) parts permitted under UG-11 shall be included in Curve B even when fabricated from plate that otherwise would be assigned to a different curve.

(c) Curve C
  (1) SA-182 Grades 21 and 22 if normalized and tempered
  SA-302 Grades C and D
  SA-336 F21 and F22 if normalized and tempered
  SA-387 Grades 21 and 22 if normalized and tempered
  SA-516 Grades 55 and 60 if not normalized
  SA-533 Grades B and C
  SA-662 Grade A;
  (2) all material of Curve B if produced to fine grain practice and normalized and not listed for Curve D below.

(d) Curve D
  SA-203
  SA-508 Class 1
  SA-516 if normalized
  SA-524 Classes 1 and 2
  SA-537 Classes 1 and 2
  SA-612 if normalized
  SA-662 if normalized
  SA-738 Grade A

(e) For bolting the following impact test exemption temperature shall apply:

Spec. No.	Grade	Impact Test Exemption Temperature, °F
SA-193	B5	−20
SA-193	B7	−40
SA-193	B7M	−50
SA-193	B16	−20
SA-307	B	−20
SA-320	L7, L7A, L7M, L43	Impact tested
SA-325	1, 2	−20
SA-354	BC	0
SA-354	BD	+20
SA-449	. . .	−20
SA-540	B23/24	+10

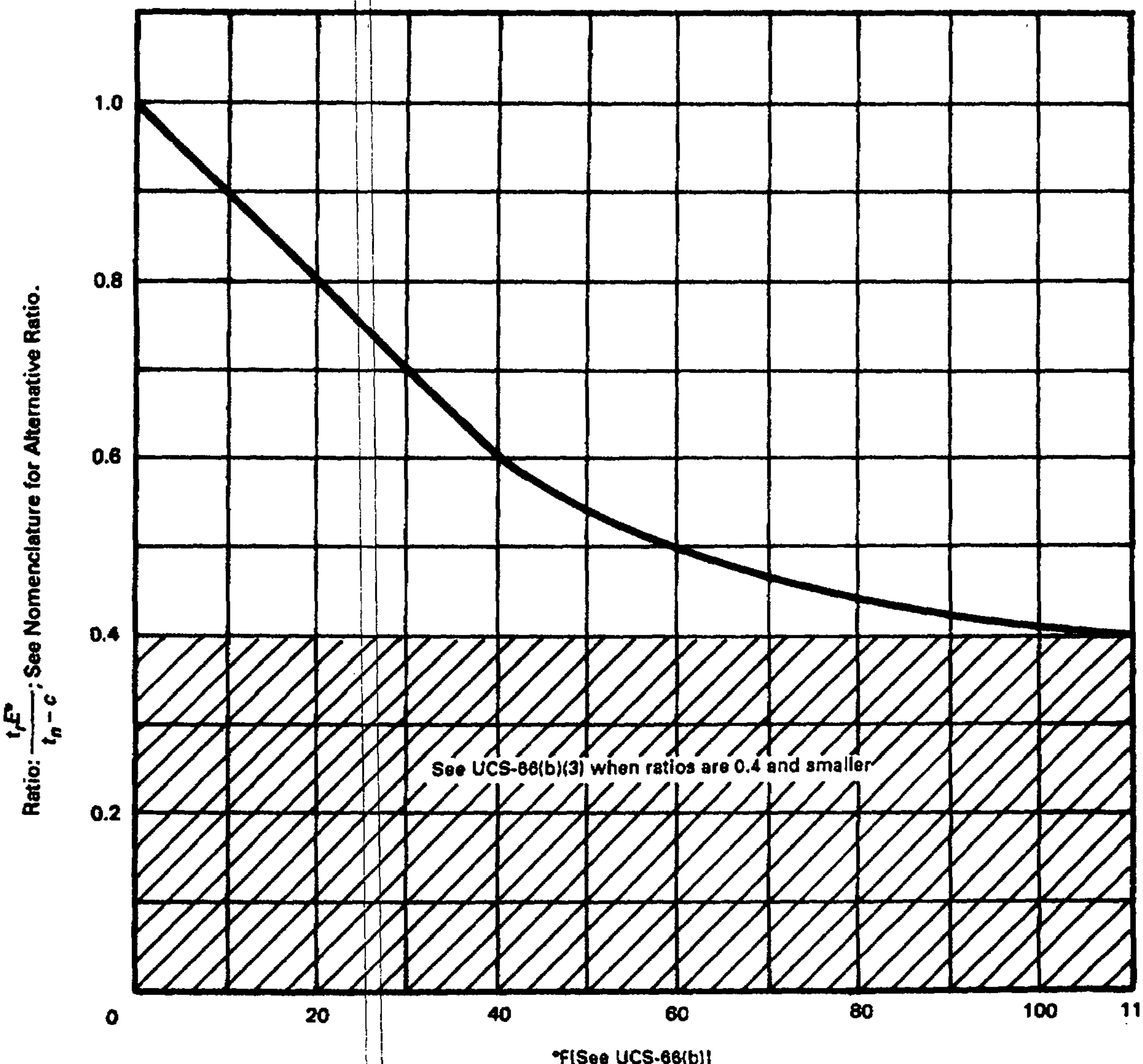
(f) When no class or grade is shown, all classes or grades are included.

(g) The following shall apply to all material assignment notes.
  (1) Cooling rates faster than those obtained by cooling in air, followed by tempering, as permitted by the material specification, are considered to be equivalent to normalizing or normalizing and tempering heat treatments.
  (2) Fine grain practice is defined as the procedure necessary to obtain a fine austenitic grain size as described in SA-20.

A96 NOTES:

(1) Tabular values for this Figure are provided in Table UCS-66.

(2) Castings not listed in General Notes (a) and (b) above shall be impact tested.



New

NOMENCLATURE (Note references are to General Notes of Fig. UCS-66.2.)

t_r = required thickness of the component under consideration in corroded condition for all applicable loadings [General Note (2)], based on the applicable joint efficiency E [General Note (3)], in.

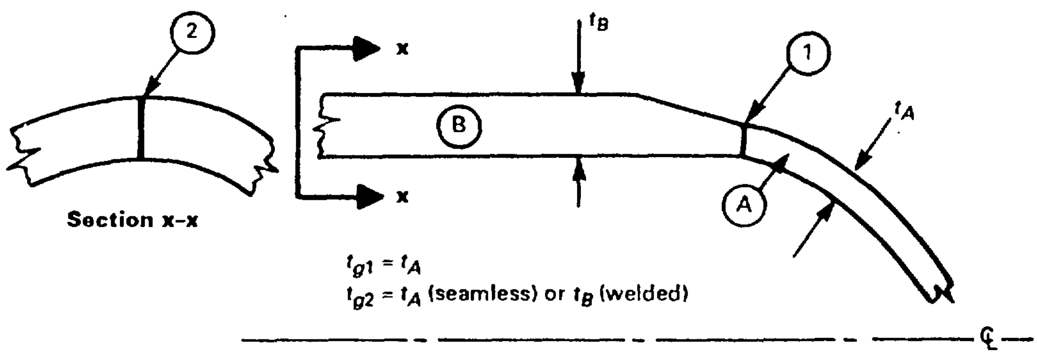
t_n = nominal thickness of the component under consideration before corrosion allowance is deducted, in.

c = corrosion allowance, in.

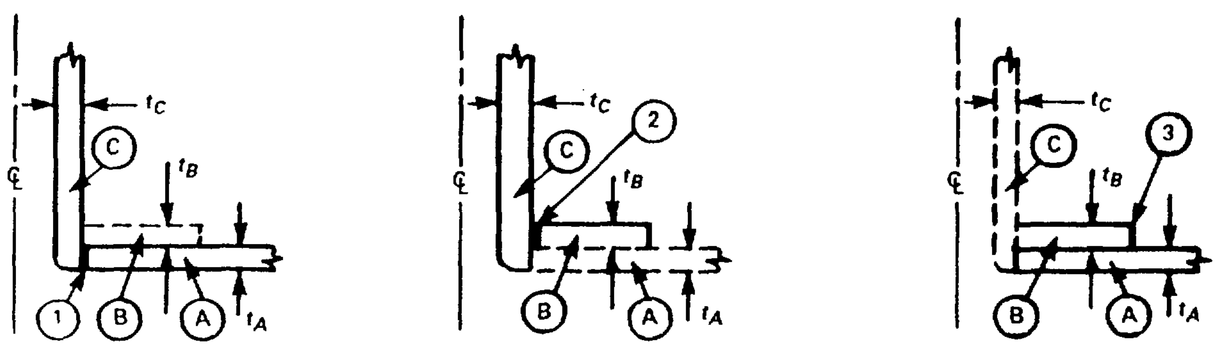
E^* = as defined in General Note (3)

Alternative Ratio = S^*E^* divided by the product of the maximum allowable stress value in tension from Table UCS-23 times E, where S^* is the applied general primary membrane tensile stress and E and E^* are as defined in General Note (3).

A96 FIG. UCS-66.1 REDUCTION IN MINIMUM DESIGN METAL TEMPERATURE WITHOUT IMPACT TESTING



(a) Butt Welded Components



t_{g1} = the thinner of t_A or t_C

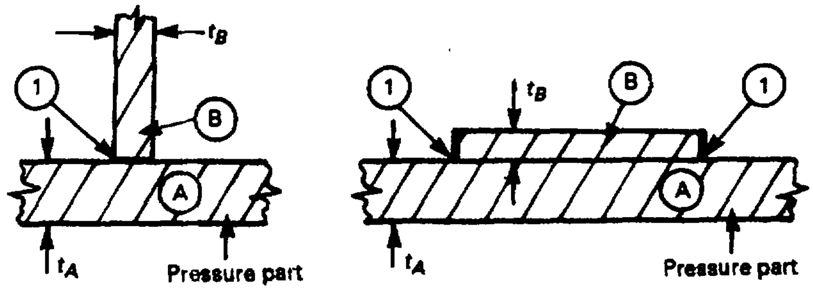
t_{g2} = the thinner of t_B or t_C

t_{g3} = the thinner of t_A or t_B

NOTE: Using t_{g1}, t_{g2}, and t_{g3}, determine the warmest MDMT and use that as the permissible MDMT for the welded assembly.

(b) Welded Connection with Reinforcement Plate Added

FIG. UCS-66.3 SOME TYPICAL VESSEL DETAILS SHOWING THE GOVERNING THICKNESSES AS DEFINED IN UCS-66



t_{g1} = thinner of t_A or t_B

(f) Welded Attachments as Defined in UCS-66(a)

NOTE: t_g = governing thickness of the welded joint as defined in UCS-66.

FIG. UCS-66.3 SOME TYPICAL VESSEL DETAILS SHOWING THE GOVERNING THICKNESSES AS DEFINED IN UCS-66 (CONT'D)

(3) When the minimum design metal temperature is colder than −50°F and no colder than −150°F, and the coincident Ratio defined in Fig. UCS-66.1 is less than or equal to 0.4, impact testing is not required.

(c) No impact testing is required for ASME/ANSI B16.5 or ASME B16.47 ferritic steel flanges used at design metal temperatures no colder than −20°F.

(d) No impact test is required for UCS materials less than 0.098 in. thick or for nuts, but such exempted UCS materials shall not be used at design metal temperatures colder than −50°F.

(e) The material manufacturer's identification marking required by the material specification shall not be stamped on plate material less than ¼ in. in thickness unless the following requirements are met.

(1) The materials shall be limited to P-No. 1 Gr. Nos. 1 and 2.

(2) The minimum nominal plate thickness shall be 0.1875 in., or the minimum nominal pipe wall thickness shall be 0.154 in..

(3) The minimum design metal temperature shall be no colder than −20°F.

(f) Unless specifically exempted in Fig. UCS-66, materials having a specified minimum yield strength greater than 65 ksi must be impact tested.

(g) Materials produced and impact tested in accordance with the requirements of the specifications listed in Table UG-84.3 are exempt from impact testing by the rules of this Division at minimum design metal temperatures not colder than the test temperature required by the specification [see Fig. UG-84.1, General Note (c)].

UCS-67 IMPACT TESTS OF WELDING PROCEDURES

For welded construction the Welding Procedure Qualification shall include impact tests of welds and heat affected zones (HAZ) made in accordance with UG-84 when required by the following provisions.

UCS-67(a) Welds made with filler metal shall be impact tested in accordance with UG-84 when any of the following apply:

UCS-67(a)(1) when either base metal is required to be impact tested by the rules of this Division; or

UCS-67(a)(2) when joining base metals from Table UG-84.3 or Fig. UCS-66 Curve C or D and the minimum design metal temperature is colder than −20°F but not colder than −50°F, unless welding consumables which have been classified by impact tests at a temperature not warmer than −50°F by the applicable SFA specification are used; or

UCS-67(a)(3) when joining base metals exempt from impact testing by UCS-66(g) when the minimum design metal temperature is colder than −50°F.

UCS-67(b) Welds in UCS materials made without the use of filler metal shall be impact tested when the thickness at the weld exceeds ½ in. for all minimum design metal temperatures or when the thickness at the design metal temperatures or when the thickness at the weld exceeds $\frac{5}{16}$ in. and the minimum design metal temperature is colder than 50°F.

UCS-67(c) Weld heat affected zones produced with or without the addition of filler metal shall be impact tested whenever any of the following apply:

UCS-67(c)(1) when the base metal is required to be impact tested by the rules of this Division; or

A95 *UCS-67(c)(2)* when the welds have any individual weld pass exceeding $\frac{1}{2}$ in. in thickness, and the minimum design metal temperature is colder than 70°F; or

UCS-67(c)(3) when joining base metals exempt from impact testing by UCS-66(g) when the minimum design metal temperature is colder than −50°F.

UCS-67(d) Vessel (production) impact tests in accordance with UG-84(i) may be waived for any of the following:

UCS-67(d)(1) weld metals joining steels exempted from impact testing by UCS-66 for minimum design metal temperatures of −20°F and warmer; or

UCS-67(d)(2) weld metals defined in (a)(2) and (a)(3) above; or

UCS-67(d)(3) heat affected zones (HAZ) in steels exempted from impact testing by UCS-66, except when (c)(3) above applies.

UCS-68 DESIGN[5]

A96 *UCS-68(a)* Welded joints shall comply with UW-2(b) when the minimum design metal temperature is colder than −50°F, unless the coincident Ratio defined in Fig. UCS-66.1 is less than 0.4.

A96 *UCS-68(b)* Welded joints shall be postweld heat treated in accordance with the requirements of UW-40 when required by other rules of this Division or when the minimum design metal temperature is colder than −50°F, and the coincident Ratio defined in Fig. UCS-66.1 is 0.4 or greater.

UCS-68(c) If postweld heat treating is performed when it is not otherwise a requirement of this Division, a 30°F reduction in impact testing exemption temperature may be given to the minimum permissible temperature from Fig. UCS-66 for P-No. 1 materials. The resulting exemption temperature may be colder than −50°F.

UCS-68(d) The allowable stress values to be used in design at the minimum design metal temperature shall not exceed those given in Section II, Part D, Tables 3 for bolting and 1A for other materials for temperatures of 100°F.

[5] No provisions of this paragraph waive other requirements of this Division, such as UW-2(a), UW-2(d), UW-10, and UCS-56.

FABRICATION

UCS-75 GENERAL

The rules in the following paragraphs apply specifically to the fabrication of pressure vessels and vessel parts that are constructed of carbon and low alloy steel and shall be used in conjunction with the general requirements for *Fabrication* in Subsection A, and with the specific requirements for *Fabrication* in Subsection B that pertain to the method of fabrication used.

UCS-79 FORMING SHELL SECTIONS AND HEADS

(a) The following provisions shall apply in addition to the general rules for forming given in UG-79.

(b) Carbon and low alloy steel plates shall not be formed cold by blows.

(c) Carbon and low alloy steel plates may be formed by blows at a forging temperature provided the blows do not objectionably deform the plate and it is subsequently postweld heat treated.

(d) Vessel shell sections, heads, and other pressure boundary parts of carbon and low alloy steel plates fabricated by cold forming shall be heat treated subsequently (see UCS-56) when the resulting extreme fiber elongation is more than 5% from the as-rolled condition and any of the following conditions exist.

(1) The vessel will contain lethal substances either liquid or gaseous (see UW-2).

(2) The material requires impact testing.

(3) The thickness of the part before cold forming exceeds $\frac{5}{8}$ in.

(4) The reduction by cold forming from the as-rolled thickness is more than 10%.

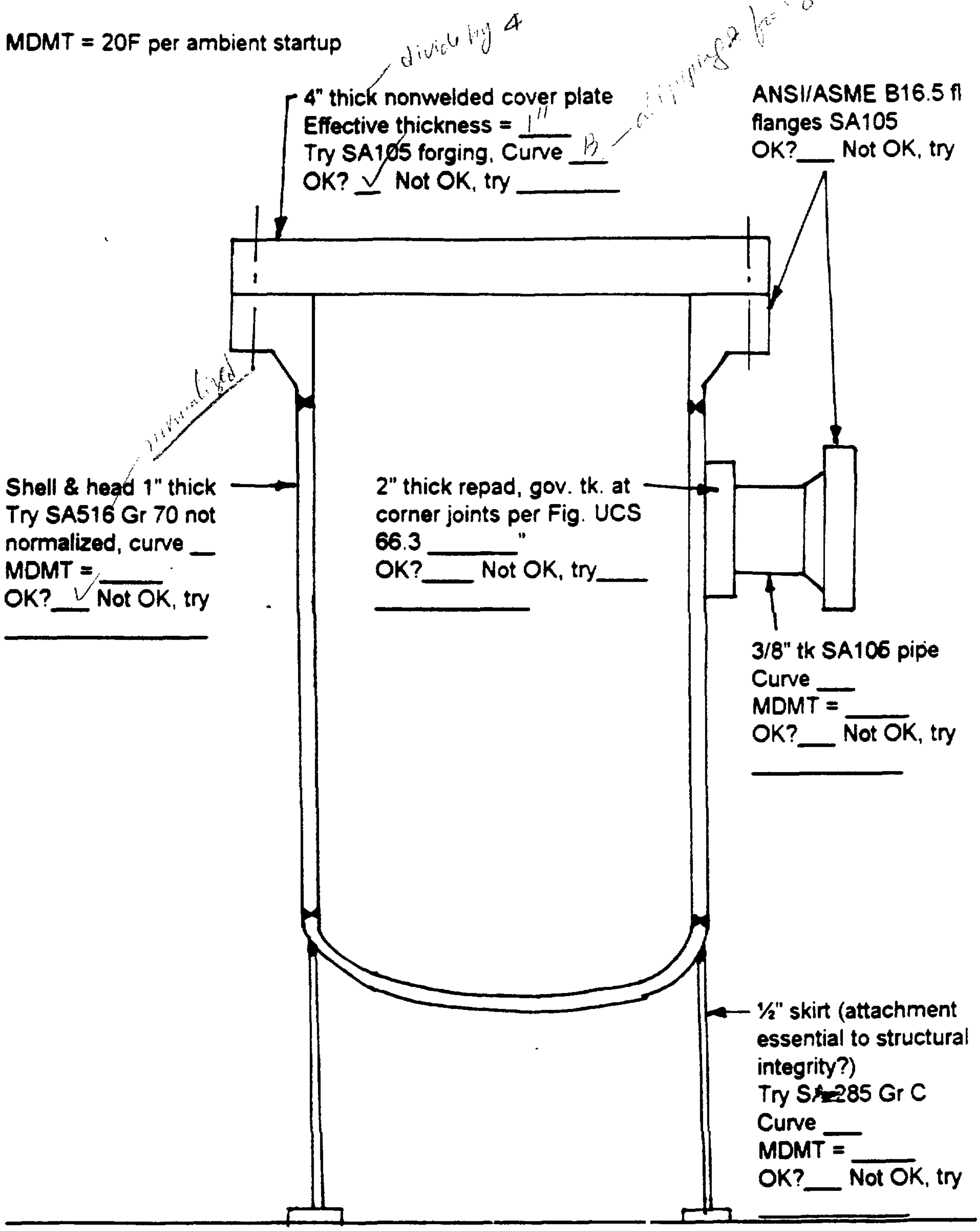
(5) The temperature of the material during forming is in the range of 250°F to 900°F.

For P-No. 1 Group Nos. 1 and 2 materials the extreme fiber elongation may be as great as 40% when none of the conditions listed above in (1) through (5) exist.

The extreme fiber elongation shall be determined by the following formulas:

TOUGHNESS EXERCISE

MDMT = 20F per ambient startup



AMERICAN SOCIETY OF MECHANICAL ENGINEERS

SHORT COURSE ON

ASME BOILER & PRESSURE VESSEL CODE

SECTION VIII, DIVISION 1
PRESSURE VESSELS

MATERIALS & DESIGN

NOTICE

All statements and text given by the author represent his opinion alone and do not necessarily represent the position of the American Society of Mechanical Engineers nor the ASME Boiler and Pressure Vessel Committee.

For official Interpretations of the ASME Code, write, phone, or FAX the Secretary of the ASME Boiler & Pressure Vessel Committee

Section VIII, Division 1

Pressure Vessels

GENERAL INFORMATION

Code Editions, Addenda, and Code Cases

A new Edition of the Code is issued every three years. The current edition is the 1998 Edition and the next edition will be the 2001 Edition. The issue date for the Edition is July 1st of the year of the Edition. Use of the Edition is **mandatory** at that date. Changes to the Code are given in an Addenda which is issued on July 1st of the year of the Addenda. The latest Addenda is **optional** thru Dec. 31st of the year of the Addenda and use is optional during that time. The Addenda becomes **mandatory** starting Jan. 1st of the next year. Each separate Addenda is issued on a different color paper to make identity easier.

The Edition and Addenda to which a pressure vessel is designed and constructed is set by the date of contract or of a letter of intent. If the latest Addenda has not become mandatory, the choice of whether to apply it may be decided by the contract. **For a contract signed today, the 1998 Edition, 1998 Addenda, and the 1999 Addenda are mandatory.**

A paid subscription to this Section of the Code includes a copy of one Edition including the first Addenda, two separate Addenda (2nd & 3rd years), and six issues of Interpretations (two each year).

Code Cases are issued following each meeting of the Main Committee after they have been subjected to the same approval procedure as those items being considered for the next Addenda. The purpose of the Code Cases is to expand the coverage of Code requirements by permitting usage before the item is available in the text or to provide requirements for materials and construction not covered by existing Code rules. Code Cases are numbered in sequential order and are **optional** to use. **Code Cases are not mandatory.** However, if the use of one is chosen, it must be used in its entirety. They are reviewed by the Committee for adoption into the Code text, if possible. Code Cases are automatically annulled after three years from the date of issuance unless they are extended or annulled previously by Main Committee voted action. Paid subscription to Code Cases is independent from paid subscriptions to Code Editions. The subscriber to Code Cases receives the latest Code Cases four time each year for three years for a total of twelve sets of Code Cases with each subscription.

Technical Inquiries

Guidance is given for submitting technical inquiries to the Committee for consideration. These inquiries include requests for additions, deletions, and revisions to the Code, additions of Code Cases, and issuance of Code Interpretations. Format for submitting an inquiry shall include the *Purpose, Background,* and may require a *Presentation* to the Committee.

Code Additions, Deletions, and Revisions

These requests shall provide *Proposed Revision, Statement of Need,* and *Background Information*.

Code Cases

These requests shall provide *Statement of Need* and *Background Information.*

Code Interpretation

When a user of the Code, as well as anyone else, has difficulty understanding some part of the Code text, the person may send a technical inquiry to the Code Committee for an **Interpretation**. These requests shall provide *Inquiry, Proposed Reply,* and *Background Information*. Within Section VIII, Interpretations are approved by two methods. The first method is consideration of the inquiry by a special interpretation group of four experts plus the subcommittee secretary. The group must vote unanimously on the proposed reply. If that method does not apply or the proposed reply did not receive a unanimous vote, the inquiry is reviewed by the appropriate subgroup or special working group and then considered for approval by the entire subcommittee at a regular meeting. Once the inquiry and reply are approved, an Interpretation in the form of a question and reply is sent to the inquirer informing that person of the results. In addition, the question and reply, with all identification of the inquirer and company being removed, is published in *Interpretations* which are issued to Code subscribers twice each year.

It is important that anyone sending a Code Technical Inquiry meet all of the requirements described in Appendix 16 of Section VIII, Division 1, so that action can be taken without delay. There are several reasons that an Inquiry and request for Interpretation will not be answered by the Committee. These are:

(1) Indefinite Question - no reference to a Code paragraph or to a Code rule or requirement is given;

(2) Semi-Commercial Question - doubt as to whether question is related to Code requirement or is asking for approval of a specific design or method;

(3) Approval of a Specific Design - inquirer wants approval of specific design or construction details and is asking for a statement that it complies with Code requirements; and

(4) Basis or Background of Code requirements - inquirer wants rationale or basis of Code requirements. This is not available to inquirer!

The details in Appendix 16 also contain methods for preparing correspondence for suggesting Code additions, revisions, and any other items to improve the book.

<u>Sending Inquiries and Other Correspondence to the Code Committee</u>

The general public has a simple method to request guidance from the committee or to request a revision or addition to the present rules. When the inquiry is regarding Section VIII and follows the requirements of Appendix 16, prepare and send a letter to:

Mr. Alan J. Roby
ASME International
Three Park Avenue
New York, NY 10016-5990

MATERIALS

General Requirements (UG-4)

Material subject to stress due to pressure used in the construction of pressure vessels and pressure parts shall conform to one of the specifications given in the ASME Boiler and Pressure Vessel Code, Section II, Materials, Parts A and B, and shall be limited to those materials listed in Subsection C of this Division or in a Code Case except as permitted by UG-9, UG-10, UG-11, UG-15, and the Mandatory Appendices. Nonpressure part material need not conform to an ASME material specification but shall be of weldable quality as described in an applicable Welding Procedure Specification (WPS). The User knows the service and should be sure that the materials listed in the design specifications or purchase order are suitable for the application considering items such as strength, corrosion, erosion, oxidation, and toughness.

Welding Materials (UG-9)

Welding materials shall either meet the requirements of Section II, Part C, or be described in a Welding Procedure Specification.

Materials Identified With or Produced to a Specification Not Permitted by This Division and Material Not Fully Identified (UG-10)

UG-10 provides rules to establish the identity of material for use in vessels and parts constructed to this Division. Rules are given for material which has its identification established and certified

TO A SPECIFICATION NOT PERMITTED BY THIS DIVISION OR PROCURED ONLY TO CHEMICAL COMPOSITION REQUIREMENTS MAY BE ACCEPTED AS SATISFYING THE REQUIREMENTS BY RECERTIFICATION BY AN ORGANIZATION OTHER THAN THE VESSEL OR PART MANUFACTURER OR BY THE VESSEL OR PART MANUFACTURER. IF THE MATERIAL TEST REPORT IS NOT AVAILABLE, ADDITIONAL TESTS MAY BE REQUIRED TO FULLY ESTABLISH THE IDENTITY OF THE MATERIAL TO AN ASME SPECIFICATION OR AN ACCEPTABLE ASTM SPECIFICATION. WHEN ALL REQUIREMENTS ARE MET, THE MATERIAL MAY BE MARKED ACCORDINGLY. MATERIAL WHICH IS NOT FULLY IDENTIFIED MAY BE ACCEPTED AS SATISFYING THE REQUIREMENTS BY TESTING AND IDENTIFICATION BY THE VESSEL OR PART MANUFACTURER ONLY. EACH PIECE OF MATERIAL REQUIRES IDENTIFICATION, NOT JUST THE LOT OR GROUP!

MISCELLANEOUS PRESSURE PARTS (UG-11)

RULES ARE GIVEN IN UG-11 FOR PREFABRICATED AND PREFORMED PRESSURE PARTS SUBJECTED TO INTERNAL OR EXTERNAL PRESSURE WHICH ARE FURNISHED BY A SHOP OTHER THAN THAT OF THE VESSEL MANUFACTURER THAT WILL CODE-STAMP THE VESSEL. CAST, FORGED, ROLLED, OR DIE-FORMED STANDARD AND NON-STANDARD PRESSURE PARTS AND WELDED STANDARD PRESSURE PARTS FOR USE OTHER THAN THE SHELL OR HEADS OF A VESSEL DO NOT REQUIRE A PARTIAL DATA REPORT NOR DOES THE MANUFACTURER OF THE PART HAVE TO HOLD A CERTIFICATION OF AUTHORIZATION. THE PART MANUFACTURER IS REQUIRED TO MARK THE PART WITH A NUMBER OR SYMBOL WHICH IDENTIFIES THE PART TO A MATERIAL SPECIFICATION, ESTABLISHES PRESSURE/TEMPERATURE RATINGS, AND GIVES OTHER PERTINENT DESIGN DATA IN THEIR PRODUCT CATALOG. THESE MISCELLANEOUS PRESSURE PARTS SHALL BE MADE TO EITHER AN ASME STANDARD AS LISTED IN UG-44 AND TABLE U-3 OR

TO A MANUFACTURER'S STANDARD. IF THE MISCELLANEOUS PRESSURE PART CONTAINS A WELD, IT SHALL BE CONSTRUCTED TO CODE REQUIREMENTS USING A CERTIFIED WELDER, AND A PARTIAL DATA REPORT SHALL BE GIVEN. IF THE PART REQUIRES RADIOGRAPHY OR OTHER NDE OR REQUIRES HEAT TREATMENT, THOSE OPERATIONS MAY BE DONE AT EITHER THE PARTS MANUFACTURER PLANT OF THE VESSEL MANUFACTURER'S PLANT.

PARTICULAR GRADE OF A MATERIAL SPECIFICATION NOT GIVEN (UG-15)

UG-15 PROVIDES RULES FOR ACCEPTANCE OF MATERIAL FOR WHICH THERE IS NO GRADE OF A PARTICULAR SPECIFICATION GIVEN IN THE LIST OF SUBSECTION C EVEN THOUGH THERE ARE OTHER GRADES OF THAT SPECIFICATION GIVEN. THE RULES REQUIRE THAT THE MATERIAL WITH NO GRADE GIVEN IN THE SPECIFICATION HAS TO BE SHOWN TO HAVE THE REQUIRED CHEMICAL AND PHYSICAL PROPERTIES, TOLERANCES, AND OTHER ENGINEERING DATA REQUIRED FOR THE GRADES WHICH ARE GIVEN. WHEN THIS IS SHOWN, THE GRADE NOT GIVEN IN SUBSECTION C MAY BE USED.

DESIGN

GENERAL DESIGN REQUIREMENTS

SECTION VIII, DIVISION 1, DOES NOT REQUIRE CERTIFIED DESIGN SPECIFICATIONS TO BE SUPPLIED BY THE USER. DESIGN REQUIREMENTS

SHALL BE GIVEN IN THE USER'S DESIGN SPECIFICATION OR PURCHASE ORDER. IT SHALL CONTAIN ALL INFORMATION REQUIRED TO SATISFY UG-22 INCLUDING DESIGN PRESSURE, DESIGN TEMPERATURE, AND MINIMUM DESIGN METAL TEMPERATURE WHICH WILL BE STAMPED ON THE VESSEL. THE VESSEL SHALL BE DESIGNED FOR AT LEAST THE MOST SEVERE CONDITION OF COINCIDENT PRESSURE AND TEMPERATURE AT NORMAL OPERATING CONDITIONS. THERE COULD BE MORE THAN ONE SET OF DESIGN CONDITIONS. IN ADDITION, OTHER INFORMATION SUCH AS INSTALLATION LOCATION FOR WIND AND/OR EARTHQUAKE LOADINGS, WEIGHT OF CONTENTS, LOCAL LOADINGS, TEMPERATURE GRADIENTS, AND ALL OTHER LOADINGS TO MEET UG-22 ARE REQUIRED.

GENERAL DESIGN AND CONSTRUCTION REQUIREMENTS, RULES, FORMULAS, AND CURVES ARE GIVEN IN SUBSECTION A. FABRICATION REQUIREMENTS FOR WELDING, BRAZING, AND FORGINGS ARE GIVEN IN SUBSECTION B. AND SUBSECTION C CONTAINS A LIST OF THOSE MATERIALS WHICH ARE PERMITTED. ALLOWABLE STRESS VALUES AND OTHER MATERIALS DESIGN DATA ARE GIVEN IN SECTION II, PART D.

MINIMUM THICKNESS OF SHELLS AND HEADS (UG-16(B))

EXCEPT FOR THE SPECIAL PROVISIONS GIVEN BELOW, THE MINIMUM THICKNESS OF SHELLS AND HEADS AFTER FORMING AND REGARDLESS OF PRODUCT FORM AND MATERIAL IS 1/16-IN. EXCLUSIVE OF CORROSION ALLOWANCE. PROVISIONS ARE:

(1) NOT APPLICABLE TO HEAT TRANSFER PLATES OF PLATE-TYPE HX;

(2) NOT APPLICABLE TO THE INNER PIPE OF DOUBLE-PIPE HEAT EXCHANGERS NOR TO TUBES IN TUBE-AND-SHELL HEAT EXCHANGERS WHERE EITHER THE PIPES OR TUBES ARE NPS 6 AND LESS;

(3) THE MINIMUM THICKNESS OF SHELLS AND HEADS OF AN UNFIRED STEAM BOILER IS 1/4-IN.;

(4) THE MINIMUM THICKNESS OF SHELLS AND HEADS USED IN COMPRESSED AIR, STEAM, AND WATER SERVICE MADE FROM MATERIALS LISTED IN TABLE UCS-23 IS 3/32-IN.

MANUFACTURING UNDERTOLERANCES (UG-16(c)&(d))

(1) PLATE MATERIAL MAY BE FURNISHED AND USED WITH A MILL UNDERTOLERANCE OF NOT MORE THAN THE SMALLER OF 0.01-IN. OR 6% OF THE ORDERED THICKNESS WITHOUT HAVING TO CONSIDER THE REDUCTION IN CALCULATIONS.

(2) PIPE AND TUBE MATERIAL MAY BE ORDERED BY NOMINAL THICKNESS; HOWEVER, THE MANUFACTURING UNDERTOLERANCE SHALL BE TAKEN INTO ACCOUNT.

METHODS OF FABRICATION IN COMBINATION (UG-17)

A VESSEL MAY BE DESIGNED AND CONSTRUCTED BY A COMBINATION OF METHODS OF FABRICATION PROVIDED THE RULES OF THE RESPECTIVE METHODS ARE MET. VESSELS SHALL BE DESIGNED AND CONSTRUCTED TO THE MOST RESTRICTIVE RULES OF FABRICATION AND OPERATION.

MATERIALS IN COMBINATION (UG-18)

A VESSEL MAY BE DESIGNED AND CONSTRUCTED OF ANY COMBINATION OF MATERIALS PERMITTED BY THIS SECTION PROVIDED ALL OF THE RULES FOR EACH MATERIAL ARE SATISFIED. PARTICULAR ATTENTION SHALL BE GIVEN TO DISSIMILAR WELDING, EXPANSION COEFFICIENTS, AND ELEVATED TEMPERATURE PROPERTIES.

Design Temperature

(a) Design & Maximum - The temperature used for selecting the allowable stress value for determining the minimum required thickness during normal operation shall be the mean metal temperature through the thickness. The temperature at any point shall not exceed the maximum temperature given at the highest allowable stress value for that material specification and grade or the maximum temperature on the external pressure chart, whichever is lower.

(b) Minimum - The minimum design metal temperature (MDMT) shall be the lowest temperature expected in service considering the lowest operating temperature, upset conditions, autorefrigeration, and any other source of cooling. MDMT is used to determine if impact testing is required according to UG-84. When all conditions of UG-20 are met, impact testing is not mandatory.

Design Pressure and Temperature

Vessels shall be designed for the most severe condition of coincident pressure and temperature expected in normal operation. A vessel may be designed for more than one combination of coincident pressure and temperature. The MDMT may be different for each of the pressure/temperature conditions. A margin should be provided above the operating pressure for pressure surges relative to the pressure relief device set pressure.

Design Loadings (UG-22)

Section VIII, Division 1, does not contain design requirements, procedures, rules, nor allowable stresses for many of the loadings given in UG-22. When any of these is not given, it is the responsibility of the Manufacturer to determine the acceptability of the vessel for that loading according to U-2(g).

A review of the loadings given in UG-22 and comments on each loading follow:

(a) internal and/or external design pressure. Design procedures and formulas are given for most common geometries. Allowable stresses are also included. For geometries not given, analysis is according to U-2(g).

(b) weight of vessel and contents including static head of liquids. The static head is required to determine the additional pressure loading which is added to the design pressure for setting the minimum required thickness from the design formulas. The static head of liquid is especially important in vertical vessels containing liquid with a high specific gravity and horizontal vessels with large diameters and low pressure. No specific design rules are given; however, these loadings of weight of vessel and contents generate primary stress which adds to the pressure stress.

(c) weight of attached equipment and parts such as that from motors, other vessels, piping, insulation, and linings. These loadings are required for determining both general and local effects. For example, part of the weight of piping to the first pipe support can be carried through a nozzle and the vessel; some multi-vessel arrangements may pass weight of vessels above into and through lower vessels; and motors, mixers, and other equipment may be supported on pressure components. All of these loadings cause both local and general stresses on the vessel. No design rules are given; consequently, U-2(g) applies.

(d) local loadings on pressure components such as those caused by internals (see Appendix D), lugs, saddles, support rings and legs, stiffening rings (see Appendix G), and other loadings noted above in (c), and by piping thermal expansion forces and moments. No design rules are given; consequently, U-2(g) applies. There are many good texts and papers containing design methods proposed by authors, such as Zick's design rules for horizontal vessels on two saddles and WRC Bulletins No. 107 and 297 for local loadings on shells and heads.

(e) cyclic and dynamic reactions due to pressure and/or thermal variations such as those caused by operational cycles or from equipment attached to or mounted on the vessel. No design rules are given; however, guidance may be obtained from cyclic design procedures given in Appendix 5 of Section VIII, Division 2 or other texts and papers.

(f) wind, earthquake, and snow loadings, and any other loadings from natural forces which cause transient loadings. No design rules are given. For guidance, Appendix L-2.1 contains an example problem of a vertical vessel with an overturning moment generated from an applied local loading such as that from an earthquake. UG-23(b) gives rules for determining allowable longitudinal compressive stresses in cylinders, and UG-23(d) describes the method to set allowable stresses when transient loadings, such as earthquake loading and wind loading, are applied. The magnitude and method for setting earthquake loading, wind loading, and other transient loadings are determined according to the applicable building code at the location where the vessel is to be installed. If the applicable building code can not be determined or there is none, the Manufacturer shall set the loadings and analysis methods according to company standards and policy which meet U-2(g). Local effects from these transient loadings shall also be considered.

(g) impact reactions such as those due to fluid shock (water hammer). No design rules are given. The allowable stress is usually increased from the basic Code allowable stress due to the short time of the applied loading. The usual values based on 0.2% offset yield strength which are used to set basic allowable stresses may not be applicable due to the short time duration of applying the loading. The effective yield strength for this short time duration yield strength may be two or three times greater than the 0.2% yield strength.

(h) temperature gradients and differential thermal expansion. This includes through-thickness and axial gradients and includes differences in thermal expansion growth in all directions due to differences in thermal expansion coefficients for different temperatures and materials. No design rules are given; however, thermal expansion coefficients and other data may be obtained from Section II, Part D.

As previously stated, no design rules, procedures, nor example problems exist for many of the loadings to be considered. As stated in U-2(g), when complete details of analysis are not given in the text, the Manufacturer, subject to acceptance of the Authorized Inspector, shall provide a design and construction which is as safe as one established by using the formulas, rules, and allowable stress values given in the text. The Manufacturer may use whatever sources, methods, and procedures that are necessary to determine the acceptability of each loading listed in UG-22.

<u>Allowable Tensile Stress Values</u>

The maximum allowable tensile stress values permitted for various materials listed in Subsection C are given in either the text or Subpart 1 of Section II, Part D. The basis for establishing the allowable tensile stress values is given in Appendix 1 of Section II, Part D, and is shown in Table P-1 of this text. For wrought or cast ferrous and nonferrous materials, except for the following:
bolting material with strength enhanced by heat treatment or strain

HARDENING, STRUCTURAL GRADES OF FERROUS PLATE, CAST IRON, AND CAST DUCTILE IRON, THE ALLOWABLE TENSILE STRESSES ARE BASED ON THE FOLLOWING CRITERIA:

(1) AT TEMPERATURES BELOW THE CREEP/RUPTURE RANGE, THE LOWEST OF:

(A) 1/3.5 OF THE SPECIFIED MINIMUM TENSILE STRENGTH;

(B) 1/3.5 OF THE TENSILE STRENGTH AT TEMPERATURE;

(C) 2/3 OF THE SPECIFIED MINIMUM YIELD STRENGTH;

(D) 2/3 OF THE YIELD STRENGTH AT TEMPERATURE.

IN ADDITION, FOR AUSTENITIC STAINLESS STEELS AND CERTAIN NICKEL AND NICKEL ALLOYS, EXCLUDING BOLTING, FLANGES, AND OTHER STRAIN SENSITIVE USEAGE WHERE SLIGHTLY GREATER DEFORMATION IS NOT OBJECTIONABLE, THE FACTOR ON THE YIELD STRENGTH AT TEMPERATURE MAY BE INCREASED FROM 2/3 TO 0.9. THIS GIVES ANOTHER, HIGHER, SET OF ALLOWABLE STRESSES FOR DESIGN.

(2) AT TEMPERATURES IN THE CREEP/RUPTURE RANGE, THE LOWEST OF:

(A) 1.0 OF THE AVERAGE STRESS TO PRODUCE A CREEP RATE OF 0.01% IN 1,000 HOURS (ASSUMED TO EQUAL 1% IN 100,000 HOURS);

(B) 0.8 OF THE MINIMUM STRESS TO PRODUCE RUPTURE AT THE END OF 100,000 HOURS;

(C) 0.67 OF THE AVERAGE STRESS TO PRODUCE RUPTURE AT THE END OF 100,000 HOURS.

FOR BOLTING WITH THE STRENGTH ENHANCED BY HEAT TREATMENT OR BY

STRAIN HARDENING, THE FOLLOWING SPECIAL LIMITS ARE APPLIED AT TEMPERATURES BELOW THE CREEP/RUPTURE RANGE REPLACING THOSE CRITERIA IN ITEM (1), ABOVE:

(A) 1/5 OF THE SPECIFIED MINIMUM TENSILE STRENGTH INSTEAD OF THE 1/3.5 VALUE USED ABOVE;

(B) 1/4 OF THE SPECIFIED MINIMUM YIELD STRENGTH INSTEAD OF THE 2/3 VALUE USED ABOVE.

FOR STRUCTURAL GRADES, THE FOLLOWING SPECIAL LIMITS SHALL BE APPLIED AT TEMPERATURES BELOW THE CREEP/RUPTURE RANGE WITH NO ALLOWABLE STRESSES BEING GIVEN IN THE CREEP/RUPTURE RANGE:

(A) 0.92 ADDITIONAL MULTIPLYING FACTOR APPLIED TO ALL ALLOWABLE STRESSES AND THE TENSILE STRENGTH VALUED USED NOT TO EXCEED 55 KSI REGARDLESS OF THE ACTUAL VALUE IN THE SPECIFICATION.

FOR CAST IRON, WHICH IS PERMITTED ONLY AT TEMPERATURES BELOW THE CREEP/RUPTURE RANGE, THE ALLOWABLE TENSILE STRESS VALUES ARE BASED ON THE LOWER OF THE FOLLOWING:

(A) 1/10 OF THE SPECIFIED MINIMUM TENSILE STRENGTH;

(B) 1/10 OF THE TENSILE STRENGTH AT TEMPERATURE.

FOR CAST DUCTILE IRON, WHICH IS PERMITTED ONLY AT TEMPERATURES BELOW THE CREEP/RUPTURE RANGE, THE ALLOWABLE TENSILE STRESS VALUES ARE BASED ON THE LOWEST OF THE FOLLOWING:

(A) 1/5 OF THE SPECIFIED MINIMUM TENSILE STRENGTH;

(B) 1/5 OF THE TENSILE STRENGTH AT TEMPERATURE;

(C) 2/3 OF THE SPECIFIED MINIMUM YIELD STRENGTH;

(D) 2/3 OF THE YIELD STRENGTH AT TEMPERATURE.

METHOD FOR DEVELOPMENT OF ALLOWABLE TENSILE STRESS VALUES

ALLOWABLE TENSILE STRESS VALUES FOR THE MATERIALS LISTED IN SUBSECTION C WERE DEVELOPED BY THE SUBCOMMITTEE ON MATERIALS (SC II) BY APPLYING VARIOUS FACTORS OF SAFETY TO DATA WHICH THE COMMITTEE HAS OBTAINED FROM TESTS CONDUCTED BY MATERIALS SUPPLIERS, TESTING LABORATORIES, AND OTHER SOURCES. THE ALLOWABLE TENSILE STRESS VALUES ARE ESTABLISHED BY CONSTRUCTING CURVES ALONG PLOTS OF THE LOWER BOUND VALUES WHICH ARE OBTAINED BY APPLYING THE CRITERIA.

FIG. 1-1 SHOWS HOW THIS METHOD IS APPLIED TO MATERIALS DATA FOR 18-8 TYPE 304H AUSTENITIC STAINLESS STEEL. ALLOWABLE TENSILE STRESS VALUES WERE DEVELOPED FOR TWO DIFFERENT CRITERIA USING THE SAME BASIC TENSILE, CREEP, AND RUPTURE DATA BUT USING TWO DIFFERENT FACTORS OF SAFETY ON THE YIELD STRENGTH DATA. THE FIRST COLUMN OF ALLOWABLE STRESSES IS BASED ON APPLYING 1/3.5 FACTOR ON TENSILE STRENGTH DATA AND 2/3 FACTOR ON YIELD STRENGTH DATA WITH THE STANDARD FACTORS APPLIED IN THE CREEP/RUPTURE RANGE. THE SECOND COLUMN OF THE TABLE IS BASED ON APPLYING 1/3.5 FACTOR ON THE TENSILE STRENGTH DATA AND 0.9 FACTOR ON THE YIELD STRENGTH DATA WITH THE CREEP/RUPTURE DATA BEING THE SAME. THE CHOICE OF WHICH SET OF VALUES TO USE IS BASED ON THE DESIGNER'S OPINION OF HOW MUCH DEFORMATION IS PERMITTED. THE HIGHER ALLOWABLE STRESS VALUES PERMIT MORE DEFORMATION TO OCCUR.

FIG. 1-1 ALLOWABLE TENSILE STRESS VALUES FOR 18-8 TYPE 304H STAINLESS STEEL FOR SECTION VIII, DIVISION 1

Temp. (°F)	Tens. Stgr. (ksi)	Yield Stgr. (ksi)	1/3.5 Spec. Min. T.S. (ksi)	1/3.5 T.S. (ksi)	2/3 Spec. Min. Y.S. (ksi)	2/3 Y.S. (ksi)	0.9 Y.S. (ksi)	0.8 Min. Rupt. 10^5Hr. (ksi)	Allowable Stresses Note (1) (ksi)	Allowable Stresses Note (2) (ksi)
100	75.0	30.0	21.4	21.4	(20.0)	20.0	27.0	.	20.0	20.0
200	71.0	25.1	21.4	20.3	(20.0)	16.7	22.6	-	16.7	20.0
300	66.0	22.5	21.4	(18.9)	20.0	15.0	20.3	-	15.0	18.9
400	64.4	20.8	21.4	(18.4)	20.0	13.8	18.7	-	13.8	18.4
500	63.5	19.4	21.4	18.1	20.0	12.9	(17.5)	-	12.9	17.5
600	63.5	18.3	21.4	18.1	20.0	12.1	(16.6)	-	12.1	16.6
650	63.5	18.0	-	18.1	-	12.0	(16.2)	-	12.0	16.2
700	63.5	17.7	-	18.1	-	11.8	(15.9)	-	11.8	15.9
750	63.1	17.3	-	18.0	-	11.5	(15.6)	-	11.5	15.6
800	62.7	16.9	-	17.9	-	11.2	(15.2)	-	11.2	15.2
850	61.9	16.6	-	17.7	-	11.0	(14.9)	-	11.0	14.9
900	61.0	16.3	-	17.4	-	10.9	(14.7)	25.5	10.9	14.7
950	59.4	16.0	-	17.0	-	10.6	(14.4)	20.0	10.6	14.4
1000	57.7	15.6	-	16.5	-	10.4	(14.1)	15.9	10.4	14.1
1050	55.2	15.2	-	15.8	-	10.1	13.6	(12.4)	10.1	12.4
1100	52.0	14.7	-	14.9	-	9.8	13.2	(9.8)	9.8	9.8
1150	48.8	14.4	-	13.9	-	9.6	13.0	(7.7)	7.7	7.7
1200	45.6	14.1	-	13.0	-	9.4	12.7	(6.1)	6.1	6.1

Notes:
(1) Underlined __ numbers are based on lowest of 1/3.5 Min.TS; 1/3.5 TS; 2/3 Min.YS; 2/3 YS; and 0.8 Min.Rupture.
(2) Bracketed ( ) numbers are based on lowest of 1/3.5 Min.TS; 1/3.5 TS; 2/3 Min.YS; 0.9 YS; and 0.8 Min.Rupture.
(3) All values are approximate and may vary in round-off from Sect. II-D

ALLOWABLE COMPRESSIVE STRESS

THE BASIS FOR ESTABLISHING ALLOWABLE COMPRESSIVE STRESSES IS GIVEN IN APPENDIX 3 OF SECTION II, PART D. THESE BASIS ARE:

(1) FOR CYLINDERS UNDER AXIAL LOADINGS, THE STRESS IS THE LOWEST OF:

(A) 1/4 CRITICAL BUCKLING STRESS PLUS 50% FACTOR FOR TOLERANCE;

(B) 1/2 SPEC. MINIMUM YIELD STRENGTH OR YIELD STRENGTH AT TEMP.;

(C) 100% AVERAGE STRESS TO PRODUCE CREEP RATE OF 1% IN 100,000 HR.;

(D) 100% ALLOWABLE STRESS IN TENSION AT DESIGN TEMPERATURE.

(2) FOR CYLINDERS & TUBES UNDER EXTERNAL PRESSURE, THE LOWEST OF:

(A) 1/3 CRITICAL BUCKLING STRESS PLUS 80% FACTOR FOR TOLERANCE;

(B) 1/3 SPEC. MINIMUM YIELD STRENGTH OR YIELD STRENGTH AT TEMP.;

(C) 67% AVERAGE STRESS TO PRODUCE CREEP RATE OF 1% IN 100,000 HR.;

(D) 100% ALLOWABLE STRESS IN TENSION AT DESIGN TEMPERATURE.

(3) FOR SPHERES & SPHERICAL PARTS OF HEADS UNDER EXTERNAL PRESSURE, THE LOWEST OF:

(A) 1/4 CRITICAL BUCKLING STRESS PLUS 60% FACTOR FOR TOLERANCE;

(B) 1/4 SPEC. MINIMUM YIELD STRENGTH OR YIELD STRENGTH AT TEMP.;

(C) 50% AVERAGE STRESS TO PRODUCE CREEP RATE OF 1% IN 100,000 HR.;

(D) 100% ALLOWABLE STRESS IN TENSION AT DESIGN TEMPERATURE.

Design Criteria

For internal pressure and other tensile loadings, the design criteria is inferred in UG-23(c). In general, the minimum wall thickness shall be such that the primary membrane stress due to the sustained loadings given in UG-22 do not exceed the allowable tensile stress value. For external pressure and other compressive loadings, the design criteria is that where buckling and yield strength are kept below acceptable limits by following design procedures in UG-23(b) and UG-28. Also, when short time transient loadings, such as earthquake and wind loads, are applied in addition to the sustained loadings, the allowable stress value for both tensile and compressive loadings may be increased by 1.2 times the allowable stress value given for the material. In addition, the sustained loadings shall not cause a combined primary membrane stress plus a primary bending stress which exceeds 1.5 times the allowable stress value in tension.

It is recognized that secondary stresses such as discontinuity stresses and high local stresses at notches exist in some areas of vessels designed and fabricated in accordance with Code rules. Insofar as practical, design rules and formulas have been written to keep such stresses at a safe level consistent with experience without having to do a detailed analysis to determine these secondary and peak stresses.

This means that the minimum required thickness of specific components, such as shells and heads, is established by design rules and formulas using the allowable stresses from the Code. Section VIII, Div. 1, requires a detailed stress analysis, proof test, or stress analysis report only when required by the contract specifications or by the Authorized Inspector to satisfy U-2(g). A report may be necessary to provide details of design and construction where no rules nor formulas exist within the text. If the vessel is subjected to cyclic operation or some other severe service requirements or has a complex geometry not covered by the text, additional analysis will be required.

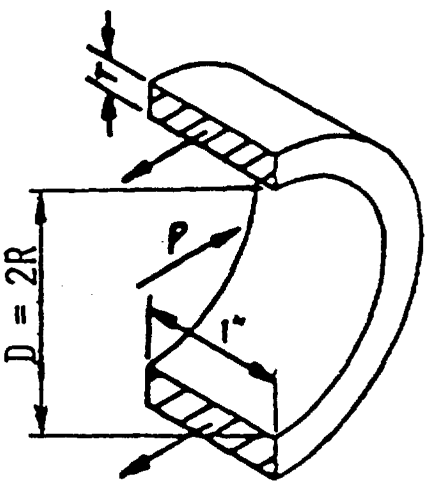
Strength Theory

The strength theory used for Section VIII, Div. 1, for internal pressure and other tensile loadings is the maximum stress theory. This theory considers the stress in each direction independently from the other directions. This theory is easier to use than other, more complex, theories; and it is acceptable with the factors of safety which are used in Section VIII, Div. 1, to set allowable stress values.

DESIGN FORMULAS FOR INTERNAL PRESSURE AND TENSILE LOADINGS

CIRCUMFERENTIAL OR HOOP STRESS IN CYLINDRICAL SHELLS

IN ORDER TO MAINTAIN STATIC EQUILIBRIUM FORCES IN THE HOOP DIRECTION, THE INTERNAL PRESSURE FORCE EXERTED AGAINST THE INSIDE WALL MUST BE RESISTED BY THE STRENGTH OF THE METAL IN THE WALL THICKNESS FOR A SEAMLESS SHELL AND BY THE LONGITUDINAL WELD JOINTS IN A WELDED SHELL. DEVELOPMENT OF THE HOOP STRESS FORMULAS IN UG-27(c)(1) FOLLOWS. ASSUMING A THIN-WALL VESSEL, FOR EACH LENGTH OF SHELL OR LONGITUDINAL WELD (ASSUME 1-IN.), THE INTERNAL PRESSURE EXERTS A FORCE EQUAL TO (P)(2R)(1-IN.) AND THE WALL EXERTS A RESISTING FORCE AT THE TWO CROSS-SECTIONS OF 2(t)(1-in.)(S). WHEN THESE TWO FORCES ARE EQUATED, 2PR = 2tS AND SOLVING FOR t = PR/S. IF A BUTT-WELD JOINT EFFICIENCY, QUALITY FACTOR, OR LIGAMENT EFFICIENCY IS INCLUDED TO MODIFY THE ALLOWABLE STRESS, THE FORMULA BECOMES:



$$t = \frac{PR}{SE} \text{ OR } P = \frac{SEt}{R} \text{ WHERE } E = \text{LOWEST EFFICIENCY OR JOINT FACTOR}$$

THIS THIN-WALL OR MEMBRANE FORMULA WAS USED IN THE CODE UNTIL THE 1943 EDITION AT WHICH TIME IT WAS MODIFIED TO MORE ACCURATELY DETERMINE RESULTS FOR THICKER WALLS DUE TO HIGHER PRESSURES AND/

OR HIGHER TEMPERATURES. THE FORMULAS WERE CHANGED AT THAT TIME TO THE ONES WHICH ARE PRESENTLY IN UG-27(c)(1) OF THE LATEST EDITION OF DIVISION 1 AS FOLLOWS:

$$t = \frac{PR}{SE - 0.6P} \quad \text{AND} \quad P = \frac{SEt}{R + 0.6t}$$

THESE FORMULAS ARE LIMITED TO A THICKNESS NOT TO EXCEED ONE-HALF OF THE INSIDE RADIUS AND TO A PRESSURE NOT TO EXCEED 0.385 SE. WHEN EITHER LIMIT IS EXCEEDED, THE THICK-WALL OR LAMÉ EQUATION GIVEN IN 1-2(A) SHALL BE USED. THESE FORMULAS ARE:

$$t = R(Z^{\frac{1}{2}} - 1) = R_o\left(\frac{Z^{\frac{1}{2}} - 1}{Z^{\frac{1}{2}}}\right) \quad \text{WHERE } Z = \frac{SE + P}{SE - P} \quad \text{AND}$$

$$P = SE\,\frac{(Z - 1)}{(Z + 1)} \quad \text{WHERE } Z = \left(\frac{R + t}{R}\right)^2 = \left(\frac{R_o}{R}\right)^2 = \left(\frac{R_o}{R_o - t}\right)^2$$

IN TERMS OF THE OUTSIDE RADIUS, R_o, THE CODE FORMULAS ARE GIVEN IN 1-1(A)(1) AS FOLLOWS:

$$t = \frac{PR_o}{SE + 0.4P} \quad \text{AND} \quad P = \frac{SEt}{R_o - 0.4t}$$

A COMPARISON OF FORMULAS FOR HOOP STRESS FOR CYLINDRICAL SHELLS FOR MEMBRANE (THIN-WALL), CODE (MODIFIED MEMBRANE), AND LAMÉ (EXACT, THICK-WALL) IS SHOWN IN FIGURE 2-1.

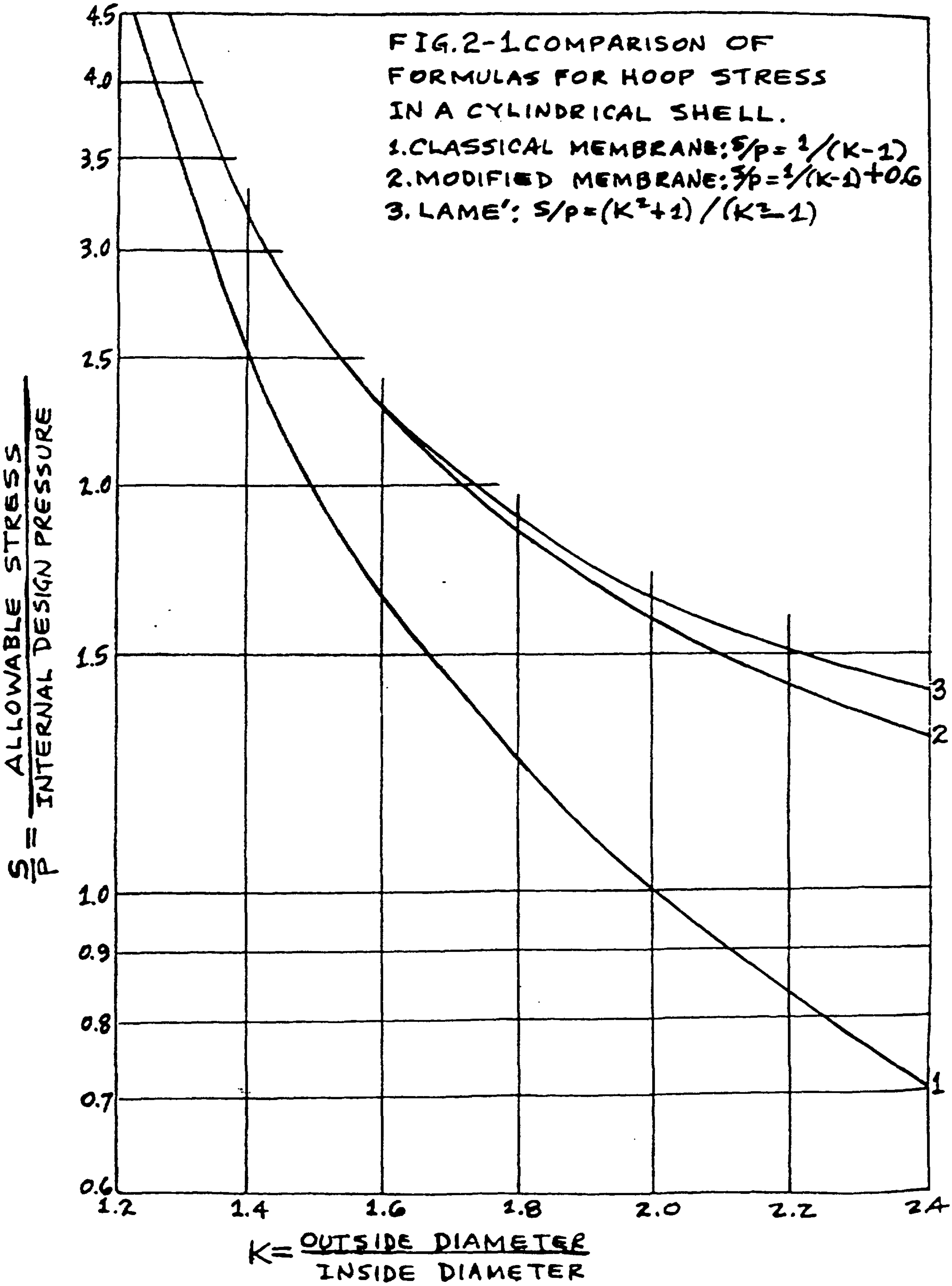
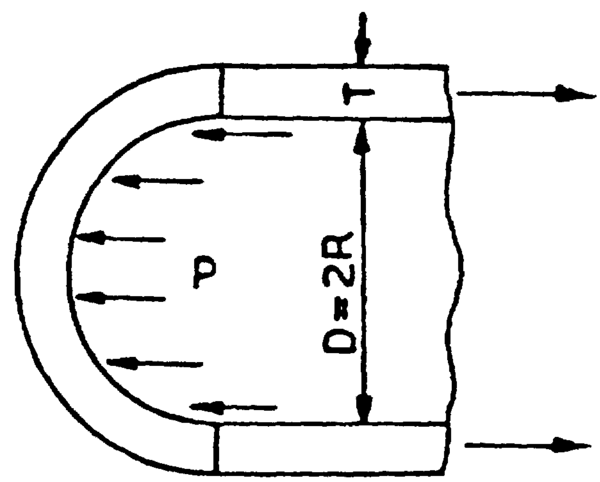


FIG. 2-1 COMPARISON OF FORMULAS FOR HOOP STRESS IN A CYLINDRICAL SHELL.

1. CLASSICAL MEMBRANE: $S/p = 1/(K-1)$
2. MODIFIED MEMBRANE: $S/p = 1/(K-1) + 0.6$
3. LAME': $S/p = (K^2+1)/(K^2-1)$

Longitudinal or Axial Stress in Cylindrical Shells

The stress formulas for the longitudinal or axial direction are developed in the same way as the hoop formulas. In order to maintain equilibrium in the axial direction, the internal force exerted against the vessel end closure must be resisted by the strength of the metal in the cross-sectional wall of the vessel for seamless shells and by the circumferential weld joints in a welded shell. The pressure force equals $(\pi/4)D^2P$, while the resisting force equals $(\pi)DtS$. When the two are equated, $(\pi/4)D^2P = (\pi)DtS$ and solving for $t = PD/4S = PR/2S$. If a joint efficiency is included for the circumferential weld to modify the allowable stress, the formula becomes:



$$t = \frac{PR}{2SE} \text{ or } P = \frac{2SEt}{R}$$ where E = circumferential weld joint eff.

When the longitudinal stress formula was added in 1957, the modified version was used. These are the same formulas which are presently in UG-27(c)(2) of the latest edition as follows:

$$t = \frac{PR}{2SE + 0.4P} \quad \text{or} \quad P = \frac{2SEt}{R - 0.4t}$$

These formulas are limited to thickness not to exceed one-half

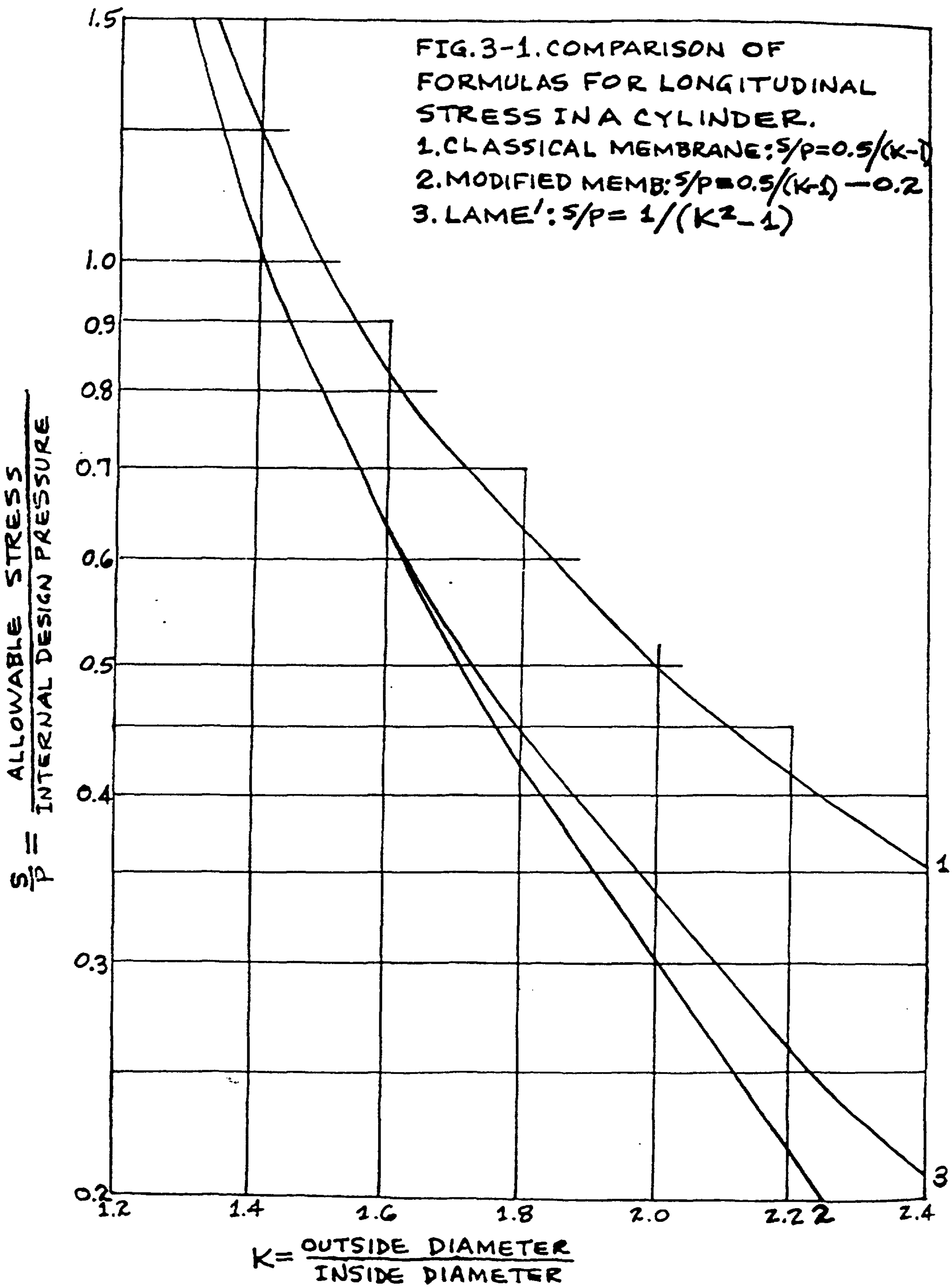


FIG. 3-1. COMPARISON OF FORMULAS FOR LONGITUDINAL STRESS IN A CYLINDER.
1. CLASSICAL MEMBRANE: S/P=0.5/(K-1)
2. MODIFIED MEMB: S/P=0.5/(K-1) —0.2
3. LAME': S/P= 1/(K²-1)
1.5
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
1.2
1.4
1.6
1.8
2.0
2.2
2.4
1
2
3
S/P = ALLOWABLE STRESS / INTERNAL DESIGN PRESSURE
K= OUTSIDE DIAMETER / INSIDE DIAMETER

OF THE INSIDE RADIUS AND TO A PRESSURE NOT TO EXCEED 1.25 SE. WHEN THESE LIMITS ARE EXCEEDED, THE FORMULAS OF 1-2(A)(2) SHALL BE USED. THESE THICK-WALL OR LAMÉ EQUATIONS ARE:

$$T = R(Z^{\frac{1}{2}} - 1) \text{ WHERE } Z = \left(\frac{P}{SE} + 1\right) \qquad \text{AND}$$

$$P = SE(Z - 1) \text{ WHERE } Z = \left(\frac{R + T}{R}\right)^2 = \left(\frac{R_O}{R}\right)^2 = \left(\frac{R_O}{R_O - T}\right)^2$$

A COMPARISON OF FORMULAS FOR AXIAL STRESS FOR CYLINDRICAL SHELLS FOR MEMBRANE, CODE, AND LAMÉ IS SHOWN IN FIGURE 3-1.

<u>MERIDIONAL STRESS IN SPHERICAL SHELLS</u>

THE MEMBRANE FORMULA FOR SPHERICAL SHELLS IS THE SAME AS THAT FOR THE LONGITUDINAL STRESS IN A CYLINDRICAL SHELL. WHEN A WELD JOINT EFFICIENCY IS INCLUDED, THE FORMULA IS:

$$T = \frac{PR}{2SE} \quad \text{OR} \quad P = \frac{2SET}{R} \text{ WHERE } E = \text{WELD JOINT EFFICIENCY}$$

WHEN THE 1950 EDITION OF SECTION VIII WAS ISSUED, THE SPHERICAL SHELL EQUATION WAS ALSO MODIFIED TO THAT IN THE UG-27(D) OF THE LATEST EDITION OF DIVISION 1 AS FOLLOWS:

$$T = \frac{PR}{2SE - 0.2P} \quad \text{AND } P = \frac{2SET}{R + 0.2T}$$

THESE FORMULAS ARE LIMITED TO A THICKNESS NOT TO EXCEED 0.356R AND TO A PRESSURE NOT TO EXCEED 0.665 SE. WHEN THESE LIMITS ARE EXCEEDED, THE THICK-WALL OR LAMÉ EQUATIONS OF 1-3 SHALL BE USED. THESE THICK-WALL EQUATIONS ARE:

$$t = R(Y^{1/3} - 1) = R_o\left(\frac{Y^{1/3} - 1}{Y^{1/3}}\right) \text{ where } Y = \frac{2(SE + P)}{2SE - P} \text{ and}$$

$$P = 2SE\left(\frac{Y - 1}{Y + 2}\right) \text{ where } Y = \left(\frac{R + t}{R}\right)^3 = \left(\frac{R_o}{R_o - t}\right)^3$$

In terms of the outside radius, R_o, the Code formulas are given in 1-1(a)(2) as follows:

$$t = \frac{PR_o}{2SE + 0.8P} \text{ and } P = \frac{2SEt}{R_o - 0.8t}$$

A comparison of formulas for stress in a spherical shell for membrane, Code, and Lamé is shown in Figure 4-1.

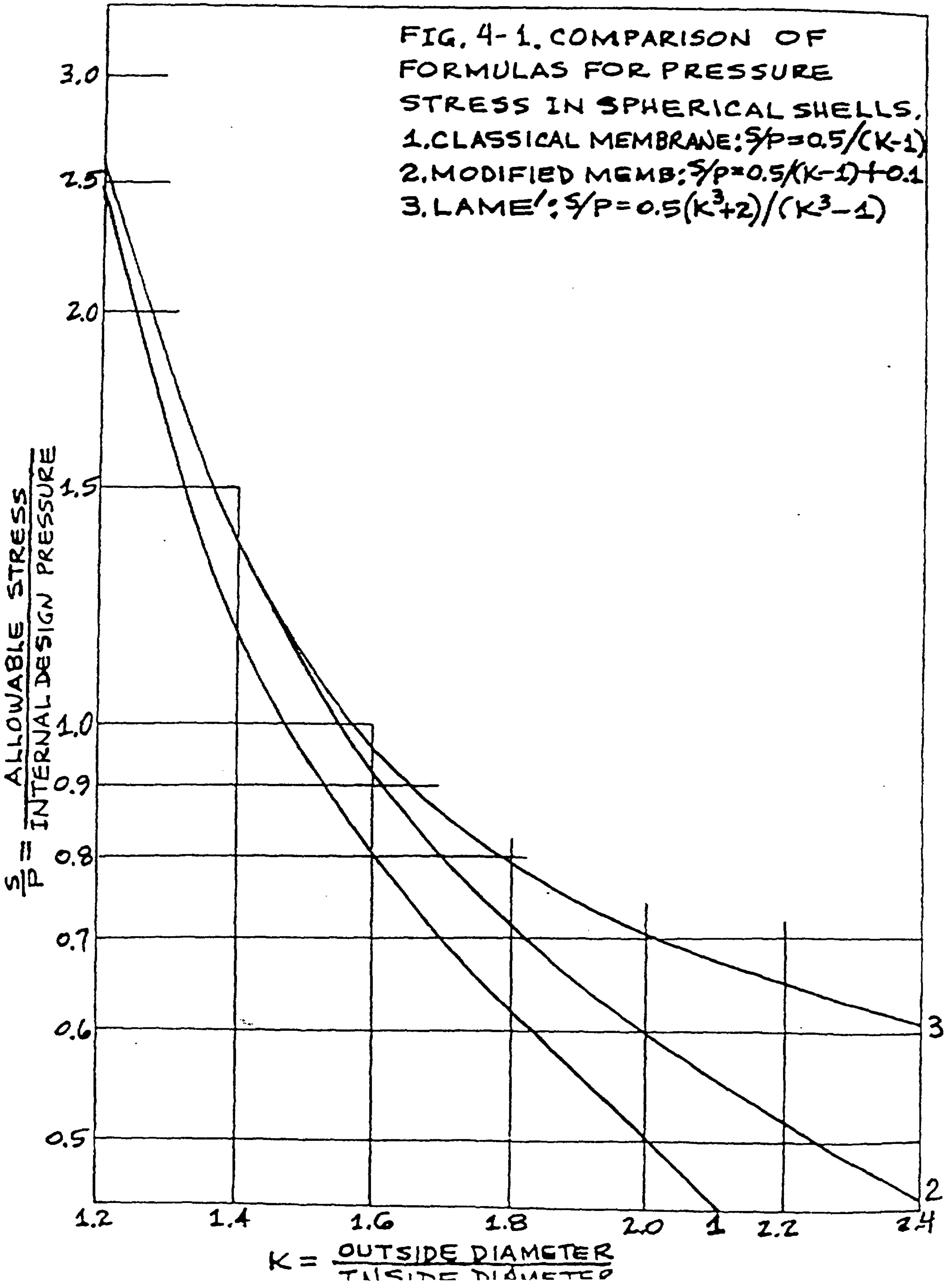


FIG. 4-1. COMPARISON OF FORMULAS FOR PRESSURE STRESS IN SPHERICAL SHELLS.

1. CLASSICAL MEMBRANE: $S/P = 0.5/(K-1)$
2. MODIFIED MEMB: $S/P = 0.5/(K-1) + 0.1$
3. LAME': $S/P = 0.5(K^3+2)/(K^3-1)$

Formed Heads Under Internal Pressure (Pressure on Concave Side)

For formed heads with pressure on the concave side (see Fig. 5-1), the formulas for minimum required thickness and maximum allowable working pressure are given in UG-32 and in 1-4. Formulas for the more commonly used geometries of formed heads based on inside dimensions are given in UG-32 and formulas for general geometries of formed heads based on either inside or outside dimensions are given in 1-4.

Ellipsoidal Heads

For 2:1 ellipsoidal heads, the formulas in UG-32(d) are:

$$t = \frac{PD}{2SE - 0.2P} \quad \text{and} \quad P = \frac{2SEt}{D + 0.2t}$$

For general geometries, the formulas in 1-4(c) are:

$$t = \frac{PDK}{2SE - 0.2P} \quad \text{and} \quad P = \frac{2SEt}{KD + 0.2t} \quad \text{where } K = \frac{1}{6}\left[2 + \left(\frac{D}{2h}\right)^2\right]$$

K = 1.0 for 2:1 ellipsoidal heads. Values of K are given in Table 1-4.1. When K > 1.0, special rules apply to the allowable stress values if the specified minimum UTS exceeds 80 ksi, see Footnote 1 of 1-4.

Torispherical Heads

For 6% knuckle heads, the formulas in UG-32(e) are:

$$t = \frac{0.885PL}{SE - 0.1P} \quad \text{and} \quad P = \frac{SEt}{0.885L + 0.1t}$$

For general geometries, the formulas in 1-4(d) are:

$$t = \frac{PLM}{2SE - 0.2P} \quad \text{and} \quad P = \frac{2SEt}{LM + 0.2t} \quad \text{where } M = \frac{1}{4}\left[3 + \sqrt{\frac{L}{r}}\right]$$

M = 1.77 for a 6% knuckle radius head. Values of M are given in Table 1-4.2 of the text. Special rules apply to the allowable stress values when the specified minimum UTS exceeds 80 ksi (see Footnote 1 of 1-4).

Hemispherical Heads

The formulas for hemispherical heads in UG-32(f) are the same as for spherical shells and are:

$$t = \frac{PL}{2SE - 0.2P} \quad \text{and} \quad P = \frac{2SEt}{L + 0.2t}$$

Conical Heads and Sections (Without a Transition Knuckle)

For conical heads and conical shell sections without a transition knuckle, which have a half-apex angle, α, not greater than 30°, the formulas in UG-32(g) are:

$$t = \frac{PD}{2\cos\alpha(SE - 0.6P)} \quad \text{and} \quad P = \frac{2SEt\cos\alpha}{D + 1.2t\cos\alpha}$$

A reinforcing ring shall be provided when required by the rules of 1-5(d) and (e).

For conical heads and sections without a transition knuckle, which have a half-apex angle, α, greater than 30°, see either 1-5(g) for special analysis or Code Case 2150, if applicable.

WHEN THE HALF-APEX ANGLE, α , EXCEEDS 30^0 AND WHEN RADIOGRAPHY OF THE CIRCUMFERENTIAL JOINT IS NOT REQUIRED, CODE CASE 2150 MAY BE USED WITHOUT HAVING TO COMPLY WITH THE SPECIAL ANALYSIS RULES IN 1-5(G). THE CODE CASE IS LIMITED TO HALF-APEX ANGLE, α, NOT GREATER THAN 60^0 AND THE AXIAL FORCES COME SOLELY FROM INTERNAL PRESSURE AND NOT FROM EXTERNAL LOADS SUCH AS WIND LOAD.

TORICONICAL HEADS

FOR CONICAL HEADS AND CONICAL SHELL SECTIONS WITH A TRANSITION KNUCKLE WHICH HAS A MINIMUM RADIUS OF NO LESS THAN THE LARGER OF 6% D_O OR 3T, THE CONICAL PORTION SHALL BE DETERMINED THE SAME AS FOR A CONICAL HEAD WITHOUT A TRANSITION WITH D_I IN PLACE OF D WHERE $D_I = D - 2R(1 - \cos\alpha)$. THE THICKNESS OF THE KNUCKLE IS DETERMINED BY FORMULA (3) OF 1-4(D) WHERE $L = D_I/2\cos\alpha$. TORICONICAL HEADS AND SECTIONS MAY BE USED WHEN α IS EQUAL TO OR LESS THAN 30^0 AND ARE MANDATORY WHEN α IS GREATER THAN 30^0 UNLESS THE DESIGN COMPLIES WITH 1-5(G) OR CODE CASE 2150.

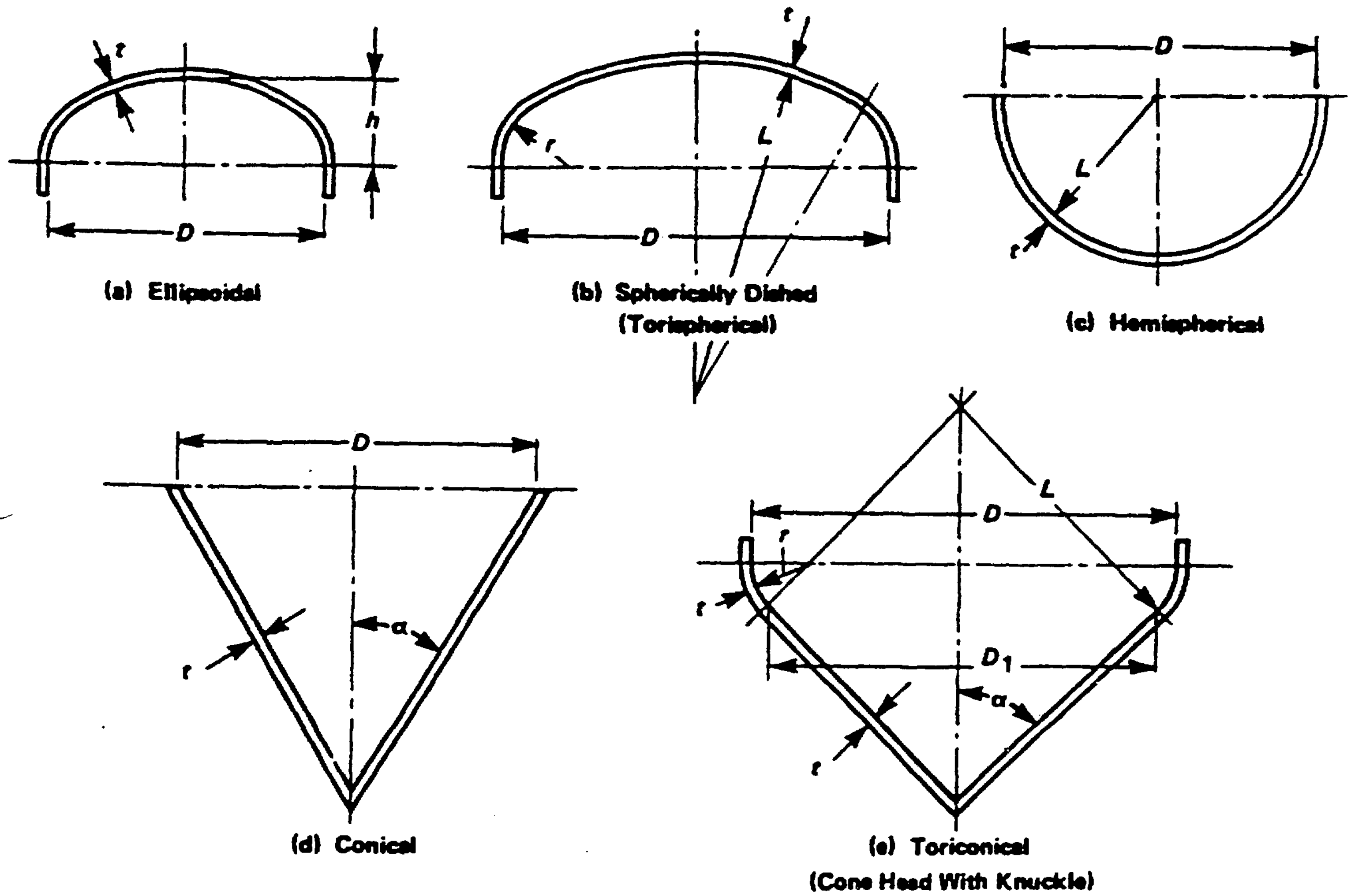


FIG. 5-1 PRINCIPAL DIMENSIONS OF TYPICAL HEADS

Unstayed Flat Heads, Covers, and Blind Flanges (UG-34)

Before designing an unstayed flat head or cover using the equations of UG-34, review the standard blind flanges listed in Table U-3 to determine if there is an acceptable blind flange for which no calculations need be made.

The equations in UG-34 are used to calculate the minimum required thicknesses of unstayed flat heads and covers as shown in Fig. 6-1. In all formulas, the value of C varies depending upon the corner details, weld details, and whether the cover is integral or bolted. For circular heads which are integral (no bolts), the equation is:

$$t = d\sqrt{CP/SE} \qquad (1)$$

For circular heads attached with bolts, the equation is:

$$t = d\sqrt{CP/SE + 1.9Wh_G/SEd^3} \qquad (2)$$

For square and other noncircular heads which are integral (no bolts), the equation is:

$$t = d\sqrt{ZCP/SE} \quad \text{where } Z = 3.4 - 2.4(d/D) = 2.5 \qquad (3)$$

For square and other noncircular heads attached with bolts, the equation is:

$$t = d\sqrt{ZCP/SE + 6Wh_G/SELd^2} \qquad (4)$$

In all cases, E is the joint efficiency of any Category A butt-weld joint within the head only. It is not the joint efficiency of the head-to-shell joint which is a corner joint.

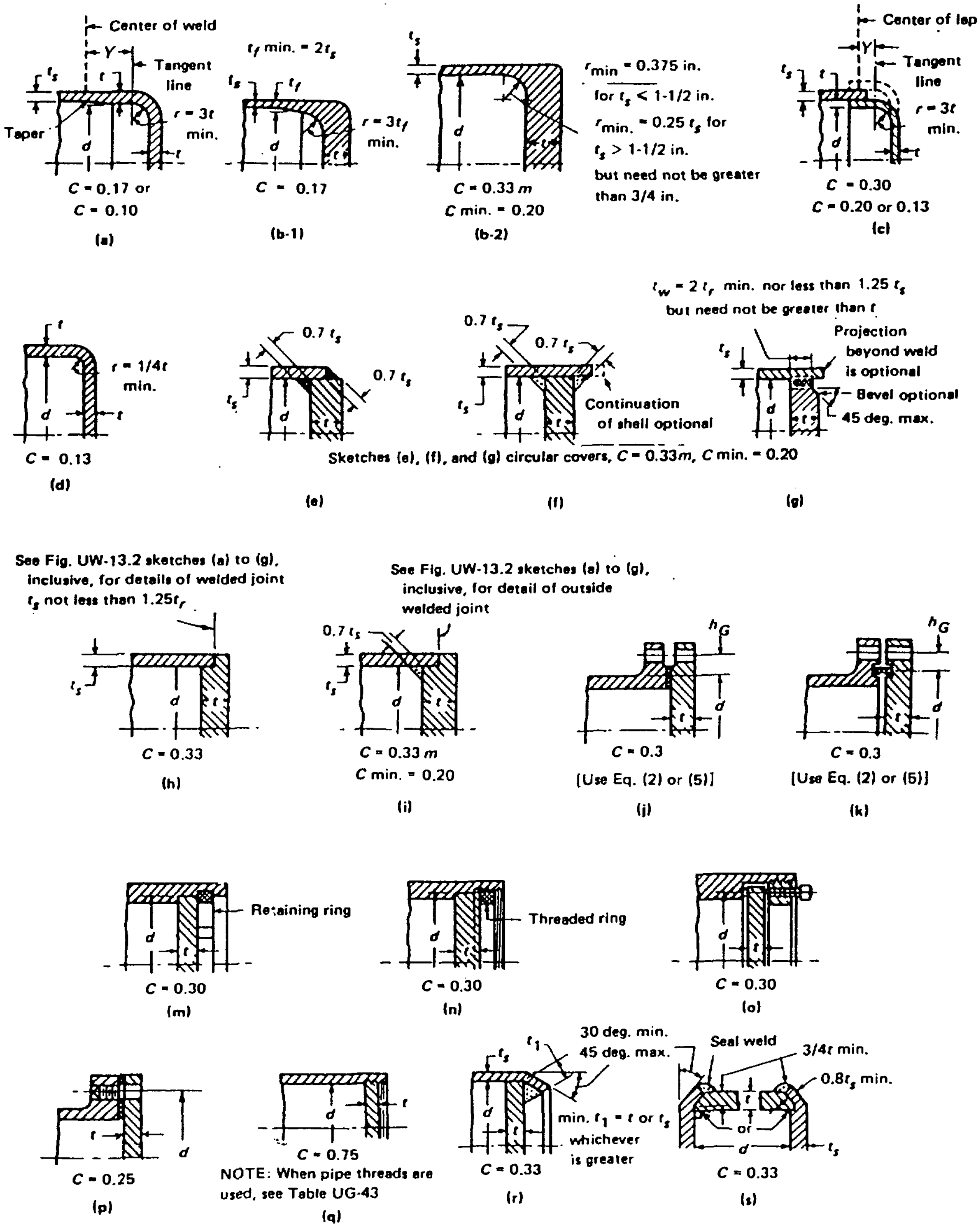


FIG. 6-1 UNSTAYED FLAT HEADS, COVERS, & BLIND FLANGES

SPHERICALLY-DISHED HEADS (UG-35(A) AND 1-6) (SEE FIG. 7-1)

DESIGN RULES FOR SPHERICALLY-DISHED HEADS WITH BOLTING FLANGES ARE GIVEN IN 1-6. THEY ARE APPLICABLE TO BOTH INTERNAL AND EXTERNAL PRESSURE.

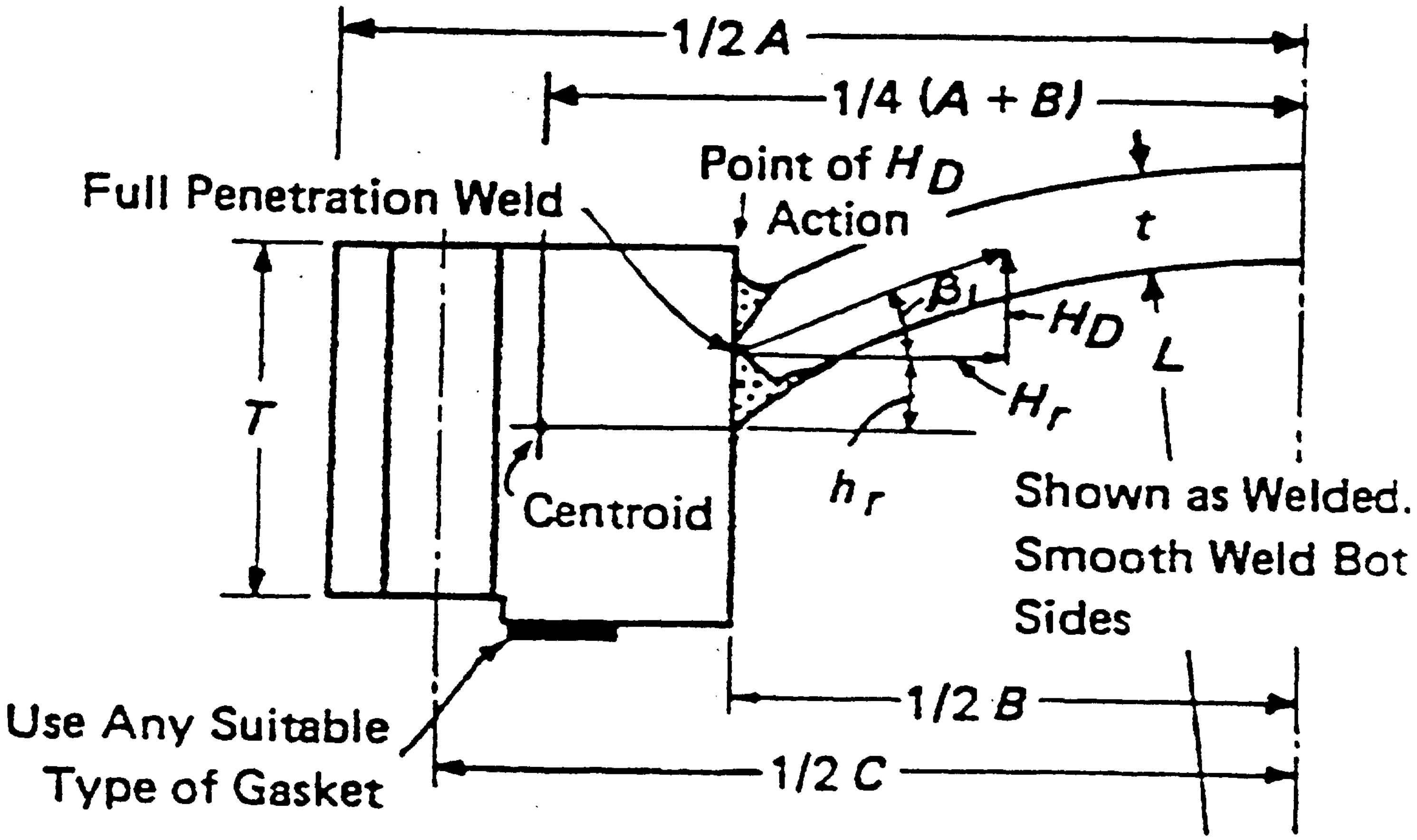


FIG. 7-1 SPHERICALLY-DISHED HEAD

DESIGN PROCEDURES FOR SHELLS UNDER EXTERNAL PRESSURE AND AXIAL COMPRESSION OF A CYLINDRICAL SHELL

THE DESIGN BASIS FOR EXTERNAL PRESSURE OF CYLINDRICAL AND SPHERICAL SHELLS, AS APPLIED IN UG-28, AND FOR AXIAL (LONGITUDINAL) COMPRESSION OF CYLINDRICAL SHELLS, AS APPLIED IN UG-23(c), IS GIVEN IN APPENDIX 3 OF SECTION II, PART D. FOR A CYLINDRICAL SHELL, THE LENGTH AND/OR THICKNESS ARE VARIED TO CONTROL THE BUCKLING AND YIELDING. STIFFENING RINGS MAY BE USED TO DECREASE THE BUCKLING LENGTH. IF STIFFENING RINGS ARE USED, THEY ARE DESIGNED ACCORDING TO UG-29 AND ARE ATTACHED ACCORDING TO UG-30. FOR SPHERICAL VESSELS, ONLY THE WALL THICKNESS CAN BE INCREASED TO PROVIDE AN ADEQUATE DESIGN. REFERENCE TO THE APPROPRIATE EXTERNAL PRESSURE CHART TO USE WITH EACH MATERIAL SPECIFICATION IS GIVEN IN THE TABLES OF ALLOWABLE TENSILE STRESS VALUES. THE EXTERNAL PRESSURE CHARTS AND THE PLOTTING DATA ARE GIVEN IN SUBPART 3 OF SECTION II, PART D.

BOTH THE MAXIMUM ALLOWABLE EXTERNAL PRESSURE AND THE MAXIMUM ALLOWABLE LONGITUDINAL COMPRESSIVE STRESS ARE DETERMINED BY A "TRIAL-AND-ERROR" BASIS WHERE THE THICKNESS, T, IS ASSUMED FOR BOTH CALCULATIONS AND THE BUCKLING LENGTH, L, IS ASSUMED FOR THE EXTERNAL PRESSURE CALCULATIONS, SEE FIG. 8-1. THIS PROCEDURE IS REPEATED WITH DIFFERENT VALUES OF T AND/OR L UNTIL THE DESIRED MAXIMUM ALLOWABLE EXTERNAL PRESSURE OR MAXIMUM ALLOWABLE LONGITUDINAL COMPRESSIVE STRESS IS OBTAINED. SEE FIG. 9-1 FOR LINES OF SUPPORT FOR CYLINDRICAL SHELLS UNDER EXTERNAL PRESSURE.

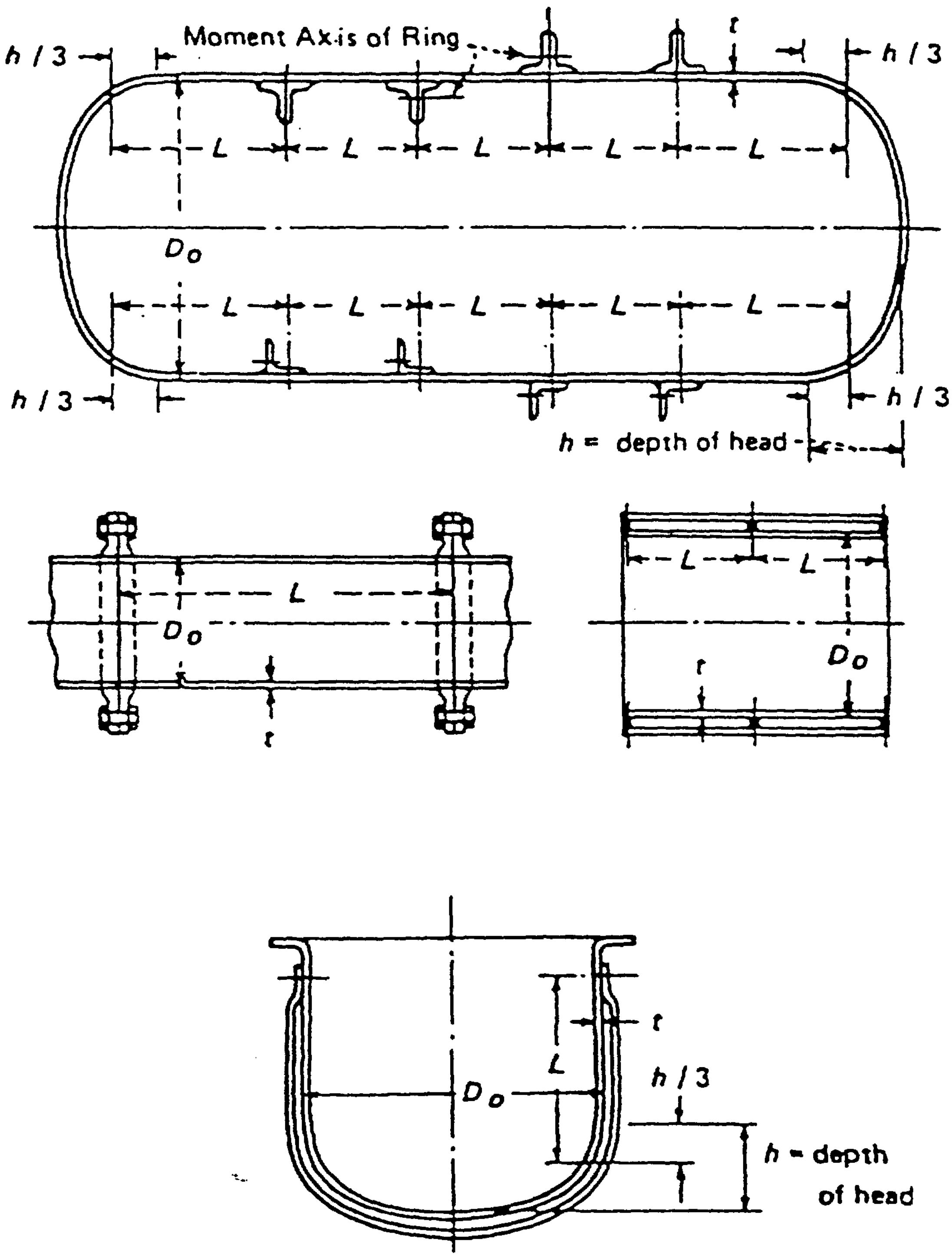
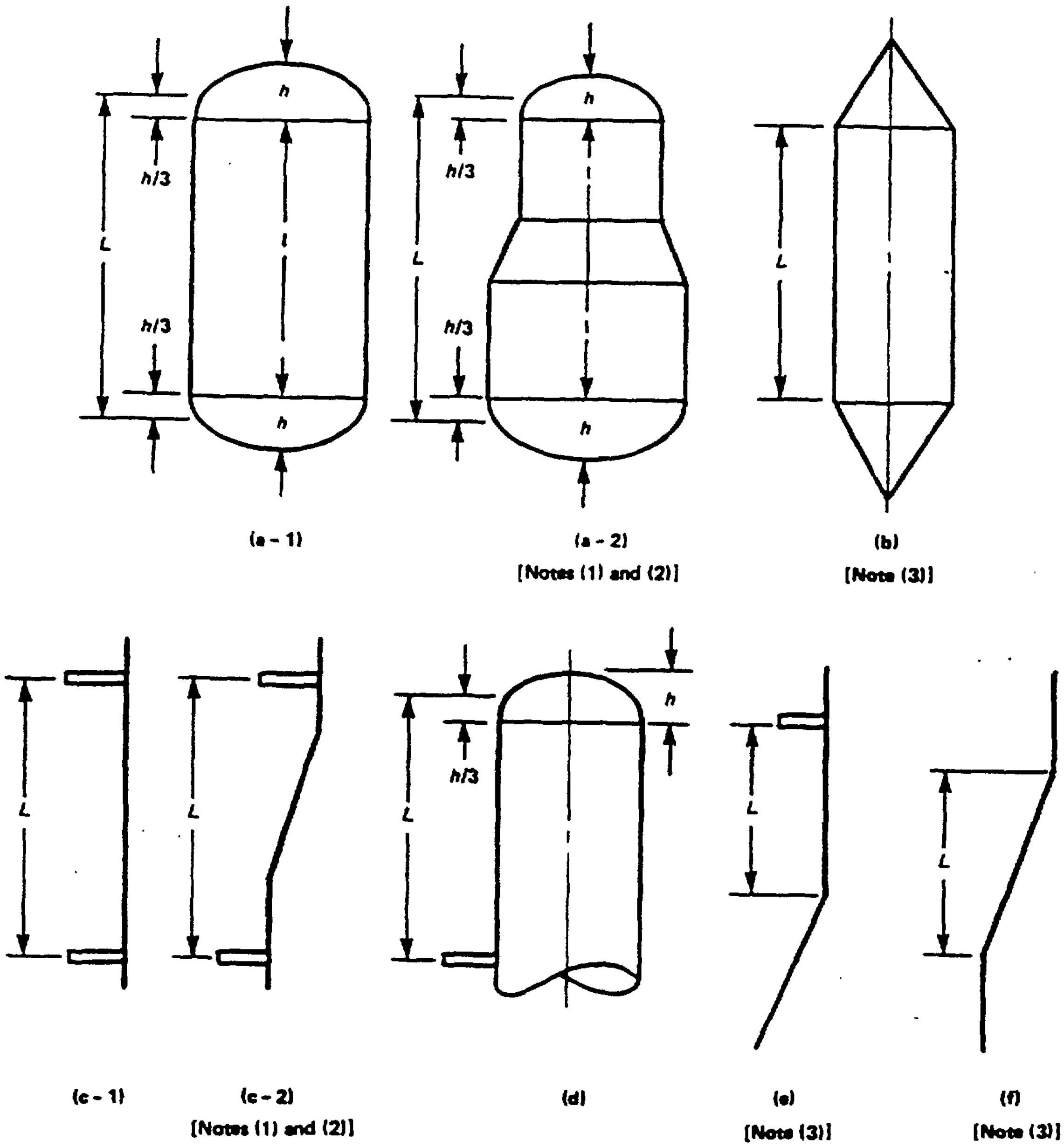


FIG. 8-1 TERMS FOR EXTERNAL PRESSURE ANALYSIS



NOTES:

(1) When the cone-to-cylinder or the knuckle-to-cylinder junction is not a line of support, the nominal thickness of the cone, knuckle, or toriconical section shall not be less than the minimum required thickness of the adjacent cylindrical shell.

(2) Calculations shall be made using the diameter and corresponding thickness of each section with dimension L as shown.

(3) When the cone-to-cylinder or the knuckle-to-cylinder junction is a line of support, the moment of inertia shall be provided in accordance with 1-8.

FIG. 9-1 LINES OF SUPPORT - CYLINDRICAL SHELLS UNDER EXTERNAL PRESSURE

Cylindrical Shells and Tubes Under External Pressure

The required minimum thickness for cylindrical shells and tubes is given in UG-28(c). First, determine D_O/T. Different rules are followed for $D_O/T \geq 10$ and $D_O/T < 10$. There is also a refinement when $D_O/T < 4$. Except when $D_O/T < 4$, Factor A is determined as follows. A value of T and L is assumed and L/D_O and D_O/T are calculated. Fig. G of Subpart 3, Section II, Part D, as shown in Fig. 10-1, is entered to determine Factor A. For L/D_O above 50 or below 0.05, use 50 or 0.05, respectively. For $D_O/T < 4$, the mode of failure is compressive stress only, so no buckling length is involved. Consequently, Factor $A = 1.1/(D_O/T)^2$. When Factor A exceeds 0.1, the value of Factor $A = 0.1$ is used.

Using Factor A, the applicable material chart listed for each specification in Subpart 3, Section II, Part D, is entered. A sample of one of these charts, Fig. CS-1, is shown in Fig. 11-1. If the value of Factor A is off of the right side of the chart, the appropriate temperature line is extended horizontally to the right, and the value of Factor B is read. If the value of Factor A appears, the value of Factor B is read directly using the appropriate temperature line. If the value of Factor A is off of the left side of the chart, the value of the maximum allowable external pressure, P_A, is determined by the equation:

$$P_A = \frac{2AE}{3(D_O/T)}$$

FOR THOSE CASES WHERE P_A HAS NOT BEEN DETERMINED BY THE FORMULA ABOVE, WHEN $D_O/T \geq 10$, THE MAXIMUM ALLOWABLE EXTERNAL PRESSURE IS:

$$P_A = \frac{4B}{3(D_O/T)}$$

WHEN $D_O/T < 10$, THE VALUES OF P_{A1} AND P_{A2} ARE BOTH DETERMINED AS FOLLOWS:

$$P_{A1} = \left[\frac{2.167}{(D_O/T)} - 0.0833\right] B$$

$$P_{A2} = \frac{2S}{D_O/T}\left[1 - \frac{1}{D_O/T}\right]$$

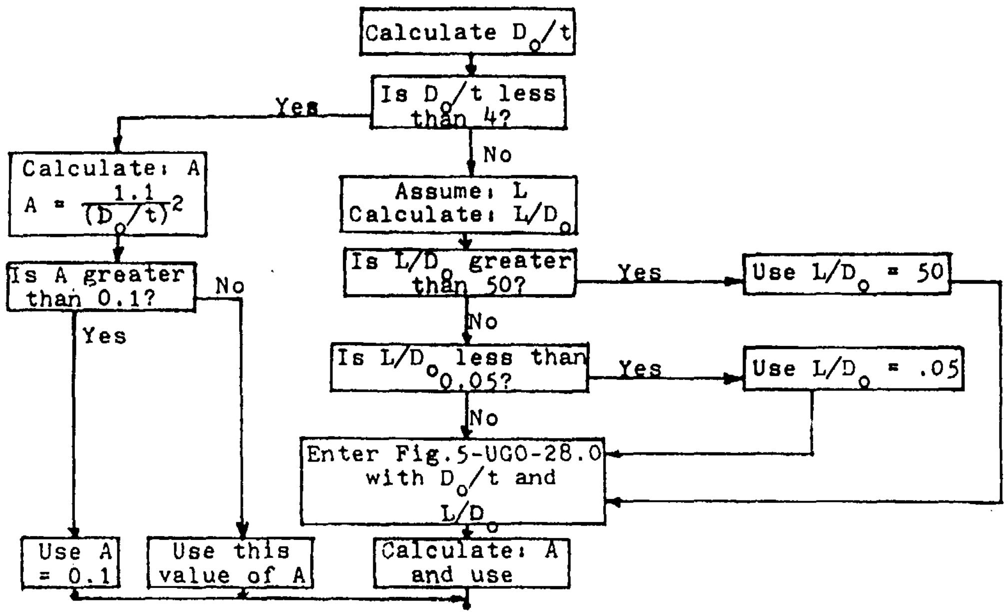
WHERE, S IS THE LESSER OF:

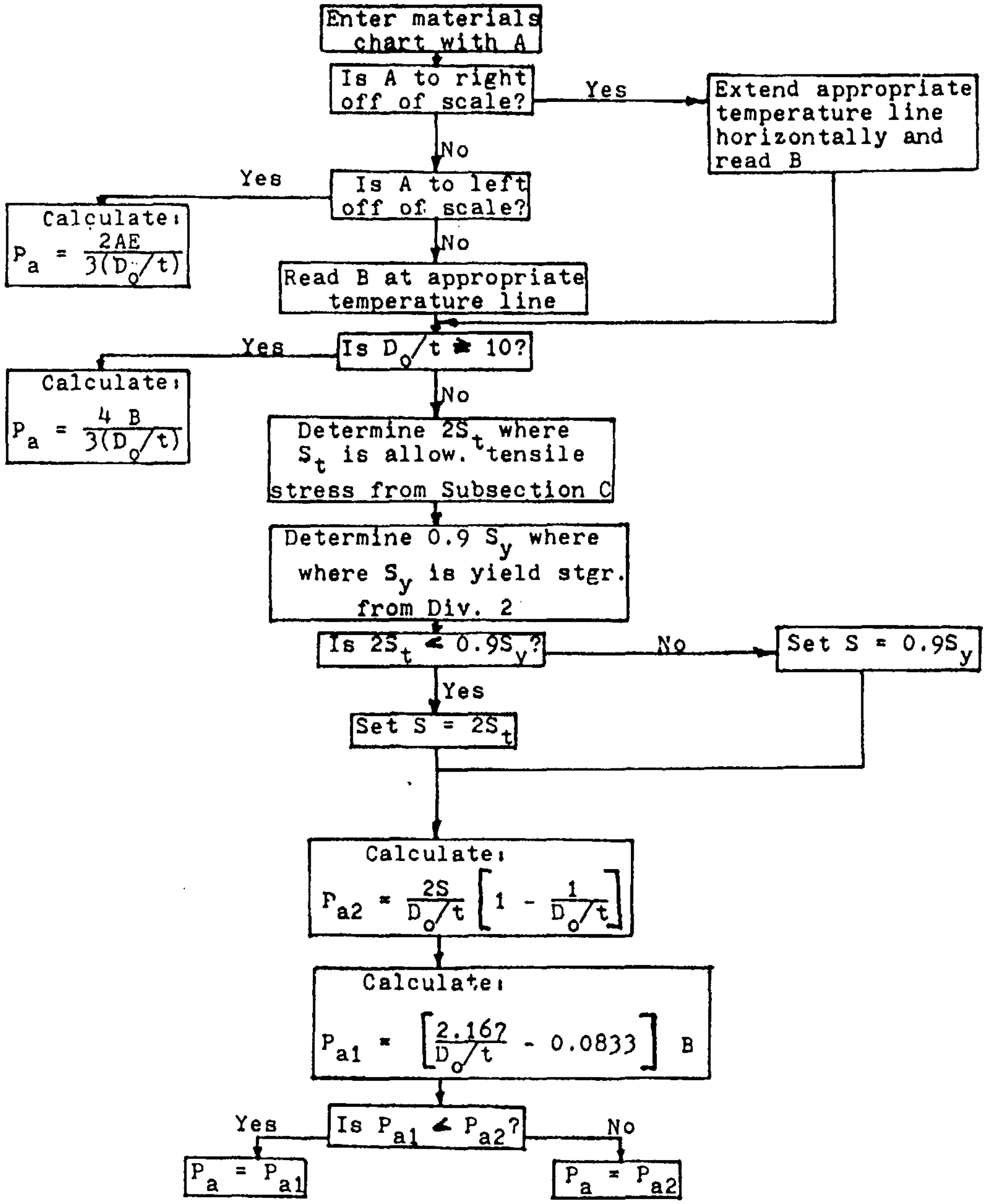
2 TIMES THE ALLOWABLE TENSILE STRESS FROM SUBSECTION C OR
0.9 TIMES THE YIELD STRENGTH, BOTH AT DESIGN TEMPERATURE

YIELD STRENGTH IS DETERMINED FROM THE APPLICABLE EXTERNAL PRESSURE CHART BY DETERMINING B FOR THE APPLICABLE TEMPERATURE CURVE AT THE MAXIMUM VALUE. THE YIELD STRENGTH = 2B.

THE MAXIMUM ALLOWABLE EXTERNAL PRESSURE, P_A, IS THE LESSER OF P_{A1} OR P_{A2}.

Maximum allowable external pressure on cylinder




Enter materials chart with A
Is A to right off of scale?
Yes
Extend appropriate temperature line horizontally and read B
No
Is A to left off of scale?
Yes
Calculate:
$P_a = \frac{2AE}{3(D_o/t)}$
No
Read B at appropriate temperature line
Is $D_o/t \geq 10$?
Yes
Calculate:
$P_a = \frac{4B}{3(D_o/t)}$
No
Determine $2S_t$ where S_t is allow. tensile stress from Subsection C
Determine $0.9 S_y$ where where S_y is yield stgr. from Div. 2
Is $2S_t < 0.9S_y$?
No
Set $S = 0.9S_y$
Yes
Set $S = 2S_t$
Calculate:
$P_{a2} = \frac{2S}{D_o/t}\left[1 - \frac{1}{D_o/t}\right]$
Calculate:
$P_{a1} = \left[\frac{2.167}{D_o/t} - 0.0833\right] B$
Is $P_{a1} < P_{a2}$?
Yes
$P_a = P_{a1}$
No
$P_a = P_{a2}$

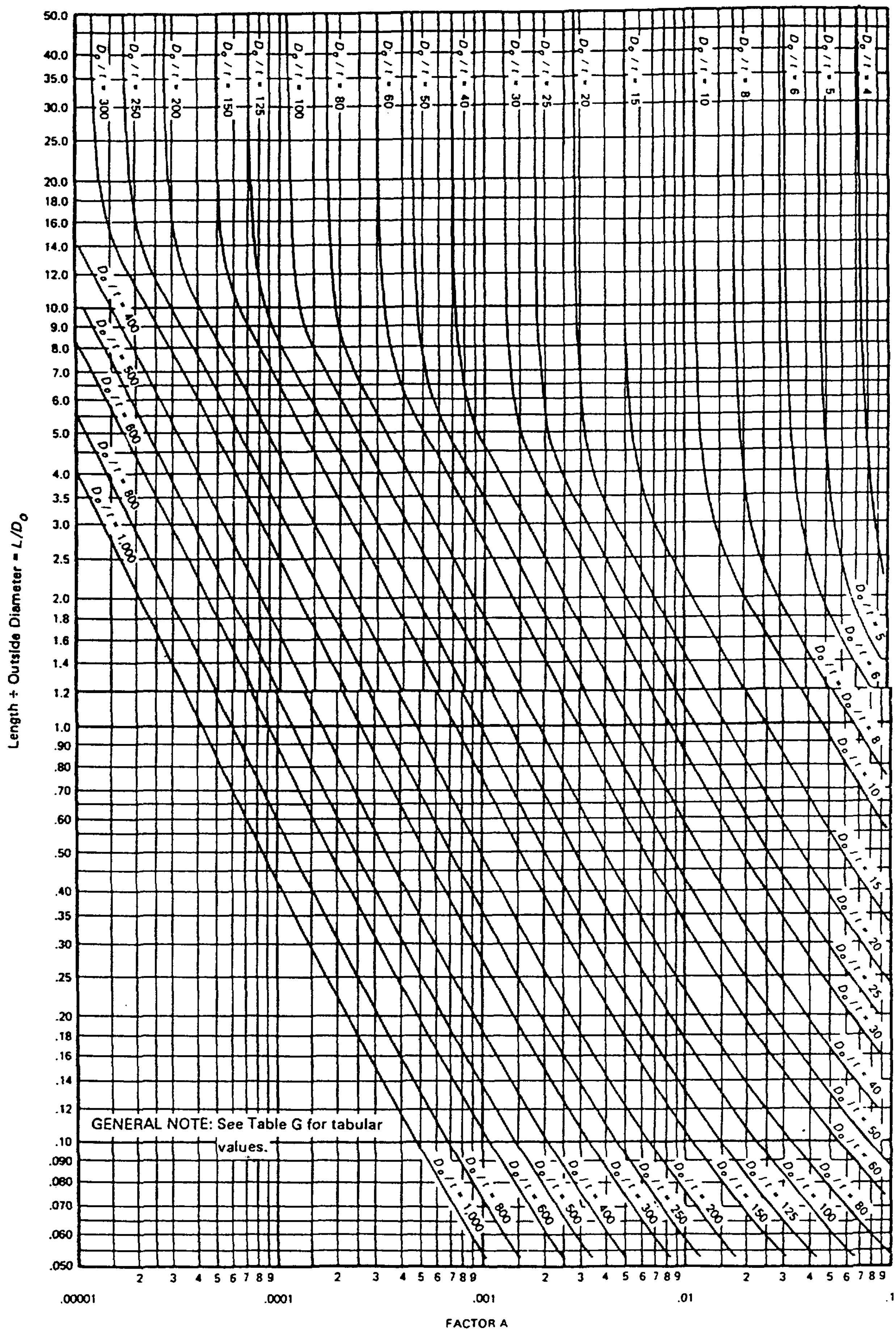


FIG. 10-1 GEOMETRIC CHART FOR ALL MATERIALS

FIG. 11-1 MATERIAL CHART FOR CARBON STEEL

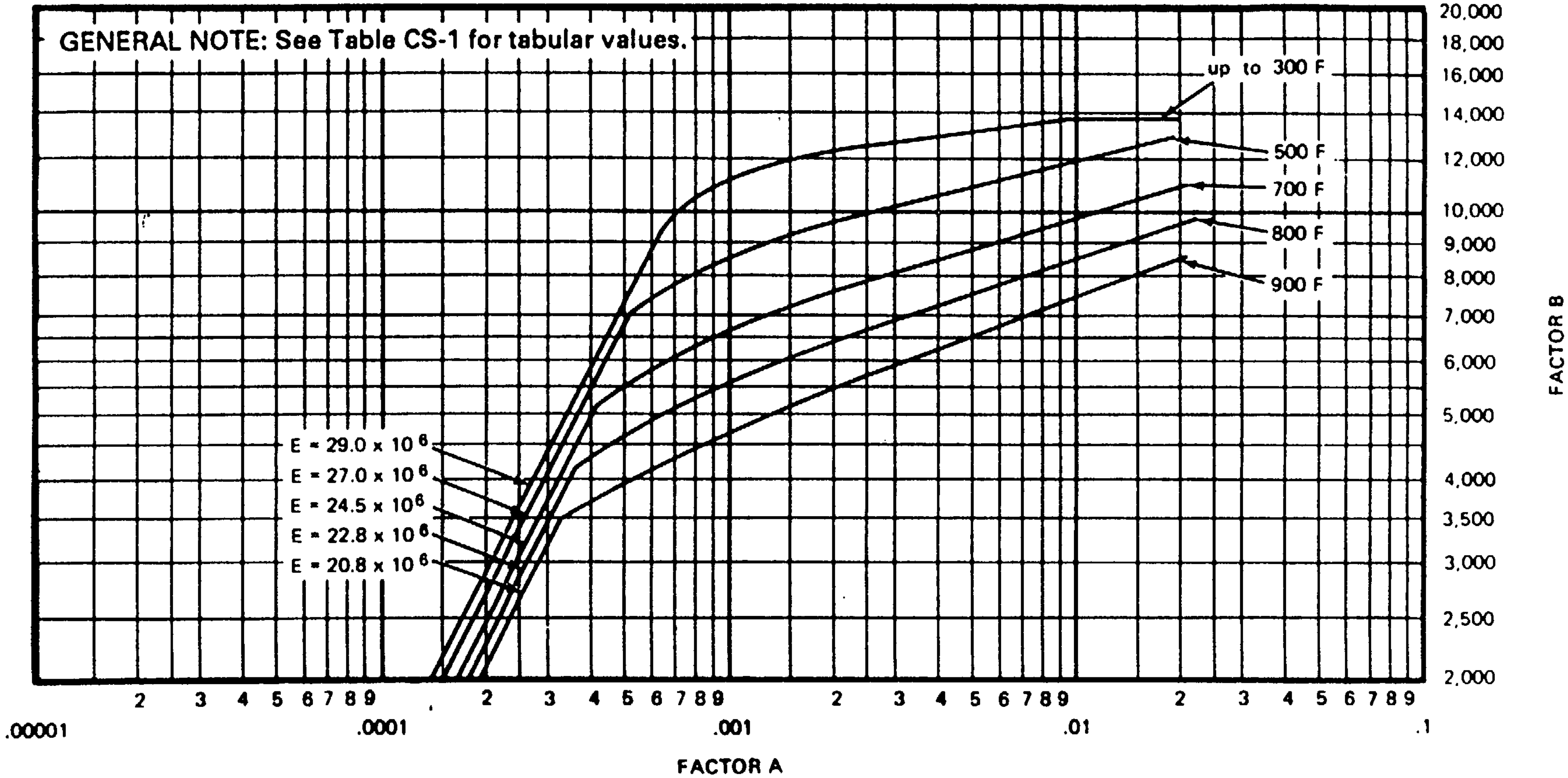


FIG. CS-1 CHART FOR DETERMINING SHELL THICKNESS OF COMPONENTS UNDER EXTERNAL PRESSURE WHEN CONSTRUCTED OF CARBON OR LOW ALLOY STEELS (SPECIFIED MINIMUM YIELD STRENGTH 24,000 psi TO, BUT NOT INCLUDING, 30,000 psi) [NOTE (1)]

Longitudinal Compressive Stress in Cylinders

For cylindrical shells and tubes, the maximum allowable longidutinal compressive stress is given in UG-23(b). A value of t is assumed and Factor A is calculated from:

$$\text{Factor } A = \frac{0.125}{(R_o/t)}$$

Using Factor A, the applicable materials chart is entered. If the value of Factor A is off the right end, the appropriate temperature line is extended horizontally and the value of Factor B is read. If the value of Factor A appears, the value of Factor B is read using the appropriate temperature line. If the value of Factor A is off the left end, the value of Factor B is:

$$\text{Factor } B = \frac{AE}{2}$$

Factor B is the maximum allowable longitudinal compressive stress.

Spherical Shells Under External Pressure

The maximum allowable external pressure on a spherical shell is also determined by a "trial and error" basis where the thickness, t, is assumed. The value of t is changed until the desired maximum allowable external pressure is obtained. This procedure is given in UG-28(d). It is similar to the procedure for maximum allowable longitudinal compressive stress. A value of t is assumed and Factor A is calculated from:

$$\text{Factor } A = \frac{0.125}{(R_o/t)}$$

Using Factor A, enter the applicable material chart of Section II, Part D. When Factor A is off the right side of the chart, the temperature lines are extended horizontally and Factor B is read directly. When Factor A can be read, Factor B is read directly using the appropriate temperature line. Using Factor B, P_A is determined by:

$$P_A = \frac{B}{(R_O/T)}$$

When Factor A is off the left side of the chart, P_A is determined by:

$$P_A = \frac{0.0625E}{(R_O/T)^2}$$

Formed Heads Under External Pressure (On Convex Side)(UG-33)

For ellipsoidal and torispherical heads under external pressure, the required thickness is determined by two methods, and the larger value of T is the minimum required thickness. The first method uses the UG-32 formulas with an adjusted value for internal pressure, and the second method uses the UG-33(c) and UG-28(d) procedures with an adjusted value of R_O.

Ellipsoidal Heads

Use the larger value for minimum required thickness.

(1) T from UG-32(d) with $P_I = 1.67\ P_E$

(2) T from UG-28(d) with $R_O = K_O\ D_O$

Torispherical Heads

Use the larger value for minimum required thickness

(1) T from UG-32(e) with $P_I = 1.67\ P_E$

(2) T from UG-28(d) with $R_O = L_1$ = outside crown radius

Hemispherical Heads

Same as for spherical shell described in UG-28(d)

Conical Heads and Sections (see Fig. 12-1)

When α, the half apex-angle, is $\leq$ 60 deg., design rules are given in UG-33(f). The rules are the same as those for external pressure on a cylindrical shell except adjustments are made for the buckling length of the cone where the equivalent length of cone, L_E, is used. The junction of the cone with the shell shall be examined to ensure adequacy. Rules for cone-to-cylinder junction are given in 1-8.

It is important to examine the buckling length when cones with a low α angle are attached to a cylindrical shell. It is possible to miss that one part is a cone and requires that stiffening rings be located at each end of the cone.

Stiffening rings which are used with cones and conical shells are calculated the same way as those for a cylinder except some of the equations and constants have to be corrected for the cone.

When $\alpha > 60$ deg., the cone thickness shall be the same as for an unstayed flat head in UG-34 assuming a diameter equal to the large diameter of the cone.

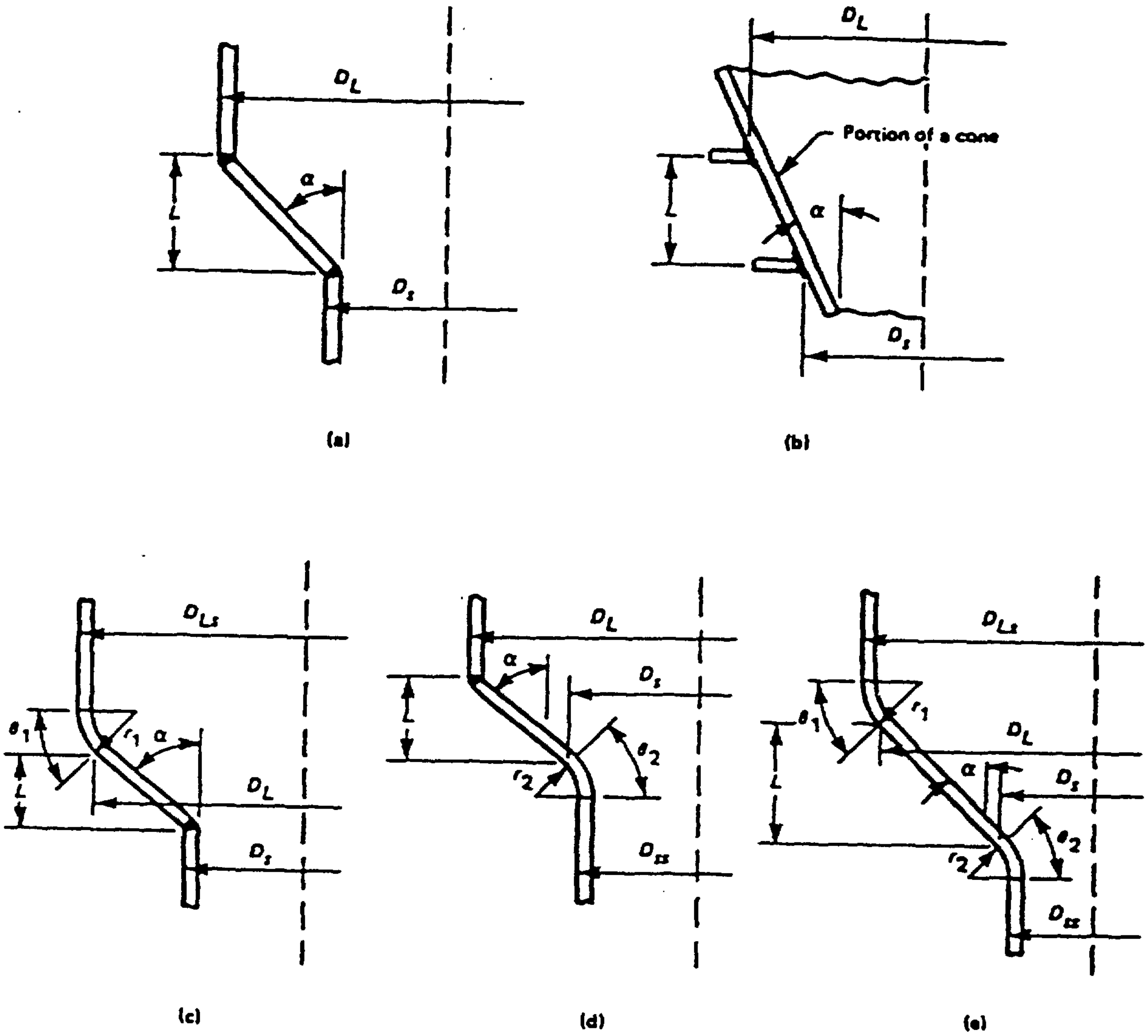


FIG. 12-1 LENGTH L OF SOME TYPICAL CONICAL SECTIONS

Stiffening Rings for External Pressure (UG-29)

In this division, the designer has the choice of considering only the stiffening ring or a combination of a stiffening ring and part of the shell wall for external pressure design. In either case, the available moment of inertia of the ring or the combined ring and shell shall be equal to or greater than the required moment of inertia of the shell or ring and shell, respectively.

(1) For the stiffening ring alone:

(a) Determine Factor B,

$$B = \frac{3}{4}\left(\frac{PD_o}{t + A_s/L_s}\right)$$

where, A_s = cross-sectional area of stiffening ring, in.^{2}

L_s = length of load to stiffening ring, in.

t and D_o are shell dimensions, in.

(b) Enter appropriate external pressure chart and determine Factor A.

(c) Determine the required moment of inertia of the stiffening ring only, I_s, in.^{4}:

$$I_s = \left[D_o^2 \, L_s \, (t + A_s/L_s)(\text{Factor A})\right] / 14.0$$

(d) Determine actual moment of inertia of ring only, I, in.^{4}

(e) I shall be equal to or greater than I_s

(2) FOR THE STIFFENING RING-SHELL COMBINATION:

(A) DETERMINE FACTOR B, AS ABOVE, WHERE A_S IS THE CROSS-SECTIONAL AREA OF THE STIFFENING RING ONLY, $IN.^2$

NOTE: WHEN THE RING-SHELL COMBINATION IS USED, THE MOMENT OF INERTIA IS DETERMINED USING A LENGTH OF SHELL OF $0.55 \sqrt{D_O T}$ ON EACH SIDE OF THE CENTERLINE. (DO NOT OVERLAP THIS SPAN WITH THE NEXT SPAN.)

(B) USING FACTOR B, ENTER THE APPROPRIATE EXTERNAL PRESSURE MATERIAL CHART FROM SECTION II-D AND DETERMINE FACTOR A.

(C) DETERMINE THE REQUIRED MOMENT OF INERTIA OF THE RING-SHELL COMBINATION, I_S', $IN.^4$:

$$I_S' = \left[D_O^2 \, L_S \, (T + A_S/L_S)(\text{FACTOR } A)\right] / \, 10.9$$

(D) DETERMINE THE MOMENT OF INERTIA OF THE COMBINED RING-SHELL SECTION ACTING TOGETHER, I', $IN.^4$

(E) I' SHALL EQUAL OR EXCEED I_S'.

ONE OF THE MAJOR DIMENSIONS WHICH CAN BE VARIED IN THE DESIGN OF A CYLINDRICAL SHELL UNDER EXTERNAL PRESSURE IS THE BUCKLING LENGTH, L. IN A SHELL WITHOUT STIFFENING RINGS, THE BUCKLING LENGTH IS EQUAL TO THE DISTANCE BETWEEN TANGENT LINES PLUS ONE-THIRD OF THE DEPTH OF EACH HEAD. IF STIFFENING RINGS ARE USED, THE BUCKLING LENGTH IS EQUAL TO THE LARGEST DISTANCE BETWEEN STIFFENING RINGS OR THE DISTANCE FROM THE END RING TO THE TANGENT LINE PLUS ONE-THIRD OF THE HEAD DEPTH.

Attachment of Stiffening Rings to Shell (UG-30)

Stiffening rings may be welded to either the inside or outside of a shell. Attachments between the inner shell and the outer jacket of a jacketed-vessel may also serve as a stiffening ring. Various arrangements of stiffening rings and attachment welds are shown in Fig. 13-1.

(1) Strength of Attachment Welds - Attachment welds are sized to resist the radial pressure load, PL_S, and to carry the radial shear load which is equal to 2% of the ring's compressive load.

Radial pressure load = $P_E L_S$, lb./in.

Radial shear load = $0.01 P_E L_S D_O$, in.

(2) Minimum Size of Attachment Welds - The minimum fillet weld size (leg dimension), when used, shall be the least of:

(a) 1/4-in.;

(b) shell thickness at weld;

(c) stiffening ring thickness at weld.

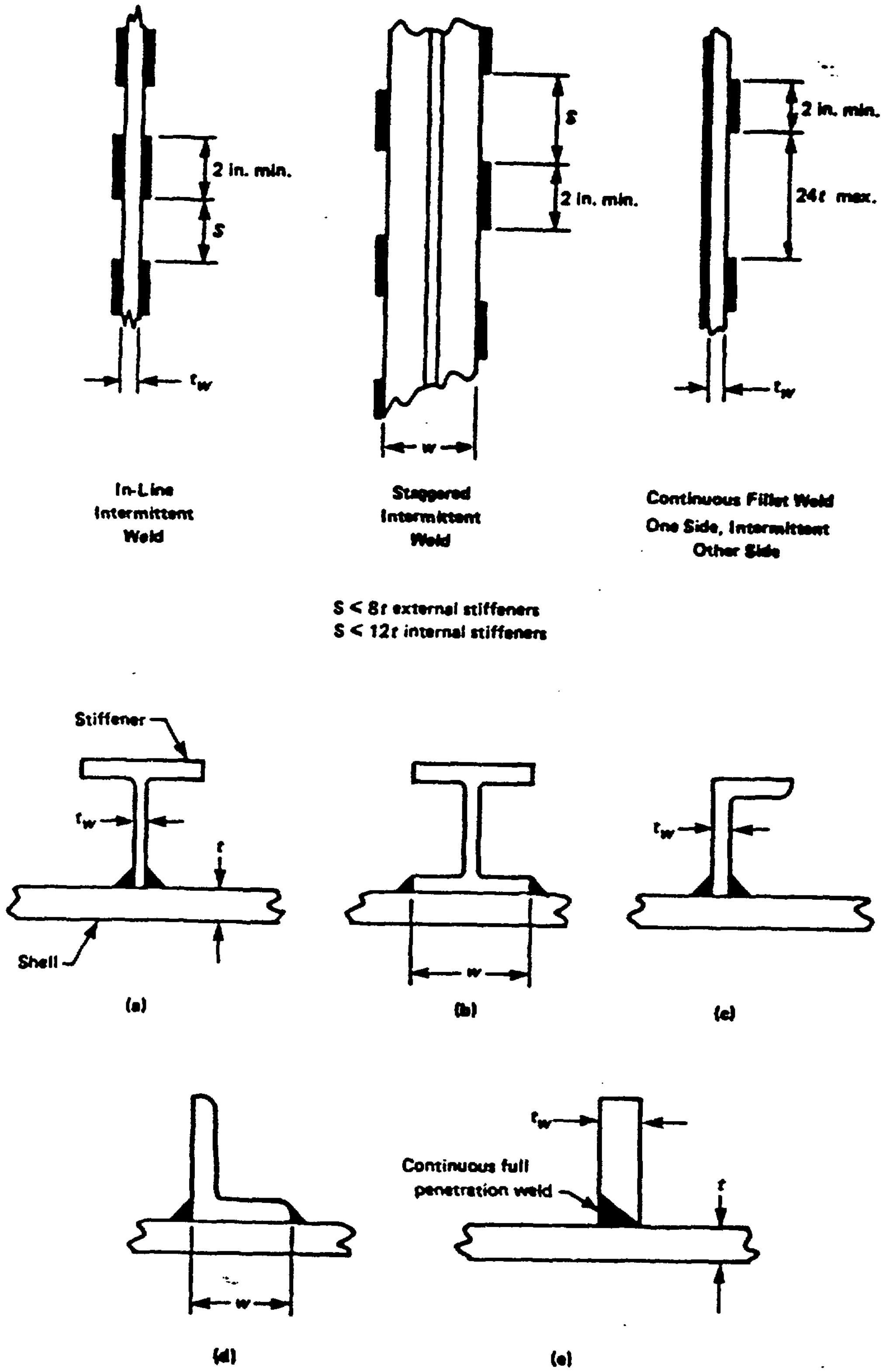


FIG. 13-1 STIFFENING RINGS AND ATTACHMENT WELDS

OPENINGS AND REINFORCEMENT

GENERAL

OPENINGS IN PRESSURE VESSELS SHALL BE EXAMINED TO DETERMINE THEIR ABILITY TO TRANSFER LOADINGS AND STRESSES ACROSS AN OPENING IN AN OTHERWISE SOLID SECTION OF A VESSEL UNDER EITHER INTERNAL PRESSURE OR EXTERNAL PRESSURE. ALL OPENINGS, REGARDLESS OF THEIR SHAPE, SHALL BE EXAMINED ACCORDING TO THE REQUIREMENTS IN UG-36 THROUGH UG-42 AND IN 1-7 IF REQUIRED BY SIZE LIMITS. IN ADDITION, MULTIPLE OPENINGS MAY BE EXAMINED ACCORDING TO THE LIGAMENT RULES IN UG-53 IN PLACE OF THE RULES OF UG-36 THROUGH UG-42 UNLESS EXEMPTED BY SIZE, TYPE, OR SPECIAL APPLICATION. FOR SPECIAL APPLICATIONS, THE RULES IN CODE CASE 2168 MAY BE USED AN ALTERNATIVE.

SIZE LIMITS FOR CYLINDRICAL SHELLS

WHEN THE SIZE LIMITS LISTED BELOW ARE MET, THE REQUIREMENTS GIVEN IN UG-36 THROUGH UG-42 APPLY. THESE LIMITS ARE:

(1) FOR VESSELS 60-IN. DIAMETER AND LESS, OPENING DIAMETER SHALL NOT EXCEED ONE-HALF OF VESSEL DIAMETER AND NOT MORE THAN 20-IN.;

(2) FOR VESSELS OVER 60-IN. DIAMETER, OPENING DIAMETER SHALL NOT EXCEED ONE-THIRD OF VESSEL DIAMETER AND NOT MORE THAN 40-IN.

WHEN THE OPENING EXCEEDS THESE LIMITS, THE REQUIREMENTS IN 1-7 SHALL APPLY IN ADDITION TO THE REQUIREMENTS IN UG-36 THROUGH UG-42. IN ORDER TO COMPLY WITH THESE REQUIREMENTS, IT MAY BE NECESSARY TO MAKE

TWO SETS OF CALCULATIONS FOR MOST OPENINGS TO DECIDE WHICH METHOD IS DESIRED. THE ADDITIONAL REQUIREMENTS OF 1-7 REQUIRE THAT AREA OF REINFORCEMENT AVAILABLE FROM EXCESS SHELL THICKNESS BE PLACED CLOSE TO THE EDGE OF THE OPENING. IN ADDITION, EFFECTS OF LOCAL BENDING STRESSES ARE ALSO EVALUATED. WHEN THE REQUIREMENTS OF 1-7 ARE NECESSARY, THE DESIGN MAY BE SET BY EITHER SET OF REQUIREMENTS; AND, CONSEQUENTLY, IT IS NECESSARY TO EVALUATE THE OPENINGS BY BOTH REQUIREMENTS OF UG-37 THROUGH UG-42 AND BY 1-7.

SIZE LIMITS FOR SPHERICAL SHELLS AND FORMED HEADS

FOR OPENINGS IN SPHERICAL SHELLS AND FORMED HEADS, THERE ARE NO SPECIFIC SIZE LIMITATIONS GIVEN; AND, CONSEQUENTLY, FOLLOWING THE LIMITS FOR OPENINGS IN CYLINDRICAL SHELLS IS RECOMMENDED. FOR LARGE OPENINGS IN SPHERICAL SHELLS AND FORMED HEADS, IT IS SUGGESTED THAT REVERSE CURVES AND CONICAL TRANSITIONS MAY BE HELPFUL IN KEEPING STRESS CONCENTRATIONS LOW.

SPECIAL REQUIREMENTS FOR SMALL OPENINGS (UG-36(c)(3))

WHEN APPLICABLE, SPECIAL DESIGN REQUIREMENTS FOR SMALL OPENINGS IN SHELLS AND HEADS MAY APPLY. WHEN THESE RULES ARE MET, THE VESSEL IS ASSUMED TO HAVE INHERENT REINFORCEMENT AND NO CALCULATIONS ARE REQUIRED. THE OPENING SHALL BE IN A VESSEL NOT SUBJECTED TO RAPID FLUCTUATION IN PRESSURE NOR TO OTHER LOADINGS GIVEN IN UG-22 AND WHICH MEET THE ADDITIONAL REQUIREMENTS WHICH FOLLOW:

(a) For welded or brazed connections:

(1) In plate of 3/8-in. or less in thickness, maximum opening is 3.5-in. in diameter;

(2) In plate greater then 3/8-in. in thickness, maximum opening is 2.375-in. in diameter.

(b) For threaded, studded, or expanded connections:

(1) In all plate thicknesses, maximum opening is 2.375-in. in diameter.

(c) No openings described in (a) or (b), above, shall have their centers closer to each other than the sum of their diameters.

(d) No two openings in a cluster of three or more as described in (a) and (b) shall have their centers closer than:

* $(1 + 1.5\cos\Theta)(d_1 + d_2)$ for cylinders and cones

** $2.5(d_1 + d_2)$ for double-curvature surfaces

<u>Openings In Or Adjacent To Welded Joints (UW-14)</u>

Openings located in or adjacent to welded joints, as determined when the opening or its reinforcement is closer to the weld centerline than *T*, plate thickness at the weld, shall have the weld joint efficiency used in calculating the available reinforcement area in the vessel wall, A_1, and shall meet the requirements of UW-14.

REINFORCEMENT REQUIREMENTS

REINFORCEMENT REQUIREMENTS ARE GIVEN FOR OPENINGS IN SHELLS AND HEADS FOR BOTH INTERNAL PRESSURE AND EXTERNAL PRESSURE. THE RULES ARE THE SAME EXCEPT THAT THE MINIMUM REQUIRED THICKNESS IS DETERMINED BY THE APPLICABLE FORMULA IN UG-27 FOR INTERNAL PRESSURE AND BY THE RULES OF UG-28 FOR EXTERNAL PRESSURE, AND THE AREA REQUIRED FOR EXTERNAL PRESSURE IS ONLY 50% OF THE INSIDE DIAMETER TIMES THE MINIMUM REQUIRED THICKNESS.

NOMENCLATURE

A_R = TOTAL AREA OF REINFORCEMENT REQUIRED IN THE PLANE BEING EXAMINED, $IN.^2$

A_1 = EXCESS AREA AVAILABLE IN VESSEL WALL, $IN.^2$

A_2 = EXCESS AREA AVAILABLE IN NOZZLE WALL OUTSIDE OF SHELL, $IN.^2$

A_3 = EXCESS AREA AVAILABLE IN NOZZLE WALL INSIDE OF SHELL, $IN.^2$

A_4 = AREA AVAILABLE IN FILLET WELDS, $IN.^2$

A_5 = AREA AVAILABLE IN REINFORCING PAD, $IN.^2$

D = DIAMETER OF OPENING ALONG LONGITUDINAL PLANE OR CHORD OF OPENING IN NON-RADIAL CIRCUMFERENTIAL PLANE, IN.

T_R = MINIMUM REQUIRED THICKNESS OF A SEAMLESS SHELL BASED ON CIRCUMFERENTIAL STRESS OR A FORMED HEAD, IN., EXCEPT:

(1) WHEN THE OPENING AND ITS REINFORCEMENT ARE ENTIRELY WITHIN THE SPHERICAL PORTION OF A TORISPHERICAL HEAD;

(2) WHEN THE OPENING IS IN A CONICAL HEAD OR SECTION;

(3) WHEN THE OPENING AND ITS REINFORCEMENT ARE IN AN ELLIPSOIDAL HEAD AND ARE LOCATED ENTIRELY WITHIN A CENTRALLY-LOCATED CIRCLE OF 80% OF THE SHELL DIAMETER;

(4) WHEN THE OPENING IS LOCATED IN A VESSEL BUILT TO THE RULES OF UW-12(c)

T_{RE} = MINIMUM REQUIRED THICKNESS OF SHELL OR HEAD FOR EXTERNAL PRESSURE, IN.

T = NOMINAL THICKNESS OF SHELL OR HEAD, IN.

T_{RN} = MINIMUM THICKNESS OF NOZZLE, IN.

T_N = NOMINAL THICKNESS OF NOZZLE, IN.

F = CORRECTION FACTOR FOR CALCULATION OF STRESSES ON DIFFERENT PLANES WITH RESPECT TO THE LONGITUDINAL PLANE (AS SHOWN IN FIG. UG-37). USE OF THE F FACTOR IS LIMITED TO INTEGRAL REINFORCEMENT LISTED UNDER THE TITLE OF FIG. UW-16.1.

F_R = MATERIAL STRENGTH REDUCTION FACTOR = $S_N/S_V \leq 1.0$

S_N = ALLOWABLE TENSILE STRESS OF NOZZLE OR OTHER PARTS SUPPLYING REINFORCEMENT AREA AT DESIGN TEMPERATURE, PSI

S_V = ALLOWABLE TENSILE STRESS OF VESSEL MATERIAL AT DESIGN TEMPERATURE, PSI

Openings in Heads and Shells

Required area of reinforcement

Internal pressure, nozzle inserted thru shell wall:

$$A_R = DT_R F + 2T_N T_R F(1 - F_{R1})$$

Internal pressure, nozzle abutting shell wall:

$$A_R = DT_R F$$

External pressure, nozzle inserted thru shell wall:

$$A_R = 0.5 \left\{ DT_{RE} + 2T_N T_{RE} (1 - F_{R1}) \right\}$$

External pressure, nozzle abutting shell wall:

$$A_R = 0.5 \quad DT_{RE}$$

Limits of reinforcement

Metal area available for reinforcement shall be located within the limits of reinforcement which are established as follows:

(1) Limits parallel to the shell surface on each side of the centerline of the opening are the larger of:

(A) D

(B) $0.5D + T + T_N$

(2) Limits perpendicular to the shell surface measured from the shell surface (adjusted for T_E) either inward or outward are the smaller of the following:

(a) 2.5 t

(b) $2.5\ t_N + t_E$

When the plane being examined is the longitudinal plane and the nozzle is perpendicular to the shell surface with the nozzle and its reinforcement not extending into areas of different minimum required thicknesses, the area available for reinforcement may be determined from the following formulas:

(1) Area available in vessel wall is the larger of:

$$A_1 = d(E_1t - Ft_R) - 2t_N(E_1t - Ft_R)(1 - f_{R1}) \text{ or}$$

$$A_1 = 2(t + t_N)(E_1t - Ft_R) - 2t_N(E_1t - Ft_R)(1 - f_{R1})$$

(2) Area available in nozzle wall is the smaller of:

$$A_2 = 2\left\{2.5t(t_N - t_{RN})f_{R1}\right\} = 5(t_N - t_{RN})f_{R1}t$$

$$A_2 = 2\left\{2.5t_N(t_N - t_{RN})f_{R1}\right\} = 5(t_N - t_{RN})f_{R1}t_N$$

When the plane being examined is a plane other than the longitudinal plane and when the opening and its reinforcement extend from an area of one minimum required thickness into an area of a different minimum required thickness (such as would happen at an opening placed thru the weld joint at a head-to-shell joint), the equations given above are not applicable. In most cases, it is best to construct the rectangle of reinforcement area to determine what area is available for reinforcement.

For openings which are of a size that require examination by the rules of 1-7 in addition to the rules of UG-36 thru UG-42, the following additional requirements shall be met:

Required Area of Reinforcement

For all conditions listed above,

$$A_{R'} = (2/3)\ A_R \quad \text{where } A_{R'} = \text{required area by 1-7 rules}$$

Limits of Reinforcement

For limit (1)(a) listed above, d becomes 0.75d and all other limits remain as given above.

Strength of Reinforcement at Openings

Not all material used for reinforcement is integral with the shell wall. Therefore, if the material is considered part of the reinforcment, the strength of the welds attaching the reinforcement must be examined. New rules, including formulas for examining these weld loads and weld strength paths have been added. As noted in Fig. UW-15 and beneath the title of Fig. UW-16.1, certain nozzle and reinforcement combinations do not require the load paths to be examined because they are considered as integral reinforcement. For those which must be examined, the various allowable stress values for the weld material are also given in UW-15.

Openings in Flat Heads and Covers (UG-39)

When single and multiple openings are located in a flat head or cover and when the opening diameter is equal to or less than one-half of the head diameter, the following rules apply:

(1) Single openings, the required reinforcement area is

$$A_R = 0.5\ dt_R$$

(2) Multiple openings, when no pair has an average diameter greater than one-quarter of the head diameter and the spacing between any pair of openings is equal to or greater than twice their average diameter, the formula given in (1), above, may be used.

When the spacing between any pairs of openings is less than twice their average diameter, but equal to or greater than 1.25 times the average diameter of the pair, the required reinforcement for each opening in the pair, using the formula in (1), above, shall be added together and distributed such that 50% of the sum is located between the two openings.

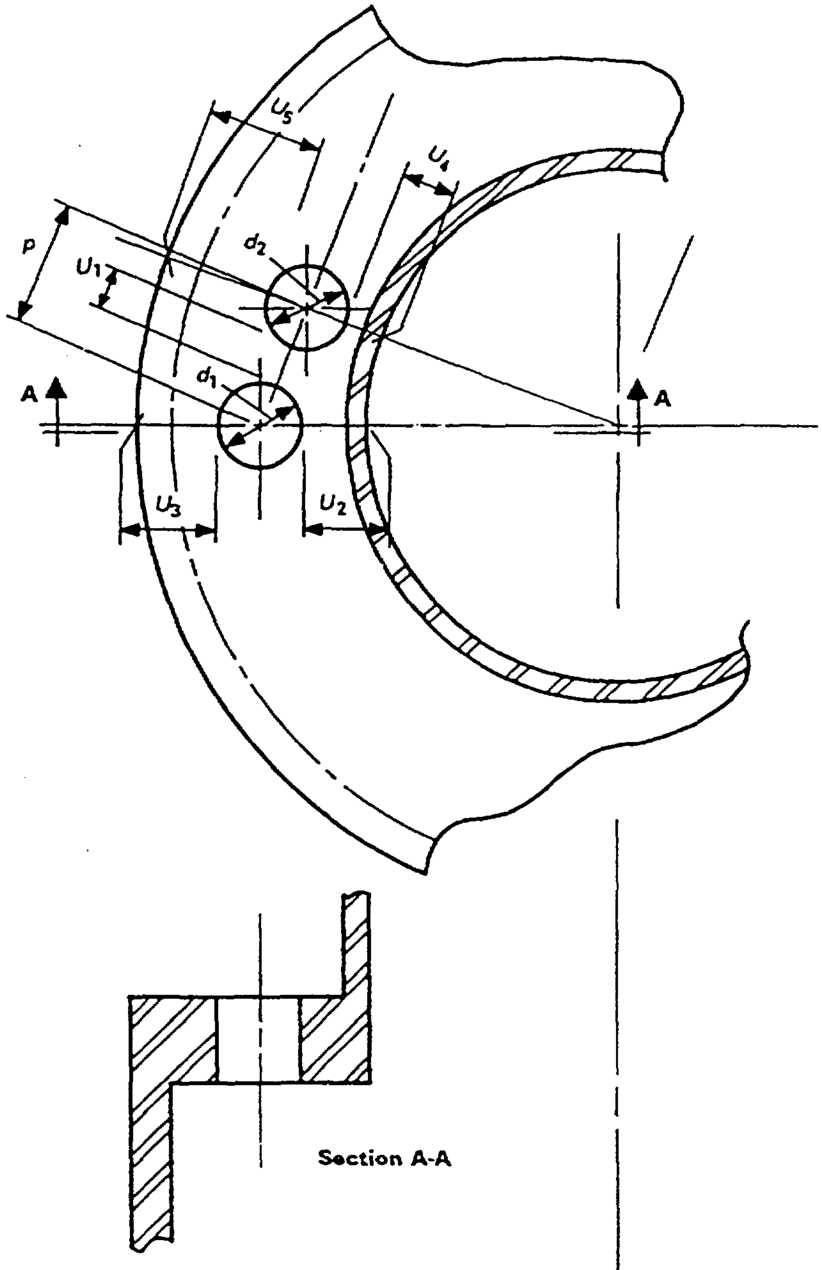
When the spacing is less than 1.25 times the average diameter, the rules of U-2(g) apply. In no case shall the ligament between pairs of openings be less than one-quarter the diameter of the smaller opening. In addition, the ligament between the edge of an opening and the edge of the flat head shall be not less than one-quarter of the diameter of the opening.

WHEN SINGLE AND MULTIPLE OPENINGS ARE LOCATED IN A FLAT HEAD OR COVER AND WHEN THE OPENING DIAMETER IS GREATER THAN ONE-HALF OF THE HEAD DIAMETER, THE FOLLOWING RULES APPLY:

(1) FOR A SINGLE, CIRCULAR, CENTRALLY-LOCATED OPENING, THE RULES OF APPENDIX 14 SHALL BE USED. THE HEAD DOES NOT HAVE TO SATISFY THE RULES OF UG-34. ANY OPENINGS IN THE RIM MAY BE CALCULATED USING THE RULES GIVEN ON THE PREVIOUS PAGE FOR EITHER SINGLE OR MULTIPLE OPENINGS USING THE REQUIRED THICKNESS DETERMINED BY APPENDIX 14. ALTERNATIVELY, THE HEAD THICKNESS DETERMINED BY APPENDIX 14 MAY BE MULTIPLIED BY 1.414 FOR A SINGLE OPENING IN THE RIM OR IF THE SPACING BETWEEN TWO OPENINGS IN THE RIM IS EQUAL TO OR GREATER THAN TWICE THE AVERAGE DIAMETER. IF THE SPACE BETWEEN TWO OPENINGS IS LESS THAN TWICE THE AVERAGE DIAMETER, THE THICKNESS FROM APPENDIX 14 SHALL BE DIVIDED BY THE SQUARE ROOT OF E, WHERE $E = (P - D_{AVE})/P_{SMALLEST}$. THE OPENINGS IN THE RIM SHALL BE NO LARGER IN DIAMETER THAN ONE-QUARTER OF THE DIFFERENCE IN HEAD DIAMETER LESS CENTRAL DIAMETER. OTHER EDGE LIGAMENTS SHOWN IN FIG. 14-1 SHALL BE MET. IF THE DETAILS ARE NOT GIVEN, U-2(G) SHALL BE USED.

(2) AS AN ALTERNATIVE, FOR SINGLE OPENINGS WITH A DIAMETER EQUAL TO OR LESS THAN ONE-HALF THE HEAD DIAMETER, THE THICKNESSES OBTAINED IN UG-34 MAY BE INCREASED BY ADJUSTING THE VALUE OF C AS FOLLOWS:

(A) IN FORMULAS (1) OR (3) OF UG-34(C), USE THE LESSER OF



p = spacing, center-to-center, between openings
$U_1, U_2, \ldots$ = ligament width
$(d_1 + d_2)/2$ = average diameter of pair of openings

MULTIPLE OPENINGS IN RIM OF HEADS WITH A LARGE CENTRAL OPENING

FIG. 14-1 OPENINGS IN FLAT HEADS

2C OR 0.75 SUBSTITUTED FOR C FOR ALL SKETCHES WHICH ARE APPLICABLE EXCEPT THOSE GIVEN BELOW;

(B) IN FORMULAS (1) AND (3) OF UG-34(C), USE THE LESSER OF 2C OR 0.50 SUBSTITUTED FOR C FOR SKETCHES (B-1), (B-2), (E), (F), (G), AND (I);

(C) IN FORMULAS (2) AND (5) OF UG-34(C), MULTIPLY BY 1.414.

OVERLAPPING REINFORCEMENT EXCEPT IN FLAT HEADS (UG-42)

WHEN ANY TWO OPENINGS (EXCEPT FLAT HEADS) ARE SPACED SO THAT THEIR LIMITS OF REINFORCEMENT OVERLAP, THE RULES OF UG-42 SHALL BE FOLLOWED. THESE RULES ESSENTIALLY STATE THAT NO PORTION OF ANY REINFORCEMENT AREA SHALL BE CONSIDERED MORE THAN ONCE AND SHALL BE PROPORTIONED ACCORDING TO THE DIAMETERS OF THE TWO OPENINGS. IF THE OPENINGS HAVE LESS THAN 50% OF THE TOTAL REQUIRED AREA BETWEEN THEM, THE RULES OF 1-7 ALSO APPLY. WHEN THE OPENINGS ARE IN A RANDOM PATTERN AND HAVE OVERLAPPING REINFORCEMENT, IT MAY BE REQUIRED TO CONSIDER THEM AS ONE LARGE NOZZLE WITH A DIAMETER OF A CIRCLE ENCOMPASSING THE ENTIRE PATTERN.

Braced and Stayed Surfaces (UG-47 thru UG-50)

THE MINIMUM REQUIRED THICKNESS AND MAXIMUM ALLOWABLE WORKING PRESSURE FOR BRACED AND STAYED FLAT PLATES IS GIVEN IN UG-47. THE RULES ARE SIMILAR TO THOSE FOR FLAT PLATES AND COVERS AND INVOLVE A C-FACTOR BASED ON THE METHOD OF ATTACHING THE BRACE OR STAYBOLT TO THE FLAT PLATE. STAYBOLT RULES ARE ALSO GIVEN.

Ligaments (UG-53)

WHEN TWO OR MORE OPENINGS IN A CYLINDRICAL SHELL ARE ARRANGED IN A PATTERN (EVEN TWO HOLES IS A PATTERN), THE RULES FOR LIGAMENTS MAY BE USED. IN CALCULATING MINIMUM REQUIRED THICKNESS BY LIGAMENT EFFICIENCY METHODS, ALL METAL WHICH IS BEING CONSIDERED TO AID THE OPENING COMES FROM EXCESS THICKNESS IN THE SHELL WALL. AT THE PRESENT TIME, THERE IS NO CONSIDERATION OF EXCESS THICKNESS IN THE NOZZLE WALL.

WHEN LIGAMENT EFFICIENCY IS CONSIDERED IN A CYLINDRICAL SHELL WHICH MAY ALSO HAVE A BUTT-WELD JOINT EFFICIENCY OR A QUALITY FACTOR TO BE CONSIDERED, THE EFFICIENCIES ARE NOT COMBINED. THE SMALLER EFFICIENCY FACTOR IS USED TO SET THE MINIMUM REQUIRED THICKNESS.

THE CODE HAS NO SPECIFIC INFORMATION TO INFORM THE DESIGNER WHEN TO IT IS BEST TO USE THE LIGAMENT EFFICIENCY METHOD AND WHEN IT IS BEST TO USE THE REINFORCEMENT RULES. THE DESIGNER MUST MAKE THAT DETERMINATION!

Pressure Testing of Vessels and Components

Introduction

All pressure vessels and components which are designed and constructed to VIII-1 are required to pass either a hydrostatic test (UG-99), a pneumatic test (UG-100), or a proof test (UG-101) of the completed vessel or vessel part before they are U-stamped. Each vessel section of the Code has a pressure test rule which requires pressure testing at or above the maximum allowable working pressure (MAWP) indicated on the nameplate or stamping before the appropriate Code-stamp mark can be applied.

Under certain conditions, a pneumatic test may be combined with or substituted for a hydrostatic test. When the testing conditions require a combination of a pneumatic test with a hydrostatic test, the testing requirements for the pneumatic test shall be followed. In all cases, hydrostatic infers not only water being the test medium but also oil and other fluids which are not dangerous or flammable as being acceptable; and pneumatic infers not only air being used but also other non-dangerous gases which may be desirable for "sniffer" detection also as being acceptable.

There is always a difference of opinion as to what is desired and accomplished with a pressure test. Some believe the pressure test is to detect major leaks while others feel there should be no leaks - large or small. Some feel that the test is necessary to invoke loadings and stresses which are equivalent to or exceed those loadings and stresses at operating conditions. Others feel that a pressure test is needed to indicate whether or not a gross error has been made in calculations or fabrication. In some cases, it appears that the pressure testing may help to round-out corners or other undesirable wrinkles or may offer some sort of a stress relief to some components.

Hydrostatic Testing Requirements (UG-99)

A hydrostatic pressure test is the preferred test method. The pneumatic test or the combination of pneumatic/hydrostatic test is conducted only when the hydrostatic test cannot be done. Except for certain types of vessels discussed later, the hydrostatic test pressure at every point in the vessel shall be at least 1.3 times the MAWP marked on the vessel multiplied by the lowest ratio of the stress value at test temperature divided by the stress value at design temperature for the material used. As an alternative, a hydrostatic test pressure may be determined by calculations by agreement between the User and the Manufacturer. In this case, the MAWP of each element is determined and multiplied by 1.3 and then adjusted by the stress ratio given above. Test pressure is the minimum of these. Test pressure is limited to that which will not cause any visible permanent distortion (yielding) of any element. For hydrostatic testing, the metal temperature is recommended to be at least 30°F above the MDMT marked on the vessel but need not exceed 120°F. Also, a liquid relief valve set at 1-1/3 times the test pressure is recommended in the test system.

After the hydrostatic test pressure is reached, the pressure shall be reduced to the test pressure divided by 1.3, and all welded joints, connections, and other areas visually examined for leaks and cracks. The visual examination may be waived if a gas leak test is to be applied, if hidden welds have been examined ahead of time, and if the vessel will not contain anything lethal. Venting shall be provided at all high locations where there is a possibility of air pockets forming during the filling of the vessel for testing. The general rules for hydrostatic testing do not call for a specific time for holding the vessel at test pressure. This time may be set by the Authorized Inspector or by a contract specification.

Pneumatic Testing Requirements (UG-100)

For some vessels, it is necessary to apply a pneumatic test in lieu of a hydrostatic test. This may be due to any number of reasons including vessels designed and supported in a manner so that they cannot be safely filled with liquid and vessels which cannot tolerate any trace of water or other liquids. If a vessel is to be pneumatically tested, it shall first be examined according to UW-50. This paragraph requires that welds around openings and attachments be examined by MT or PT before testing. Except for enameled vessels, the pneumatic test pressure at every point in the vessel shall be 1.1 times the MAWP marked on the vessel multiplied by the lowest ratio of the stress value at test temperature divided by the stress value at design temperature for the material used. For pneumatic testing, the metal temperature shall be at least 30°F above the MDMT marked on the vessel but need not exceed 120°F.

The test pressure shall be gradually increased to no more than one-half of the full test pressure and then increased in steps of one-tenth of the test pressure until the full test pressure is reached. After that, the pressure shall be reduced to the test pressure divided by 1.1; and all areas examined. All other requirements for hydrostatic testing shall be observed including the waiving of the visual examination provided the same limits are met.

Testing Requirements for Enameled or Glass-Lined Vessels

The maximum test pressure for enameled and glass-lined vessels does not have to be any greater than 1.0 MAWP unless required by the Authorized Inspector or by a contract specification. Higher test pressure may damage the enameled or glass coating. All other rules for hydrostatic testing apply.

Testing Requirements for Vessels Built to Part UCI and Part UCD

For vessels designed and constructed to the rules of Part UCI for Cast Iron and Part UCD for Cast Ductile Iron, where the factor of safety on tensile strength for setting allowable stress values is 10 and 5, respectively, the hydrostatic test pressure is set differently. For Part UCI, the test pressure shall be 2.5 MAWP but not to exceed 60 psi for a design pressure less than 30 psi and 2.0 MAWP for a design pressure equal to or greater than 30 psi.

For Part UCD, the test pressure shall be 2.0 MAWP. With these changes, the remaining rules of UG-99 are followed.

Test Requirements for Vessels Built to Part ULT

Alternative rules for the design and construction of vessels to operate at low temperatures are given in Part ULT. These rules permit the use of increased allowable tensile stress values at low temperature for 5%, 8%, and 9% nickel steels, 5083 aluminum alloy, and Type 304 stainless steels. Other material listed in Section II and Subsection C may be used for vessel parts with the allowable tensile stress values set by the value at 100°F. When the vessel is designed and constructed to Part ULT rules, special hydrostatic testing requirements are necessary due to the fact that the material is stronger at design temperature than at ambient temperature with the test pressure held for 15 minutes using either:

(a) a standard hydrostatic test as described in UG-99 but with the ratio of allowable stresses not applied and the test pressure shall be 1.6 MAWP instead of 1.3 MAWP;

(b) in applying method (a), above, the membrane stress in the vessel shall not exceed 0.95 of the specified minimum yield strength nor 0.5 of the specified minimum tensile strength. In complying with these stress limits, the ratio of hydrostatic test pressure divided by the MAWP may be reduced below 1.6, but it shall be not less than 1.25 MAWP. If the value comes out less than 1.25 MAWP, a pneumatic test shall be conducted using the rules of UG-100 but omitting the adjustment for the allowable tensile stress ratio.

A vessel to be installed vertically may be tested in the horizontal position provided the test pressure is applied for 15 minutes at not less than 1.6 MAWP including a pressure equivalent to the liquid head in operating position.

Proof Testing to Establish MAWP (UG-101)

For vessels and vessel parts for which the strength cannot be calculated, rules are given for a pressure proof test to establish the MAWP. In UG-101, provisions are made for two types of tests to determine the MAWP for internal pressure:

(1) one based on yielding of the part (limited to materials with a yield/tensile strength ratio $\leq$ 0.625),

(2) one based on bursting of the part.

Proof test methods listed below establish a pressure at which the test is terminated:

(1) brittle-coating test procedure,

(2) bursting test procedure (this test may be stopped before rupture if the desired MAWP is reached),

(3) strain measurement test procedure,

(4) displacement measurement test procedure.

For setting MAWP, the basic factor applied to internal pressure burst test results is 5 and for yield test results is variable. For vessels subject to collapse, the hydro. test pressure is 3 MAWP.

APPENDIX 1 - SUPPLEMENTARY DESIGN FORMULAS

FORMULAS ARE GIVEN FOR DETERMINING THE MINIMUM REQUIRED THICKNESS OR MAXIMUM ALLOWABLE WORKING PRESSURE OF SHELLS UNDER INTERNAL PRESSURE. FOR CYLINDRICAL SHELLS AND SPHERICAL SHELLS, EQUATIONS ARE GIVEN IN TERMS OF THE OUTSIDE RADIUS AND FOR THICK SHELLS USING LAME' EQUATIONS. FOR FORMED HEADS UNDER INTERNAL PRESSURE, GENERAL EQUATIONS ARE GIVEN IN TERMS OF INSIDE AND OUTSIDE DIAMETER. THESE INCLUDE EQUATIONS FOR ELLIPSOIDAL HEADS, TORISPHERICAL HEADS, AND CONICAL HEADS. RULES ARE GIVEN FOR CONICAL REDUCER SECTIONS AND CONICAL HEADS UNDER INTERNAL PRESSURE WHEN THE HALF-APEX ANGLE, α, IS $\leq 30^\circ$. WHEN THE HALF-APEX ANGLE IS $> 30^\circ$, NO DESIGN RULES ARE GIVEN BUT LIMITS ON VARIOUS TYPES OF STRESSES ARE GIVEN IN 1-5. ALSO, IN THIS APPENDIX ARE RULES FOR CIRCULAR SPHERICALLY-DISHED HEADS WITH BOLTING FLANGE-RINGS FOR BOTH INTERNAL AND EXTERNAL PRESSURE. RULES ARE ALSO GIVEN FOR DESIGN OF LARGE OPENINGS IN CYLINDRICAL SHELLS. FINALLY, RULES ARE GIVEN FOR THE DESIGN OF CONE-TO-CYLINDER JUNCTIONS UNDER EXTERNAL PRESSURE WHEN THE HALF-APEX ANGLE IS $\leq 60^\circ$.

APPENDIX 2 - BOLTED FLANGE CONNECTIONS WITH RING-TYPE GASKETS

RULES ARE GIVEN FOR THE DESIGN OF BOLTED FLANGED CONNECTIONS UTILIZING GASKETS INSIDE OF THE BOLT CIRCLE WITH NO METAL-TO-METAL CONTACT OF THE ADJACENT SURFACES. RULES ARE GIVEN FOR HYDROSTATIC END LOADS AND FOR GASKET SEATING LOADS. INCLUDED IN THE DESIGN OF THE FLANGE ARE SELECTION OF THE GASKET AND FLANGE FACING, BOLTING,

AND FLANGE DIMENSIONS. SPECIAL RESTRICTIONS ARE GIVEN FOR FABRICATED, HUBBED FLANGES. IT IS SUGGESTED THAT BOLTING BE KEPT TO NOT LESS THAN ½-IN. DIAMETER UNLESS ALLOY STEEL IS USED. CIRCULAR FLANGES ARE CLASSIFIED INTO THREE TYPES FOR CALCULATING STRESSES - LOOSE TYPE, INTEGRAL TYPE, AND OPTIONAL TYPE. GASKET DESIGN FACTORS OF M AND Y ARE GIVEN FOR VARIOUS TYPES OF GASKETS AND MATERIALS. ALTHOUGH THE M AND Y FACTORS ARE NOT MANDATORY, THEY GENERALLY HAVE PROVEN SATISFACTORY IN ACTUAL SERVICE. FORMULAS ARE GIVEN FOR CALCULATING FLANGE MOMENTS AND FLANGE STRESSES AND COMPARING THEM TO ALLOWABLE FLANGE STRESSES. SPECIAL RULES ARE GIVEN FOR SPLIT LOOSE FLANGES, FLANGES UNDER EXTERNAL PRESSURE, AND REVERSE FLANGES. OTHER TYPES ARE PERMITTED, BUT NO DESIGN PROCEDURES ARE GIVEN AND U-2(G) SHALL BE FOLLOWED.

APPENDIX 3 - DEFINITIONS

THIS APPENDIX CONTAINS MANY DEFINITIONS WHICH ARE USED THROUGHOUT DIVISION 1 AND IN OTHER SECTIONS OF THE CODE. DEFINITIONS FOR SPECIFIC APPLICATIONS ARE GIVEN IN THE RELATED PART OF THIS SECTION. INCLUDED ARE SOME OF THE MORE COMMON DEFINITIONS SUCH AS:

(1) MAXIMUM ALLOWABLE WORKING PRESSURE (MAWP) - THE MAXIMUM GAGE PRESSURE PERMISSIBLE AT THE TOP OF A COMPLETED VESSEL IN ITS OPERATING POSITION FOR THE DESIGN TEMPERATURE.

(2) VESSEL WALL THICKNESS - INCLUDING DIFFERENCES BETWEEN "REQUIRED THICKNESS," "DESIGN THICKNESS," AND "NORMAL THICKNESS."

(3) BUTT-WELD JOINT EFFICIENCY AND QUALITY FACTOR - RELATED TO TYPE OF NDE INSPECTION GIVEN TO THE BUTT WELDED JOINTS.

Appendix 9 - Jacketed Vessels

This appendix contains design rules for jacketed vessels beyond those given in the General Requirements of Subsection A. The parts of a jacketed vessel consist of the inner shell, the jacket closure, the jacket, and nozzle penetrations through the jacket from the inner shell. Special attention is called to the fact that the inner shell may be subjected to both internal pressure and to external pressure. Unless specific information is supplied and noted on the Manufacturer's Data Report Form that one side cannot be pressurized without the other side being pressurized, the inner shell shall be designed for each pressure condition independently from the other. Jacketed vessels are divided into five types depending upon the part covered by the jacket. One of the critical design areas is the closure member of the jacket to the shell. Each type of closure is limited to use with certain types of jackets which have various configurations and details as shown in Fig. 9-5 of the Code. The other critical design consideration is the design of nozzles and openings through the jacket space from the inner shell to the outside. Provisions for expansion and pressure motion differences are required for all nozzles from the inner shell as shown in Fig. 9-6 of the Code.

Appendix 13 - Vessels of Noncircular Cross Section

THIS APPENDIX CONTAINS REQUIREMENTS, FORMULAS, AND EXAMPLES FOR THE DESIGN AND CONSTRUCTION OF SOME SINGLE WALL VESSELS WHICH HAVE A RECTANGULAR OR OBROUND CROSS SECTION WITH OR WITHOUT INTERNAL STAYING MEMBERS OR WHICH HAVE A CIRCULAR CROSS SECTION WITH STAYING MEMBERS. RULES ARE GIVEN FOR BOTH INTERNAL AND EXTERNAL PRESSURE. THESE RULES DO NOT PROHIBIT THE DESIGN AND CONSTRUCTION OF VESSELS WITH NONCIRCULAR CROSS SECTIONS OTHER THAN THOSE DESCRIBED; HOWEVER, THOSE VESSELS SHALL BE DESIGNED ACCORDING TO U-2(G).

DESIGN IS BASED ON DETERMINING BOTH MEMBRANE AND BENDING STRESSES AND COMPARING THEM WITH THE ALLOWABLE STRESSES WHICH INCLUDE A BUTT-WELD JOINT OR A LIGAMENT EFFICIENCY, SE. ALLOWABLE STRESSES FOR COMBINED MEMBRANE AND BENDING STRESS ARE PERMITTED TO GO TO THE LIMITS GIVEN IN UG-23(C) AND 13-4(B)&(C). CONSIDERATION IS GIVEN TO BOTH MEMBRANE LIGAMENT EFFICIENCY AND BENDING LIGAMENT EFFICIENCY. WITH THE VARIOUS COMBINATIONS OF EFFICIENCIES AND STRESSES, IT MAY BE NECESSARY TO EXAMINE SEVERAL LOCATIONS TO DETERMINE THE CRITICAL STRESS LOCATION.

Appendix 14 - Integral Flat Heads With A Large, Single, Circular, Centrally-Located Opening

DESIGN RULES ARE GIVEN FOR AN INTEGRAL, CIRCULAR FLAT HEAD WITH A SINGLE, CIRCULAR, CENTRALLY-LOCATED OPENING WHERE THE OPENING IS LARGER THAN ONE-HALF OF THE FLAT HEAD DIAMETER. THIS APPENDIX IS

TO BE USED AS DESCRIBED IN UG-39(c)(1). THE METHOD IS SIMILAR TO THAT WHICH IS USED TO CALCULATE FLANGE DESIGNS AND FLANGE STRESSES. WHEN THIS PROCEDURE IS USED, IT IS NOT NECESSARY TO MEET THE RULES OF UG-34. THIS METHOD DETERMINES STRESSES AT THE HEAD-SHELL JUNCTION AS WELL AS AT THE NOZZLE-HEAD JUNCTION WITH OR WITHOUT A NOZZLE BEING ATTACHED.

APPENDIX 16 - SUBMITTAL OF TECHNICAL INQUIRIES TO THE BOILER AND PRESSURE VESSEL COMMITTEE

GUIDANCE IS GIVEN TO THE CODE USER FOR SUBMITTING A TECHNICAL INQUIRY TO THE ASME BOILER AND PRESSURE VESSEL COMMITTEE. INQUIRIES SHALL BE FOR A SPECIFIC PURPOSE AS FOLLOWS:

(1) REVISION OF PRESENT CODE RULE(S),

(2) NEW OR ADDITIONAL CODE RULE(S),

(3) CODE CASE,

(4) CODE INTERPRETATION.

DEPENDING UPON THE TYPE OF INQUIRY, BACKGROUND INFORMATION AND NEED SHALL BE STATED IN THE SUBMITTAL. FOR CODE INTERPRETATIONS, THE DESIRABLE FORMAT IS TO SUBMIT A PROPOSED INQUIRY AND REPLY WHERE THE PREFERRED REPLY CAN BE "YES" OR "NO." INQUIRES NOT MEETING THE PROCEDURES DESCRIBED IN THIS APPENDIX WILL BE RETURNED TO THE INQUIRER FOR CLARIFICATION AND RESUBMITTAL.

Appendix 17 - Dimpled or Embossed Assemblies

These assemblies are made by welding plates together which have been previously formed or are formed by using hydraulic or pneumatic pressure. The MAWP is the lowest pressure established by proof test or by calculations. Any variations in geometry requires a new proof test or calculations to set a new MAWP. Several different welding processes are permitted; however, the joints are considered to be Category C joints and service restrictions such as no lethal service nor use as unfired steam boilers nor use at low temperature operation are envoked. Special welding rules and testing procedures shall be described in the Quality Control program.

Appendix 19 - Electrically Heated or Gas Fired Jacketed Steam Kettles

Electrically heated or gas fired steam kettles are limited to a maximum design pressure of 50 psi in the jacket with no steam nor water to be taken from the jacket for external use. No structural grades of material are permitted for pressure parts. Austenitic stainless steel materials exposed to combustion gas shall be low carbon or stabilized grades. Butt weld joints exposed to combustion gas shall be of Type 1 only.

Appendix 20 - Hubs of Tubesheets and Flat Heads Machined From Plate

Plate to be used for tubesheets and flat heads with hubs shall have through thickness properties which are equal or better than those specified in the material specification. All requirements of the material specification shall be met. In addition, test specimens shall be taken in a direction parallel to the axis of the hub as described in this Appendix. Each test specimen shall be examined as described in 20-3. If no hub exists, this Appendix does not apply.

Appendix 21 - Jacketed Vessels Constructed of Work-Hardened Nickel

These jacketed vessels shall have the inner shell made from work-hardened SB-162 nickel material to increase the strength for resistance to structural collapse. The maximum inside diameter is 8-feet and the maximum design temperature is $400^{\circ}F$. The vessel shall have a hydrostatic test pressure in the jacket of three times the MAWP.

Appendix 22 - Integrally Forged Vessels

These integrally forged vessels are limited to SA-372, Types I, II, III, IV, and V; Grades 1, 2, 3, 4, and 5; and Classes A and B, materials. The maximum allowable stress value is based on 1/3 of the specified minimum tensile strength of the material.

Appendix 23 - External Pressure Design of Copper and Copper-Alloy Seamless Condenser and Heat-Exchanger Tubes With Integral Fins

RULES ARE GIVEN TO COVER PROOF TESTING PROCEDURES AND CRITERIA FOR DETERMINING THE MAXIMUM ALLOWABLE EXTERNAL WORKING PRESSURE OF TUBES WITH EXTERNAL, HELICAL FINS. THIS METHOD REQUIRES A TEST TO VISIBLE COLLAPSE FAILURE FROM APPLICATION OF EXTERNAL PRESSURE TO THREE FULL SIZE SPECIMENS. THE MAWP IS THEN DETERMINED BY:

$$P = \frac{B}{3} \times \frac{Y_S}{Y_A}$$

WHERE, B = COLLAPSE PRESSURE, MINIMUM, PSI

Y_S = SPECIFIED YIELD STRENGTH AT DESIGN TEMPERATURE, PSI

Y_A = AVERAGE ACTUAL YIELD STRENGTH OF THREE SPECIMENS AT DESIGN TEMPERATURE, PSI

THE MAWP IS LIMITED TO 450 PSI AND DESIGN TEMPERATURE IS LIMITED TO 150°F. THERE ARE OTHER ADDITIONAL RESTRICTIONS.

Appendix 24 - Design Rules for Clamp Connections

RULES ARE GIVEN FOR THE DESIGN OF CLAMP CONNECTIONS FOR BOTH VESSELS AND PIPING WITHIN THE SCOPE. THESE RULES ARE SIMILAR TO THE FLANGE RULES IN APPENDIX 2. THE GASKET DESIGN, BOLTING, AND HUB & CLAMP GEOMETRY ARE DETERMINED. GASKETS ARE EITHER LOW-LOAD OR SELF-ENERGIZING. THE STRUCTURAL ADEQUACY OF ALL COMPONENTS IS FOUND.

Appendix 26 - Bellows-Type Expansion Joints

This appendix contains rules for the design and construction of expansion joints which are an integral part of heat exchangers and other pressure vessels. The expansion joints are thin-wall, single layered, and bellows-type geometry. The present rules apply only to axial deflection and hydrostatic end force due to internal pressure. The minimum thickness limitations are waived. Design rules are given for both the unreinforced bellows and the reinforced bellows. Additional requirements are given for fabrication, examination, testing, and marking or stamping of these expansion joints.

Appendix 27 - Alternative Requirements for Glass-Lined Vessels

Rules of this Appendix give alternative requirements which are applicable to glass-lined (enamel-lined) vessels. All other requirements of this Division are applicable. Due to the inability of the glass-lining to deform during hydrostatic testing or to accommodate adjustment for out-of-roundness and to prevent problems with the lining during heat treatments, special alternative requirements are permitted. If the out-of-roundness exceeds the limitations set in UG-80, provisions are made to determine a reduced MAWP. Hydrostatic test pressure does not have to exceed 1.0 MAWP. Other special limits apply when heat treatment of test specimens and PWHT are required. These rules are limited to SA-285, SA-516, and SA-836 materials.

Appendix 28 - Alternative Corner Joint Weld Details for Box-Headers for Air-Cooled Heat Exchangers When Only One Member Is Beveled

This Appendix provides alternative weld details for box-headers for air-cooled heat exchangers using multi-pass corner weld joint construction as shown in Figure 28-1. The requirements of "A + B not less than $2t_s$" in UG-13(e)(4) and the weld joint details of Fig. UW-13.2 are voided and replaced by additional requirements of this Appendix. All other requirements of this Division pertaining to weld joints apply.

Appendix 29 - Requirements for Steel Bars of Special Section for Helically Wound Interlocking Strip Layered Pressure Vessels

This Appendix provides requirements for steel bars of special section for helically-wound interlocking strip layered pressure vessels. The material is essentially SA-29/SA-29M with modification in the chemical composition, mechanical properties, heat treating temperatures, testing, and maximum thickness.

Supplemental Materials and Design Rules in Nonmandatory Appendices

Appendix A - Basis for Establishing Allowable Loads for Tube-to-Tubesheet Joints

Rules are given for determining the ability of various types of tube-to-tubesheet joints to transfer axial loads and to "stay" the tubesheet. These rules do no apply to U-tube construction.

Appendix B - Approval of New Material for ASME Code Construction

Instructions are given for adoption of new materials for ASME Code construction. An inquirer shall furnish mechanical properties, weldability, and structural stability characteristics of the new material. At the present time, it is noted that the ASME will adopt only ASTM specifications for materials in Section II, Parts A and B, and AWS specifications for weld materials in Section II, Part C. Adoption of other materials is being reviewed.

Appendix D - Suggested Good Practice for Internal Structures

Suggestions are given to help prevent local failures and distortions where loads are transferred into the vessel wall and to make provisions for differences in thermal expansion.

Appendix G - Suggested Good Practice Regarding Piping Reactions and Design of Supports and Attachments

Warnings are given regarding primary and secondary-type stresses generated by local loadings from support attachments. In general,

NO DESIGN RULES ARE GIVEN AND NO METHOD FOR DETERMINING ALLOWABLE STRESSES IS LISTED. SUGGESTIONS FOR THE NUMBER OF SUPPORTS, THE SUPPORT LOCATION, AND OTHER WARNINGS AND SUGGESTIONS TO IMPROVE THE VESSEL DESIGN ARE GIVEN.

APPENDIX L - EXAMPLES SHOWING APPLICATION OF FORMULAS AND RULES

THIS APPENDIX CONTAINS EXAMPLE PROBLEMS WHICH ILLUSTRATE THE APPLICATION OF FORMULAS AND RULES FOR SOME GEOMETRIES GIVEN IN THE CODE. THERE ARE MANY EXAMPLE PROBLEMS SHOWING HOW TO DETERMINE SHELL AND HEAD THICKNESS UNDER INTERNAL PRESSURE FOR VARIOUS AMOUNTS OF INSPECTION OF THE BUTT-WELD JOINTS ACCORDING TO THE REQUIREMENTS OF UW-12 AND FIG. UW-12 AND LIGAMENT EFFICIENCIES. EXAMPLES ARE GIVEN FOR EXTERNAL PRESSURE INCLUDING THICKNESS, OUT-OF-ROUNDNESS, AND STIFFENING RING DESIGN AND ATTACHMENT DETAILS. THERE ARE ALSO EXAMPLES OF CALCULATING NOZZLE REINFORCEMENT AND LIGAMENT EFFICIENCY.

APPENDIX S - DESIGN CONSIDERATIONS FOR BOLTED FLANGE CONNECTIONS

WHEN SOME UNUSUAL FEATURE EXISTS IN A BOLTED FLANG SUCH AS A LARGE DIAMETER, HIGH PRESSURE, HIGH TEMPERATURE, SPECIAL DESIGN RULES MAY BE REQUIRED. AN IMPORTANT POINT IS THAT THE DESIGN RULES OF APPENDICES 2 AND Y GIVE SAFE DESIGNS FOR THE BOLTS AND FLANGES. HOWEVER, SPECIAL CONSIDERATION MAY BE REQUIRED TO PROVIDE A SATISFACTORY JOINT DURING HYDROSTATIC TESTING. CARE MUST BE TAKEN SO AS TO NOT YIELD THE JOINT BUT TO HAVE NO LEAKAGE.

Appendix Y - Flat Face Flanges with Metal-to-Metal Contact Outside of the Bolt Circle

THIS APPENDIX CONTAINS RULES FOR THE DESIGN OF IDENTICAL AND NON-IDENTICAL FLANGE PAIRS WHERE THE FLANGES ARE FLAT FACED AND ARE IN METAL-TO-METAL CONTACT ACROSS THE FACE. THE RULES ALSO APPLY WHEN A METAL SPACER IS USED. THIS CONSTRUCTION GENERALLY USES A SELF-SEALING GASKET WHICH LOWERS THE GASKET SEATING LOAD. IN GENERAL, LOADS ARE LOW FOR GASKET SEATING AND THE DESIGN CONDITION CONTROLS. THIS PROCEDURE IS A "TRIAL-AND-ERROR" ANALYSIS BECAUSE ASSUMPTIONS ARE MADE TO DETERMINE LOADINGS WHICH MUST BE CONFIRMED LATER.

Appendix AA - Rules for Design of Tubesheets

THIS APPENDIX CONTAINS DESIGN FORMULAS AND PROCEDURES FOR EITHER A SIMPLE-SUPPORTED OR INTEGRAL TUBESHEET WHICH IS CIRCULAR WITH AN UNPERFORATED RIM AND FULLY TUBED. RULES ARE GIVEN FOR A U-TUBE DESIGN AND FOR A FIXED TUBESHEET DESIGN. THE TUBE HOLES ARE ON EITHER A TRIANGULAR OR A SQUARE PATTERN. EXAMPLES ARE GIVEN FOR DETERMINING THE MINIMUM TUBESHEET THICKNESS DUE TO PRESSURE.

Appendix CC - Flanged and Flanged-and-Flued Expansion Joints

THIS APPENDIX CONTAINS RULES FOR THE DESIGN AND CONSTRUCTION OF FLANGED AND FLANGED-AND-FLUED EXPANSION JOINTS FOR HEAT EXCHANGERS AND PRESSURE VESSELS. THE SAME LIMITATIONS WHICH APPLY TO THE BELLOWS-TYPE EXPANSION JOINTS ALSO APPLY TO THE FLANGED AND FLANGED-AND-FLUED EXPANSION JOINTS. ONLY AXIAL DEFLECTION IS CONSIDERED.

Appendix EE - Half-Pipe Jackets

THIS APPENDIX CONTAINS DESIGN FORMULAS, CURVES, AND PROCEDURES FOR DETERMINING THE MAXIMUM PERMISSIBLE PRESSURE, P', IN HALF-PIPE JACKETS. AN EXAMPLE IS GIVEN FOR USING THE RULES.

Appendix 1

Background and Organization

of the ASME Code and Code Committees

Background

In 1911, the Council of the American Society of Mechanical Engineers appointed a committee of seven members called: "A Committee to Formulate Standard Specifications for the Construction of Steam Boilers and Other Pressure Vessels, and for Their Care in Service." Later this was shortened to the ASME Boiler and Pressure Vessel Committee.

On February 13, 1915, the first ASME Boiler Code was issued. The title was: "Rules for the Construction of Stationary Boilers and for Allowable Working Pressures, 1914 Edition." The original Code was issued as one volume and was later divided into many sections including Section I, Power Boilers. Other sections were added or deleted later with Section VIII being added in 1925 and being divided into two Divisions in 1968.

What Code Sections Were Added/Deleted and When?

Following the issuance of the 1914 Edition, what Code sections were added and deleted and what is their status?

1921 Section III - Boilers for Locomotives (with the 1962 Edition, this section was integrated into Section I and the Section III designation was later reassigned.)

1922 Section V - Miniature Boilers (with the 1962 Edition, this section was integrated into Section I and the Section V designation was later reassigned.)

1923 Section IV - (Low Pressure) Heating Boilers*

1924 Section II - Materials* (until 1924, this section was part of Section I with various titles.)

1925 Section VIII - (Unfired) Pressure Vessels* (with the 1968 Edition, this section was divided into two divisions and the word: *Unfired* was deleted.)

1926 Section VI - Recommended Rules for Care and Operation of Heating Boilers* (with the 1970 Edition, the title was changed from "Rules for Inspection of Boilers." Section VI is administered by Section IV.)

1926 Section VII - Recommended Guidelines for the Care of Power Boilers* (Section VII is administered by Section I.)

1937 Section IX - Welding and Brazing Qualifications* (this section was originally part of Section VIII, but with the 1941 Edition, it became a separate section.)

1963 Section III - Nuclear Vessels* (with the 1971 Edition, this section became: Nuclear Power Plant Components and was divided in many Subsections.)

Subsection NCA - General Requirements
Subsection NB - Class 1 Components
Subsection NC - Class 2 Components
Subsection ND - Class 3 Components
Subsection NE - MC (Metal Containment) Components
Subsection NF - Component Supports
Subsection NG - Core Support Components

Subsection NH - Class 1 Components in Elevated Temperature Service

1968 Section VIII, Division 2 - Alternative Rules for Pressure Vessels* (with this edition, Section VIII became Section VIII, Division 1, Pressure Vessels.)

1968 Section X - Fiber-Reinforced Plastic Pressure Vessels*

1970 Section XI - Rules for Inservice Inspection of Nuclear Power Plant Components*

1971 Section V - Nondestructive Examination*

1975 Section III, Division 2 - Code for Concrete Reactor Vessels and Containments*

1997 Section VIII, Division 3 - Alternative Rules for High Pressure Vessels*

* Still in use and issued as 1998 Edition

Organization of the ASME Code Committees

The original ASME Boiler Code Committee was made up of seven members. The present ASME Code Committee has about 800 people involved at all levels. These members serve as individuals and are not representatives of companies (although their employers sponsor them and can influence them). If a member changes their affiliation, they maintain their position within the Code Committee if they are continued to be financially supported and want to serve. The structure of the ASME Boiler & Pressure Vessel Committee is:

Main Committee - Consists of up to 30 members comprised of boiler and pressure vessel manufacturers, users, materials suppliers, consulting engineers, insurance companies, and state and federal government agencies. Main duties are to review and approve technical changes and committee membership.

Executive Committee - Consists of 9 Main Committee members including the chairman and vice-chairman. Main duties are to approve policies and procedures and to review and consider new proposals and other activities.

Conference Committee - Consists of members (usually Chief Inspectors) from states, provinces, and other local jurisdictions who enforce the ASME Code within their jurisdiction.

ASME Staff - Consists of secretaries, assistant secretaries, executive personnel, typists, and others. Staff issues official letters and other communications, records and minutes, and provides many other services.

In general terms, the Code is divided into three different kinds of subcommittees. First, there is the component "book" committee. This is a subcommittee responsible for one of the component sections of the Code. At the present time, the sections and their subcommittees, subgroups, and special working groups are:

Section I - Subcommittee on Power Boilers (SC-I)

- SG - On Care of Power Boilers (SG-VII)
- SG - Piping
- SG - Design
- SG - Materials
- SG - General Requirements
- SG - Fabrication and Examination

Section III - Subcommittee on Nuclear Power (SC-III)

- SG - General Requirements
- SG - Materials, Fabrication, & Examination
- SG - Design
- SG - Pressure Relief
- SG - Containment Systems for Spent Fuel and High Level Waste Transport Packaging
- SWG- Editing and Review
- SWG- Seismic Piping Rules
- SWG- Environmental Effects

Joint ACI-ASME Committee on Concrete Components (SC-3C)

- SG - Materials, Construction, & Examination
- SG - Design

Section IV - Subcommittee on Heating Boilers (SC-IV)

- SG - Care and Operation of Heating Boilers (TG-VI)
- SG - Water Heaters
- SG - Cast Iron Boilers
- SG - Welded Boilers

Section VIII - Subcommittee on Pressure Vessels (SC-VIII)

- SG - General Requirements
- SG - Materials
- SG - Design
- SG - Fabrication & Inspection
- SWG- Heat Transfer Equipment
- SWG- High Pressure Vessels
- SWG- Toughness

Section X - Subcommittee on Fiber-Reinforced Plastic Pressure Vessels (SC-X)

Section XII* - Subcommittee on Transport Tanks (SC-XII)

- SG - General Requirements
- SG - Design and Materials
- SG - Fabrication & Inspection

* No Section XII issued at this time

Next, there are the service "book" committees. These are the subcommittees serving as a service committee to the component subcommittees but with the responsibility for a Section of the Code. At the present time, these Sections, subcommittees, and groups are:

Section II - Subcommittee on Materials (SC-II)

- SG - Ferrous Specifications
- SG - Nonferrous Alloys
- SG - Strength of Weldments
- SG - Toughness
- SG - Strength, Ferrous Alloys
- SG - External Pressure
- SG - International Material Specifications

Section V - Subcomm. on Nondestructive Examination (SC-V)

- SG - General Requirements/Personnel Requirements
- SG - Volumetric Methods
- SG - Surface Examination Methods

Section IX - Subcommittee on Welding (SC-IX)

- SG - Materials
- SG - General Requirements
- SG - Procedure Qualifications
- SG - Performance Qualifications
- SG - Brazing

Section XI - Subcommittee on Nuclear In-Service Inspection

- SG - Water-Cooled Systems
- SG - Gas-Cooled Systems
- SG - Evaluation Standards
- SG - Nondestructive Examination
- SG - Repair, Replacement, & Modification
- SG - Liquid-Metal Cooled Systems
- SWG- Plant Life Extension
- SWG- Editing and Review
- SWG- Low Temperature Heavy Water Reactors

Finally, there are the service committees. These are the subcommittees responsible for a specific area of technology which may be applicable to many Sections of the Code. At the present time, these committees are:

Subcommittee on Design (SC-D)

- SG - Openings
- SG - Design Analysis
- SG - Elevated Temperature Design
- SG - Fatigue Strength
- SWG- Bolted Flanged Joints

Subcommittee on Safety Valve Requirements (SC-SVR)

- SG - General Requirements
- SG - Testing
- SG - Design

Subcomm. on Boiler&Pressure Vessel Accreditation (SC-BPVA)

Subcommittee on Nuclear Accreditation (SC-NA)

ASME BOILER & PRESSURE VESSEL CODE

SECTIONS

I Power Boilers
II Material Specifications
III Nuclear Components
IV Heating Boilers
V Nondestructive Examination
VI Recommended Rules for Care and Operation of Heating Boilers
VII Recommended Guidelines for the Care of Power Boilers
VIII Pressure Vessels
Division 1
Division 2 - Alternative Rules
IX Welding and Brazing Qualifications
X Fiber-Reinforced Plastic Pressure Vessels
XI Rules for Inservice Inspection of Nuclear Power Plant Components

ADDENDA

Additions and revisions to individual section.
Published annually.
Effective 6 months after publication.

INTERPRETATIONS

Committee response to inquiries concerning interpretation of technical aspects of Code.
Issued semi-annually

CODE CASES

Proposed additions and revisions to the Code.
Rules for materials and constructions not covered by existing Code.
Issued semi-annually

SECTION VIII - DIVISION 1

CONTENTS

DIVISION 2 AS COMPARED TO DIVION 1

- More restrictive in the choice of materials.

- Limited to temperatures below the creep range 9 with exception of a Code Case).

- Higher stress allowables. Limited to 1/3 of specified minimum ~~yield~~ ult strength for Division 2 and 1/3.5 for Division 1.

- More precise design procedures.

- Some design details prohibited.

- Fabrication procedures specifically delineated.

- More complete examination, testing, and inspection requirements.

- Design by analysis and fatigue analysis rules provided.

- Less material cost. More NDE and design cost. Relative overall cost depends on many factors. Generally thicker vessels and those fabricated with high alloy more expensive materials are more economical with Div. 2.

- The intent has been to provide same safety margins.

FOREWORD TO SECTION VIII

- Rules do not address deterioration in service.

- Not a design handbook; engineering judgment must be exercised.

- Contains mandatory requirements, specific prohibitions, and non-mandatory guidance. Does not address all aspects, and what is not addressed should not be considered prohibited.

- Neither requires nor prohibits the use of computers for design. If used, the designer is responsible for correctness and accuracy.

- Does not fully address tolerances. Designer must determine based on engineering judgment and standard practice.

- Does not in general deal with care and inspection requirements.

- Does not approve, endorse, or recommend any propriety items.

- Revisions become mandatory after 6 months.

- Code Cases become effective at the date of approval.

- Mandatory requirements established at the date of contract.

- Material specifications approved by ASME are listed in Section II, Parts A and B. Appendix A of each part list editions of ASME and year dates of specifications that may be used in code construction.

- It is suggested that approval of revisions and Code Cases be obtained from the jurisdiction, prior to use.

- Laws and regulations of jurisdictions establish the mandatory applicability of Code rules.

Approval Date: August 12, 1993

See Numeric Index for expiration and any reaffirmation dates.

Case 2150
Large-End Cone-to-Cylinder Junction for 30 < α ≤ 60 Degrees
Section VIII, Division 1

Inquiry: Where radiography of a circumferential joint is not required, may a cone without a knuckle at the large end of the cone-to-cylinder junction having a half-apex angle α greater than 30 deg. be used in the construction of a vessel complying with the Section VIII, Division 1 rules without the special analysis specified in 1-5(g)?

Reply: It is the opinion of the Committee that when radiography of a circumferential joint is not required, a cone without a knuckle at the large end of a cone-to-cylinder junction having a half-apex angle α greater than 30 deg. may be used for Section VIII, Division 1 construction without the special analysis specified in 1-5(g), provided:

(a) Formulas (1) and (2) and Figs. 1 and 2 given below shall be used for calculating the localized stress at the discontinuity.

$$\sigma_\theta = \frac{PR}{t}\left(1 - Y\sqrt{\frac{R}{t}}\right) \qquad (1)$$

$$\sigma_x = \frac{PR}{t}\left(0.5 + X\sqrt{\frac{R}{t}}\right) \qquad (2)$$

where

σ_θ = membrane hoop stress plus average discontinuity hoop stress, psi

σ_x = membrane longitudinal stress plus discontinuity longitudinal stress due to bending, psi

X, Y = factors taken from Fig. 1 or 2 (or tabular values taken from Table 1 or 2)

P = internal pressure, psi

R = inside radius of the cylinder at large end of cone, in.

t = cylinder thickness, in.

(b) The half-apex angle α is not greater than 60 deg.

(c) The axial forces come solely from internal pressure acting on the closed ends. When other loads (such as wind loads, dead loads, etc.) are involved, the design shall be in accordance with U-2(g).

(d) σ_θ shall not be greater than $1.5S$ and σ_x shall not be greater than $3S$ where S is the maximum allowable stress value, in psi, obtained from the applicable table of stress values in Section II, Part D.

(e) After the required thickness for the shell has been determined by UG-27(c), and that for the cone by UG-32(g), the stress limits of (d) above must be checked with Formulas (1) and (2) using the calculated required thicknesses. If the limits of (d) are not met, the shell and cone thicknesses near the junction must be increased so that the limits of (d) are met. When additional thickness is required, the section of increased thickness shall extend a minimum distance from the junction as shown in Fig. 3.

(f) The angle joint between the cone and cylinder shall be designed equivalent to a double butt-welded joint and there shall be no weak zones around the joint.

(g) This Case number shall be shown on the Manufacturer's Data Report.

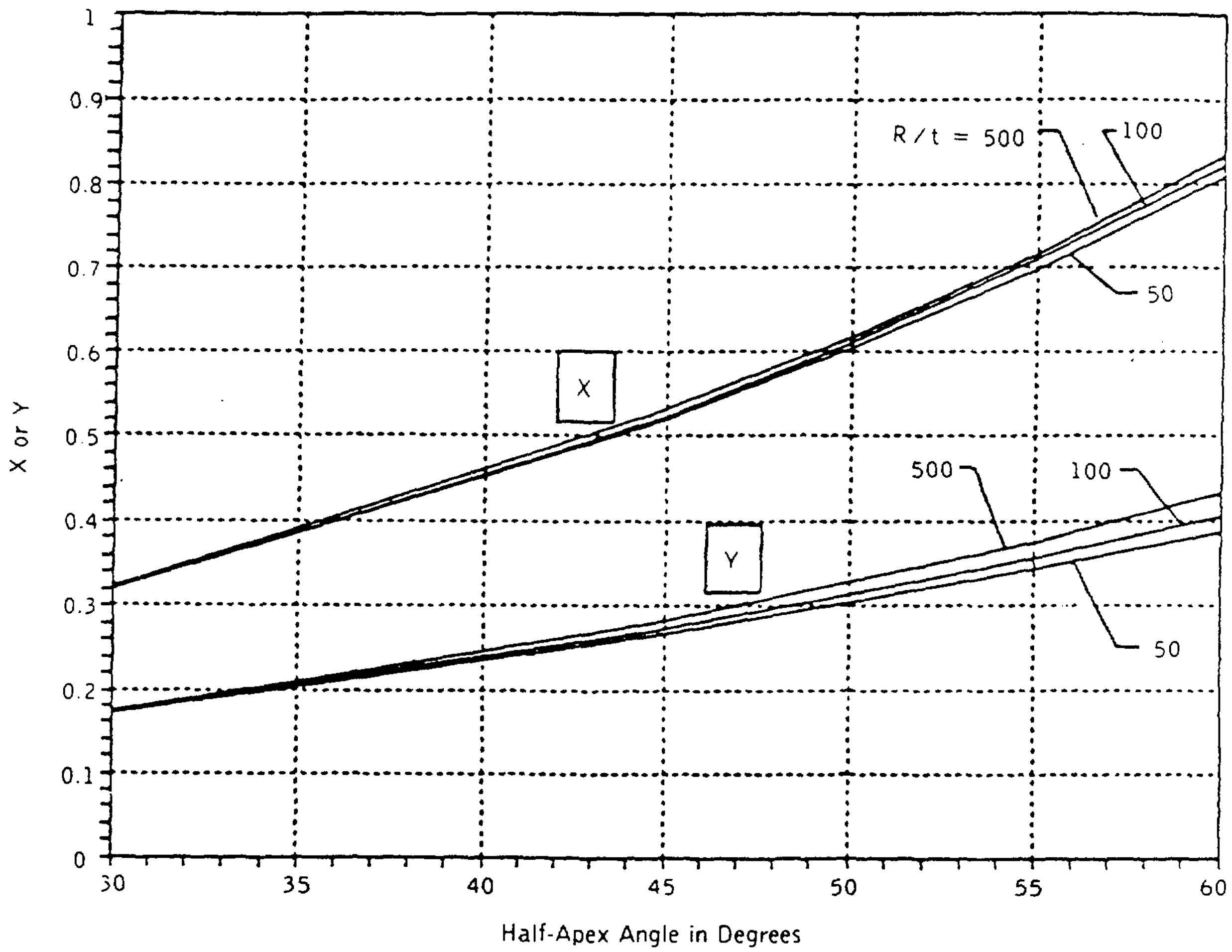


FIG. 1 X AND Y FOR CONE THICKENSS $= t$

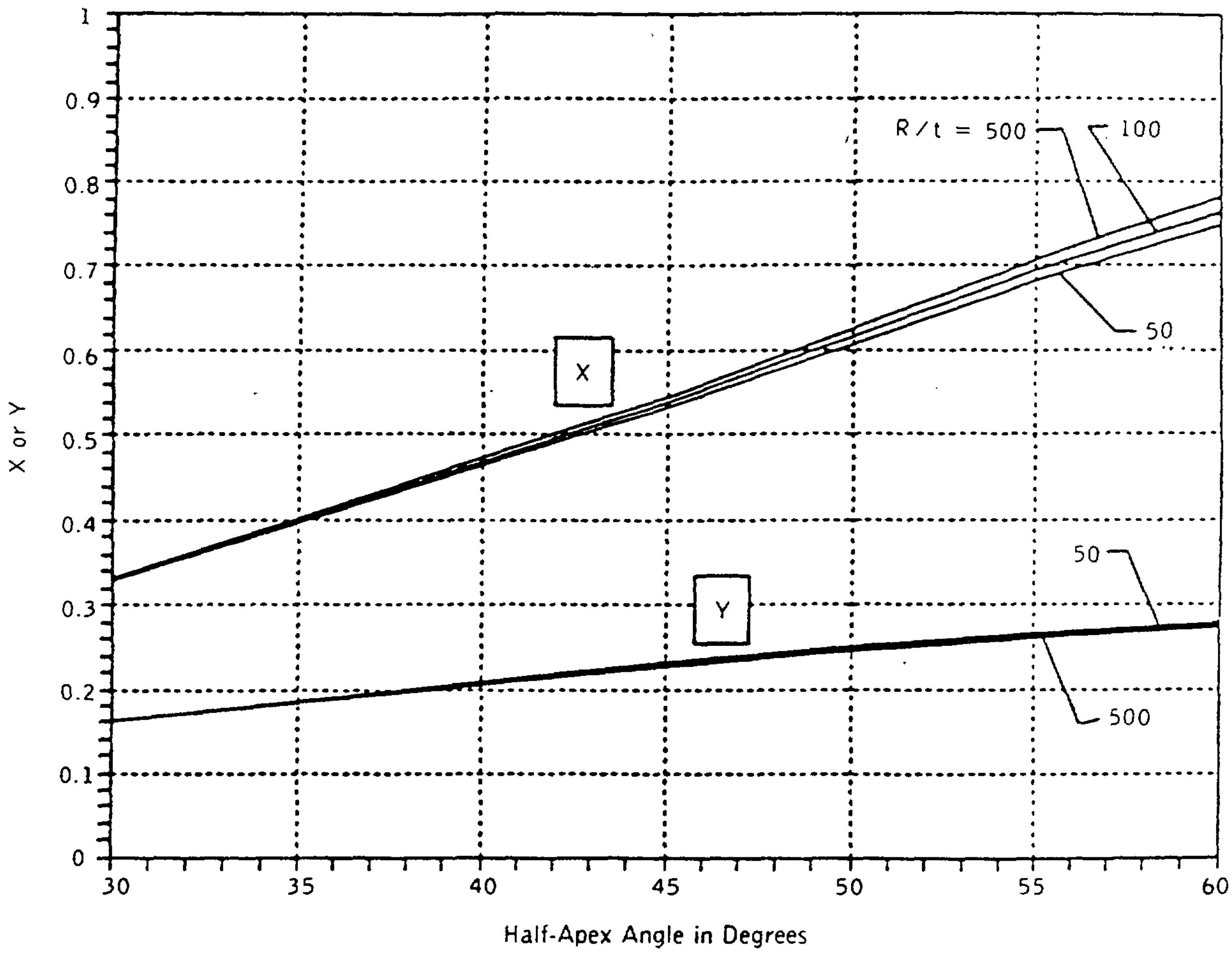


FIG. 2 *X* AND *Y* FOR CONE THICKNESS = $t/\cos\alpha$

TABLE 1
TABULAR VALUES FOR FIG. 1

Half-Apex Angle, deg.	Y			X		
	R/t = 500	R/t = 100	R/t = 50	R/t = 500	R/t = 100	R/t = 50
30	0.1750	0.1721	0.1698	0.3239	0.3211	0.3182
31	0.1822	0.1788	0.1762	0.3378	0.3348	0.3317
32	0.1894	0.1854	0.1826	0.3517	0.3484	0.3451
33	0.1966	0.1921	0.1890	0.3655	0.3621	0.3586
34	0.2038	0.1987	0.1955	0.3794	0.3758	0.3720
35	0.2110	0.2054	0.2019	0.3933	0.3894	0.3855
36	0.2182	0.2121	0.2083	0.4072	0.4031	0.3990
37	0.2254	0.2187	0.2147	0.4211	0.4168	0.4124
38	0.2327	0.2254	0.2211	0.4349	0.4304	0.4259
39	0.2399	0.2320	0.2275	0.4488	0.4441	0.4393
40	0.2471	0.2387	0.2339	0.4627	0.4578	0.4528
41	0.2543	0.2454	0.2403	0.4766	0.4714	0.4663
42	0.2615	0.2520	0.2468	0.4905	0.4851	0.4797
43	0.2687	0.2587	0.2532	0.5043	0.4988	0.4932
44	0.2759	0.2653	0.2596	0.5182	0.5124	0.5066
45	0.2831	0.2720	0.2660	0.5321	0.5261	0.5201
46	0.2918	0.2799	0.2733	0.5493	0.5432	0.5369
47	0.3005	0.2878	0.2806	0.5665	0.5604	0.5537
48	0.3092	0.2958	0.2878	0.5836	0.5775	0.5704
49	0.3179	0.3037	0.2951	0.6008	0.5947	0.5872
50	0.3266	0.3116	0.3024	0.6180	0.6118	0.6040
51	0.3365	0.3204	0.3104	0.6379	0.6314	0.6232
52	0.3464	0.3291	0.3183	0.6577	0.6509	0.6423
53	0.3563	0.3379	0.3263	0.6776	0.6705	0.6615
54	0.3662	0.3466	0.3342	0.6974	0.6900	0.6806
55	0.3761	0.3554	0.3422	0.7173	0.7096	0.6998
56	0.3877	0.3654	0.3512	0.7411	0.7322	0.7217
57	0.3993	0.3754	0.3602	0.7649	0.7548	0.7436
58	0.4110	0.3854	0.3691	0.7887	0.7773	0.7654
59	0.4226	0.3954	0.3781	0.8125	0.7999	0.7873
60	0.4342	0.4054	0.3871	0.8363	0.8225	0.8092

TABLE 2
TABULAR VALUES FOR FIG. 2

Half-Apex Angle, deg.	Y R/t = 500	Y R/t = 100	Y R/t = 50	X R/t = 500	X R/t = 100	X R/t = 50
30	0.1601	0.1604	0.1613	0.3325	0.3297	0.3264
31	0.1646	0.1650	0.1660	0.3468	0.3437	0.3402
32	0.1691	0.1696	0.1708	0.3611	0.3578	0.3539
33	0.1735	0.1742	0.1755	0.3754	0.3718	0.3677
34	0.1780	0.1788	0.1802	0.3897	0.3858	0.3815
35	0.1825	0.1834	0.1849	0.4040	0.3998	0.3952
36	0.1870	0.1880	0.1897	0.4183	0.4139	0.4090
37	0.1915	0.1926	0.1944	0.4326	0.4279	0.4228
38	0.1959	0.1972	0.1991	0.4468	0.4419	0.4365
39	0.2004	0.2018	0.2038	0.4611	0.4559	0.4503
40	0.2049	0.2064	0.2086	0.4754	0.4700	0.4641
41	0.2094	0.2110	0.2133	0.4897	0.4840	0.4778
42	0.2139	0.2156	0.2180	0.5040	0.4980	0.4916
43	0.2183	0.2202	0.2227	0.5183	0.5120	0.5054
44	0.2228	0.2248	0.2275	0.5326	0.5261	0.5191
45	0.2273	0.2294	0.2322	0.5469	0.5401	0.5329
46	0.2309	0.2331	0.2359	0.5627	0.5555	0.5479
47	0.2345	0.2367	0.2397	0.5786	0.5709	0.5628
48	0.2382	0.2404	0.2434	0.5944	0.5864	0.5778
49	0.2418	0.2440	0.2472	0.6103	0.6018	0.5927
50	0.2454	0.2477	0.2509	0.6261	0.6172	0.6077
51	0.2484	0.2508	0.2540	0.6422	0.6325	0.6224
52	0.2515	0.2538	0.2572	0.6583	0.6479	0.6371
53	0.2545	0.2569	0.2603	0.6744	0.6632	0.6518
54	0.2576	0.2599	0.2635	0.6905	0.6786	0.6665
55	0.2606	0.2630	0.2666	0.7066	0.6939	0.6812
56	0.2628	0.2652	0.2688	0.7215	0.7078	0.6942
57	0.2649	0.2673	0.2710	0.7365	0.7217	0.7073
58	0.2671	0.2695	0.2733	0.7514	0.7356	0.7203
59	0.2692	0.2716	0.2755	0.7664	0.7495	0.7334
60	0.2714	0.2738	0.2777	0.7813	0.7634	0.7464

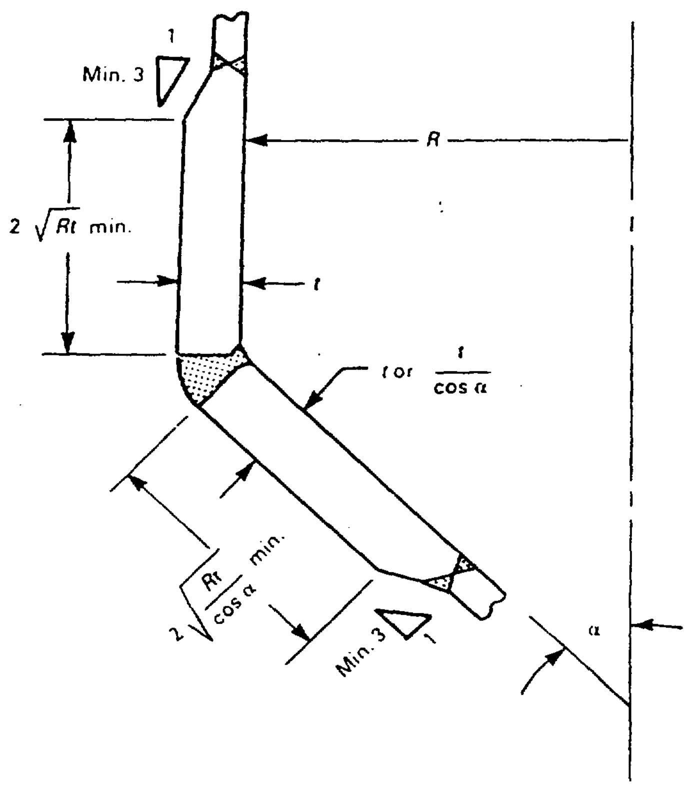

1
Min. 3
R
2 $\sqrt{Rt}$ min.
t
t or $\frac{t}{\cos\alpha}$
2 $\sqrt{\frac{Rt}{\cos\alpha}}$ min.
Min. 3
1
α

FIG. 3

Approval Date: February 14, 1994
See Numeric Index for expiration and any reaffirmation dates.

Case 2168
Alternative Method to Area Replacement Rules for Openings Under Internal Pressure
Section VIII, Division 1

Inquiry: In Section VIII, Division 1 vessels, may openings in cylindrical shells subjected to internal pressure be designed to rules other than those given in UG-37?

Reply: It is the opinion of the Committee that Section VIII, Division 1 vessel openings in cylindrical shells subjected to internal pressure as shown in Fig. UW-16.1 with full penetration welds and integral reinforcement may be designed for internal pressure using the following rules instead of those given in UG-37.

(a) Nomenclature (See Fig. 1)

B_1 = 162 for $t_n/t \leq 1.0$
= 54 for $t_n/t > 1.0$
B_2 = 210 for $t_n/t \leq 1.0$
= 318 for $t_n/t > 1.0$
d_m = mean diameter of connecting pipe, in. (corroded condition), [see (b)(9)]
D_m = mean diameter of cylindrical vessel, in. (corroded condition)
t_p = nominal wall thickness of connecting pipe, less corrosion allowance, in.
t_n = nominal wall thickness of nozzle, less corrosion allowance, in.
t = nominal wall thickness of vessel, less corrosion allowance, in.
t_r = required thickness of vessel wall calculated per UG-27(c)(1), with E = 1.00, in.
L = axial length of nozzle with thickness t_n, in.
$\lambda = (d_m/D_m)(D_m/t)^{1/2}$
t_{std} = nominal wall thickness of ANSI B36.10 standard weight pipe, in.

(b) The following conditions shall be met:

(1) Maximum design temperature shall not exceed 650°F.

(2) Material shall be carbon or low-alloy steel with allowable stress in tension per Table 1A of Section II, Part D, not exceeding 17.5 ksi.

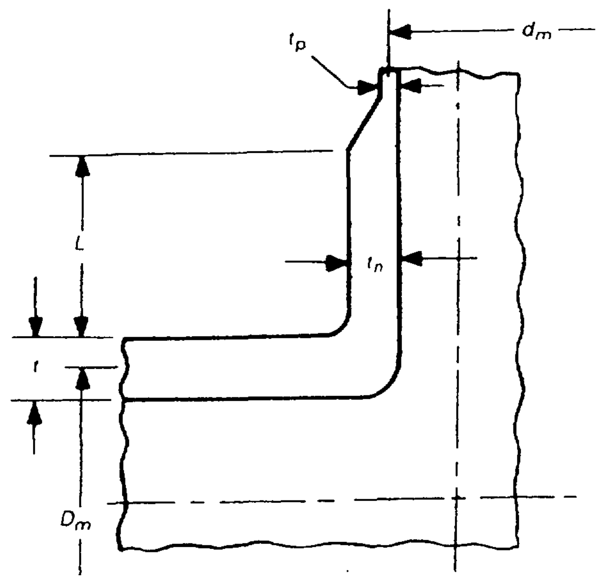


FIG. 1 NOMENCLATURE

(3) The openings shall not exceed the following:

(a) for vessels of diameter 60 in. and less, one half the vessel diameter, but not to exceed 20 in.;

(b) for vessels of diameter over 60 in., one-third the vessel diameter, but not to exceed 40 in.

(4) The ratio of vessel diameter (D_m) to vessel thickness (t) shall not exceed 250.

(5) Cyclic loading is not a controlling design requirement (see UG-22).

(6) The opening is in a cylindrical vessel. It shall be located not less than 1.8 $(D_m t)^{1/2}$ from any other gross structural discontinuity such as a head or stiffener.

(7) The spacing between the centerlines of the openings and any other opening is not less than three times their average diameter.

(8) The opening is circular in cross section and its axis is normal to the surface of the cylindrical vessel.

(9) If $L < 0.5\ (d_m t_n)^{1/2}$, use $t_n = t_p$ in Eqs. (1) and (2) below.

(10) t_n shall not be less than $\frac{7}{8}$ t_{std} through an axial length of $(d_m t_n)^{1/2}$. The other applicable rules of UG-45 shall be met.

(11) The opening shall satisfy Eqs. (1) and (2), as follows:

$$\frac{2 + 2\left(\frac{d_m}{D_m}\right)^{3/2}\left(\frac{t_n}{t}\right)^{1/2} + 1.25\lambda}{1 + \left(\frac{d_m}{D_m}\right)^{1/2}\left(\frac{t_n}{t}\right)^{3/2}} \leq 2.95\left(\frac{t}{t_r}\right) \qquad (1)$$

$$\frac{\left[B_1\left(\frac{t_n}{t}\right)^2 + 228\left(\frac{t_n}{t}\right)\left(\frac{d_m}{D_m}\right) + B_2\right]\lambda + 155}{108\lambda^2 + \left[228\left(\frac{d_m}{D_m}\right)^2 + 228\right]\lambda + 152} \geq (0.93 + 0.005\lambda)\left(\frac{t_r}{t}\right) \qquad (2)$$

(12) This Case number shall be shown on the Manufacturer's Data Report.

(c) This Code Case was developed from WRC Bulletin 335.

Code Case 2168: Alternate Method to Area Replacement
Rules for Openings Under Internal Pressure Section VIII, Division 1 *

Nomenclature (See Fig. 1)

dm = mean diameter of connecting pipe (corroded)
Dm = mean diameter of vessel (corroded)
tp = wall thk of connecting pipe (corroded)
tn = wall thk of nozzle (corroded)
t = wall thk of vessel (corroded)
tr = req. thk of vessel per UG-27(c)(1)
L = axial length of nozzle with thk = tn
tstd = nominal thk of ANSI B36.10 standard wt. pipe
P = internal design pressure
S = max. allowable stress

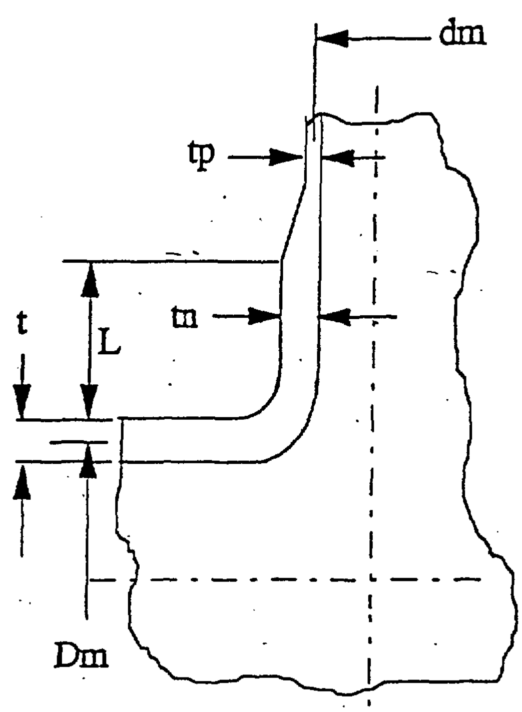


Fig. 1

Inputs:

Nozzle Mark:	Manifold	
Description:	Sea Water Intake	
Units are in:	E	M (Metric), E (English)
dm =	90.905	inches
Dm=	159.5	inches
tp =	0.905	inches
tn =	0.905	inches
t =	3.495	inches
L =	18	inches
tstd =	0.5	inches
P =	275	psi
S =	17500	psi

* Note: Nozzle detail must be with full penetration welds and integral reinforcement as defined in Fig. UW-16.1.

Outputs:

Check following required conditions per C. C. 2168

Condition Required	Condition Met ?
(1) Maximum design temperature shall not exceed 650 °F.	Yes
(2) Design must be for ASME Sect. VIII Div. 1 vessels only. Material must be carbon or low-allow steel with allowable stress less than or equal to 17.5 ksi per Table 1A of Section II, Part D.	Yes
(3) The openings shall meet the following: (a) for vessels of diameter 60 in. or less, the opening shall not exceed one half the vessel's diameter and shall not exceed 20 in. (b) for vessels of diameter over 60 in, the opening shall not exceed one third the vessel's diameter and shall not exceed 40 in.	No!
(4) The ratio of (Dm) to vessel thickness(t) shall not exceed 250.	Yes
(5) The vessel is not in cyclic service.	Yes
(6) The opening is in a cylindrical vessel and is located from any other gross structural discontinuity(head, stiffener, cone-cylinder junction, etc.) by a distance no less than $1.8(Dmt)^{1/2}$	Yes
(7) The spacing between the centerline of an opening and any other opening shall not be less than three times their average diameter.	Yes
(8) The opening must be circular and radial.	Yes
(9) If $L < 0.5(dmtn)^{1/2}$, use tn = tp in equations (1) and (2) below.	input tn & tp used
(10) tn shall not be less than 7/8 tstd through an axial length of $(dmtn)^{1/2}$. Applicable rules of UG-45 (minimum nozzle wall thickness) shall be met.	Yes
(11.1) The opening shall satisfy Eq. (1) below $$\frac{2+2\left(\frac{dm}{Dm}\right)^{3/2}\left(\frac{tn}{t}\right)^{1/2}+1.25\lambda}{1+\left(\frac{dm}{Dm}\right)^{1/2}\left(\frac{tn}{t}\right)^{3/2}} \le 2.95\left(\frac{t}{tr}\right)$$ 6.59 ≤ 8.33	Yes
(11.2) The opening shall satisfy Eq. (2) below $$\frac{\left[B_1\left(\frac{tn}{t}\right)^2+228\left(\frac{tn}{t}\right)\left(\frac{dm}{Dm}\right)+B_2\right]\lambda+155}{108\lambda^2+\left[228\left(\frac{dm}{Dm}\right)^2+228\right]\lambda+152} \ge (0.93+0.005\lambda)\left(\frac{tr}{t}\right)$$ .389 > .332	Yes
(12) This Case no. shall be shown on the Manufacturer's Data Report.	Yes

Approval Date: August 11, 1997

See Numeric Index for expiration and any reaffirmation dates.

Case 2243
Local Thin Area in Cylindrical Shell
Section VIII, Division 1

Inquiry: Under what condition for Section VIII, Division 1 vessels, may the thickness of local areas of cylindrical shells under internal pressure be less than that required by UG-27?

Reply: It is the opinion of the Committee that for Section VIII, Division 1 vessels, the thickness of local areas of cylindrical shells under internal pressure may be less than that required by UG-27 provided the local thin areas satisfy the following rules.

(a) Nomenclature (see Fig. 1):

LTA = local thin area
t_L = minimum thickness of LTA, in.
L = projected axial length of LTA, in.
C = projected circumferential length of LTA, in.
θ = see Fig. 1

See UG-27(b) for other nomenclature used in this Code Case.

(b) Local thin area or areas on inside or outside surfaces of a cylindrical shell designed for internal pressure are acceptable provided the following rules are satisfied.

(1) Single LTA

(a) The single LTA shall satisfy the following equations.

$$\frac{t_L}{t} \geq 0.9 \qquad (1)$$

$$L \leq \sqrt{(Rt)} \qquad (2)$$

$$C \leq 2L \qquad (3)$$

$$t - t_L \leq \frac{3}{16} \text{ in.} \qquad (4)$$

TABLE 1
MAXIMUM METAL TEMPERATURE

Table in Which Material is Listed	Temperature, °F
UCS-23	700
UNF-23.1	300
UNF-23.2	150
UNF-23.3	900
UNF-23.4	600
UNF-23.5	600
UHA-23	800
UHT-23	700

(b) Any edge of an LTA shall not be closer than 2.5 $\sqrt{(Rt)}$ from a structural discontinuity such as a head or stiffener.

(c) The minimum axial distance between the edge of the LTA and the edge of any nonreinforced opening shall be equal to or greater than the inside diameter of the opening plus $\sqrt{(Rt)}$.

(d) The minimum axial distance between the edge of the LTA and the reinforcement limit of a reinforced opening shall be equal to or greater than $\sqrt{(Rt)}$.

(e) The blend between the LTA and the thicker surface shall be with a taper length not less than three times the LTA depth as shown in Fig. 1, sketch (b). The minimum bottom blend radius shall be equal to or greater than two times the LTA depth as shown in Fig. 1, sketch (b).

(f) The longitudinal stresses on the LTA from mechanical loads other than internal pressure shall not exceed 0.3S.

(g) The maximum design temperature shall not exceed the maximum temperature limit specified in Table 1.

(h) The thickness at the LTA shall meet the requirements of UG-23(b) and/or UG-28 as applicable.

(i) Provisions of this Case do not apply to corrosion resistant lining/overlays.

(2) Multiple LTA. A pair of local thin areas with finished axial length L_1 and L_2 are acceptable if the individual LTA satisfies the requirements of (b)(1) above and one of the following conditions [(b)(2)(a) or (b)(2)(b)] is met.

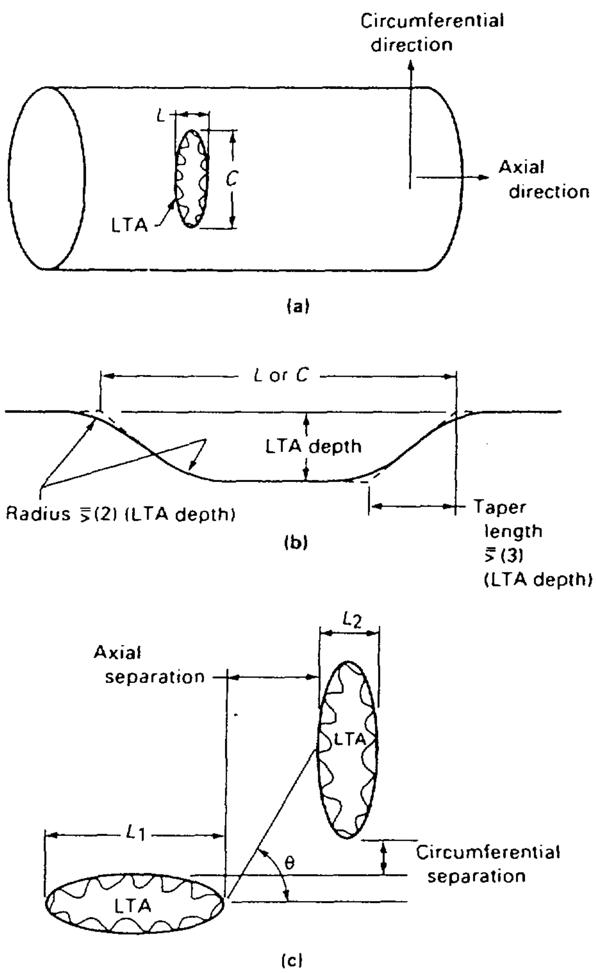


FIG. 1 NOMENCLATURE

(a) When $\theta \leq$ 45 deg., the minimum axial separation [see Fig. 1, sketch (c)] shall be the greater of:

$$\frac{(1.0 + 1.5 \cos \theta)(L_1 + L_2)}{2} \quad \text{or} \quad 2t$$

(b) When $\theta >$ 45 deg., both of the following shall be met:

(1) The minimum axial separation shall be equal to or greater than:

$$\frac{2.91 \cos \theta (L_1 + L_2)}{2}$$

(2) The minimum circumferential separation shall be equal to or greater than $2t$.

(c) Multiple pairs of LTA are acceptable provided all pairs meet the rules of a single pair specified in (b)(2) above.

(d) Multiple local thin areas may be combined as a single LTA. The resultant single LTA is acceptable if it satisfies the rules of (b)(1) above.

(e) This Case shall not be applied to Part UF vessels.

(f) All other applicable Code requirements shall be met.

(g) This Case number shall be shown on the Manufacturer's Data Report.

Approval Date: May 4, 1999

See Numeric Index for expiration and any reaffirmation dates.

Case 2310
Local Thin Areas in Spherical Segments of Shells
Section VIII, Division 1

Inquiry: Under what conditions for Section VIII, Division 1 vessels may the thickness of local areas in spherical segments of shells under internal pressure be less than that required by UG-27 and UG-32?

Reply: It is the opinion of the Committee that for Section VIII, Division 1 vessels, the thickness of local areas of spherical segments of shells (such as spherical vessels, hemispherical heads, and the spherical portion of torispherical and ellipsoidal heads) under internal pressure for vessels or pressure on the concave side for heads may be less than that required by UG-27(d) or UG-32(d), UG-32(e), and UG-32(f) provided the local thin areas satisfy the following conditions:

(a) Nomenclature

t = required shell thickness per UG-27(d) or UG-32(d), UG-32(e), or UG-32(f)

LTA = local thin area

t_L = minimum thickness of LTA, in.

D_L = maximum dimension of LTA, in.

R = inside radius for spherical segment, for ellipsoidal heads $R = K_oD$ where K_o is from Table UG-33.1

D = per UG-32

(b) LTAs on the inside or outside surface of spherical segments of shells designed for internal pressure are acceptable provided the following conditions are satisfied.

(1) Single LTAs

(a) The single LTA shall satisfy the following equations

$$\frac{t_L}{t} \geq 0.9 \qquad (1)$$

$$D_L \leq \sqrt{Rt} \qquad (2)$$

$$t - t_L \leq 3/16 \text{ in.} \qquad (3)$$

(b) The minimum distance between the edge of the LTA and the edge of any unreinforced opening shall be equal to or greater than the inside diameter of the opening plus $\sqrt{Rt}$.

(c) The minimum distance between the edge of the LTA and the reinforcement limit of a reinforced opening shall be equal to or greater than $\sqrt{Rt}$.

(d) The edges of a LTA shall not be closer than $2.5\sqrt{Rt}$ from a structural discontinuity.

(e) A constant thickness junction between head and cylindrical shell is not considered to be a structural discontinuity for LTA rules.

(f) The blend between the LTA and the thicker surface shall be with a taper length not less than three times the LTA depth. The minimum bottom blend radius shall be equal to or greater than two times the LTA depth. The blend requirements are shown in Fig. 1.

(g) The LTA for a torispherical head must lie entirely within the spherical portion of the head. See Fig. 2.

(h) The LTA for an ellipsoidal head must lie entirely within a circle, the center of which coincides with the axis of the vessel and the diameter of which is equal to 80% of the shell inside diameter. See Fig. 3.

(i) The LTA for a hemispherical head is acceptable within any portion of the head except as limited by (d) above. See Fig. 4.

(j) The maximum design temperature shall not exceed the maximum temperature limits specified in Table 1.

(k) The thickness at the LTA shall meet the requirements of UG-28(d) or UG-33 as applicable.

TABLE 1
MAXIMUM METAL TEMPERATURE

Temperature, °F	Table in Which Material is Listed
700	UCS-23
300	UNF-23.1
150	UNF-23.2
900	UNF-23.3
600	UNF-23.4
600	UNF-23.5
800	UHA-23
700	UHT-23

(l) The provisions of this case do not apply to the torus portion of either a torispherical or ellipsoidal head, to flat heads, or to conical heads.

(m) The provisions of this Case do not apply to corrosion-resistant linings or overlays.

(2) Multiple LTAs

(a) Multiple LTAs may be combined and evaluated as a single LTA. The encompassed areas of the combined LTAs shall be within the D_L dimension.

(b) Each LTA in the encompassed area shall meet the rules of (b)(1) above.

(c) Multiple LTAs may be treated as single LTAs provided their edges are no closer than $2.5\sqrt{Rt}$.

(3) Other Requirements

(a) This case shall not be applied to Part UF Vessels.

(b) All other applicable Code requirements shall be met.

(c) This Case number shall be shown on the Manufacturer's Data Report.

Approval Date: May 20, 1998

See Numeric Index for expiration and any reaffirmation dates.

Case 2260
Alternative Rules for Design of Ellipsoidal and Torispherical Formed Heads
Section VIII, Division 1

Inquiry: For Section VIII, Division 1 vessels, may ellipsoidal and torispherical formed heads subjected to internal pressure be designed to rules other than those given in UG-32(d), 1-4(c) and UG-32(e), 1-4(d) respectively?

Reply: It is the opinion of the Committee that Section VIII, Division 1 vessel ellipsoidal and torispherical formed heads subjected to internal pressure may be designed using the following rules in lieu of those given in UG-32(d), 1-4(c), and UG-32(e), 1-4(d) respectively.

(a) Nomenclature

E_{RT} = Modulus of elasticity at 70°F, psi. The value of E_{RT} for ferrous materials shall be taken from Table UF-27. For nonferrous materials except zirconium, the value of E_{RT} shall be taken from Section II, Part D, Tables TM-2, TM-3, TM-4, or TM-5. For zirconium, the value of E_{RT} shall be taken from Section II, Part D, Figures NFZ-1 and NFZ-2.

E_T = Modulus of elasticity at maximum design temperature, psi. The value of E_T for ferrous materials shall be taken from Table UF-27. For nonferrous materials except zirconium, the value of E_T shall be taken from Section II, Part D, Tables TM-2, TM-3, TM-4, or TM-5. For zirconium, the value of E_T shall be taken from Section II, Part D, Figures NFZ-1 and NFZ-2. If the maximum design temperature is greater than that shown in the above tables or figures, then use the value of E_T corresponding to the maximum temperature given in the above tables or figures.

h = one-half of the length of the inside minor axis of the ellipsoidal head, or the inside depth of the ellipsoidal head measured from the tangent line (head-bend line), in.

$D/2h$ = ratio of the major to the minor axis of ellipsoidal heads, which equals the inside diameter of the head skirt divided by twice the inside height of the head.

See UG-32(c) for other nomenclature.

TABLE 1

t/L	M for $r/D = 0.06$	M for $r/D = 0.07$	M for $r/D = 0.08$	M for $0.08 < r/D \le 0.2$
0.002	1.00	1.00	1.00	1.00
0.004	1.00	1.00	1.00	1.00
0.006	1.28	1.00	1.00	1.00
0.008	1.41	1.20	1.00	1.00
0.010	1.41	1.26	1.10	1.00
0.012	1.38	1.25	1.13	1.00
0.016	1.31	1.21	1.12	1.00
0.020	1.25	1.17	1.08	1.00
0.030	1.14	1.08	1.01	1.00
0.040	1.07	1.01	1.00	1.00
0.060	1.00	1.00	1.00	1.00

(b) Torispherical Heads. The minimum required thickness of a torispherical head having $0.002 \le t/L \le 0.06$ shall be larger of the thicknesses calculated by the following formulas (1) and (2).

$$t = \frac{PLM}{2SE - 0.2P} \qquad (1)$$

$$t = \frac{3PLKE_{RT}}{4S_aE_T} \qquad (2)$$

The value of S_a shall be 115,000 psi for all ferrous and nonferrous materials except for aluminum, aluminum alloys, copper, copper alloys, titanium and zirconium, for which the value of S_a shall be calculated by the following formula (3).

$$S_a = \frac{115{,}000 \times E_{RT}}{30 \times 10^6} \qquad (3)$$

The value of M shall be obtained from Table 1. Interpolation may be used for r/D values which fall

TABLE 2

t/L	K for r/D = 0.06	K for r/D = 0.08	K for r/D = 0.10	K for r/D = 0.14	K for r/D = 0.17	K for r/D = 0.20
0.002	7.87	6.29	5.24	3.95	3.31	2.81
0.004	6.77	5.60	4.69	3.49	2.93	2.50
0.006	6.04	5.14	4.38	3.27	2.73	2.33
0.008	5.51	4.78	4.14	3.13	2.60	2.21
0.010	5.11	4.49	3.93	3.02	2.51	2.13
0.012	4.79	4.25	3.76	2.93	2.44	2.06
0.016	4.31	3.87	3.47	2.77	2.33	1.97
0.020	3.96	3.58	3.24	2.63	2.24	1.91
0.030	3.48	3.10	2.84	2.37	2.07	1.79
0.040	3.32	2.97	2.69	2.23	1.95	1.72
0.060	3.12	2.80	2.56	2.17	1.92	1.71

The value of K shall be obtained from Table 2. Interpolation may be used for r/D values which fall within the range of the tabulated values. No extrapolation of the values is permitted.

For designs where $t/L > 0.06$, the rules of UG-32(f) or 1-3 shall be used. In 1-3 formulas (1) and (2), R shall be replaced with L.

(c) Ellipsoidal Heads. The minimum required thickness of an ellipsoidal head with $D/2h$ ratio less than or equal to 2.0 shall be established as an equivalent torispherical head using the rules given in (b) above. An acceptable approximation of a 2:1 ellipsoidal head is one with a knuckle radius of $0.17D$ and a spherical radius of $0.9D$.

(d) The requirement of UHT-32 does not apply.

(e) Size of the finished openings in the knuckle area shall not exceed the lesser of $2\frac{3}{8}$ in. or 0.5 r. For an ellipsoidal head, the knuckle area is the area located outside a circle whose center coincides with the center of the head and whose diameter is equal to 80% of the head inside diameter.

TABLE 3
MAXIMUM METAL TEMPERATURE

Table in Which Material is Listed	Temperature, °F
UCS-23	700
UNF-23.1	300
UNF-23.2	150
UNF-23.3	900
UNF-23.4	600
UNF-23.5	600
UHA-23	800
UHT-23	700

(f) This Case has been developed for fatigue life of 400 full pressure range cycles with nonintegral attachments and 1000 full pressure range cycles with integral attachments. See U-2(g) for design of heads exceeding the above fatigue life.

(g) The rules of this Code Case may result in relatively high local strains in the knuckle. The effect of these high strains in areas where structural attachments are located shall be considered. See U-2(g).

(h) This Case shall not be used for Part UCI and Part UCD heads.

(i) The maximum design temperature shall not exceed the maximum temperature limit specified in Table 3.

(j) All other applicable Code requirements including those of UG-32 shall be met.

(k) This Case number shall be shown on the Manufacturer's Data Report.

M A W P COMPARISION

MATL	DIV1 CODE CASE	DIV1 EXISTING	DIV2 CODE CASE	DIV2 EXISTING
	t/L=0.002	r/D=0.06		
304SS	39	43	39	23
516-70	39	40	39	28
517-F	39	45	39	47
5083-O	13	23	13	12
	t/L=.002	r/D=.17		
304SS	75	68	80	50
516-70	70	63	87	59
517-F	93	103	93	96
5083-O	29	36	29	30
	t/L=0.01	r/D=0.06		
304SS	267	212	284	167
516-70	248	198	300	205
517-F	300	226	300	330
5083-O	71	113	100	105
	t/L=0.01	r/D=0.17		
304SS	376	338	400	278
516-70	350	314	466	325
517-F	574	517	574	532
5083-O	200	180	191	167
	t/L=0.02	r/D=0.06		
304SS	602	425	640	450
516-70	560	395	746	525
517-F	767	452	767	850
5083-O	256	226	256	265
	t/L=0.02	r/D=0.17		
304SS	752	674	800	640
516-70	700	628	932	745
517-F	1150	1031	1322	1220
5083-O	400	359	441	385

Codes and Standards

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October 22, 1999

Mr. Guido G. Karcher
Consulting Engineer
38 North Boom Way
Little Egg Harbor, NJ 08087-2312

Subj: Section VIII, Division 1 (1998 Edition); Appendix 1, 1-7(b)

Ref: Your letter of 6/28/99

Item: BC99-370

Dear Mr. Karcher:

Our understanding of the questions in your inquiry and our replies are as follows:

Question 1: Is Appendix 1-7(b) of Section VIII, Division 1 applicable only when both of the following conditions exist:

1. The vessel diameter is greater than 60 in. ID, and
2. The nozzle diameter exceeds the greater of 40 in. ID and $3.4\sqrt{Rt}$.

Reply 1: Yes.

Question 2: For cases where both conditions in Question 1 exist and the R_n/R ratio exceeds 0.7, may the rules of U-2(g) be applied in lieu of Appendix 1-7(b) rules?

Reply 2: Yes.

BCC: T.P. PASTOR
K. MOKHTARIAN
U.R. MILLER
G.B. KOMORA
D.A. CANONICO

Very truly yours,

Alan Roby

Alan Roby
Secretary; Subcommittee on Pressure Vessels
(212) 591-8538

Approval Date: May 4, 1999

See Numeric Index for expiration and any reaffirmation dates.

Case 2055-2
Pneumatic Testing of Pressure Vessels, UG-20 Section VIII, Division 1

Inquiry: Under what conditions may pressure vessels, fabricated under the provisions of Section VIII, Division 1, be tested pneumatically in lieu of the hydrostatic test set forth in UG-20(f)(2)?

Reply: It is the opinion of the Committee that pneumatic test provisions of UG-100 and the requirements of UG-20(f) may be used provided the following additional requirements are met:

(a) The test pressure is at least 1.3 MAWP but shall not exceed 1.3 times the basis for calculated test pressure defined in Appendix 3-2;

(b) The MAWP is no greater than 500 psi;

(c) The following thickness limitations shall apply:

(1) For butt joints, the nominal thickness at the thickest welded joint shall not exceed $\frac{1}{2}$ in.

(2) For corner or lap welded joints, the thinner of the two parts joined shall not exceed $\frac{1}{2}$ in.

(d) This Case number shall be shown on the Manufacturer's Data Report.

CAUTIONARY NOTE: The vessel(s) should be tested in such a manner as to ensure personnel safety from a release of the total energy of the vessel(s).

New Interpretations

Interpretation: VIII-1-95-160

Subject: Section VIII, Division 1 (1995 Edition, 1995 Addenda); UCS-66(c) and UCS-68(c)

Date Issued: December 17, 1996

File: BC96-378

Question: May the additional 30°F reduction in Charpy impact testing exemption temperature requirement of UCS-68(c) in Section VIII, Division 1 be applied to the provision of UCS-66(c), which allows ANSI B16.5 flanges exemption from Charpy impact requirements when the MDMT is no colder than −20°F?

Reply: Yes.

Note: This Interpretation was inadvertently omitted from Volume 40 of the Interpretations.

Interpretation: VIII-1-98-13

Subject: Section VIII, Division 1 (1995 Edition); UCS-66(a)(1)(b) and Appendix 2, Fig. 2-4 Sketch (3)

Date Issued: January 20, 1998

File: BC96-342

Question: A vessel has a carbon steel cylinder which is welded by two circumferential welds, as per Fig. 2-4 sketch (3) in Section VIII, Division 1, to a custom forged carbon steel slip-on flange with a hub. Is the governing thickness t_g for impact test exemption at these fillet weld joints the thinner of the nozzle wall thickness and the flange thickness?

Reply: Yes.

Codes and Standards

ASME International

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September 10, 1999

Mr. Ronald J. Burrage
Senior Design Engineer
Plant Maintenance Service Corporation
3000 Fite Road
P.O. Box 280883
Memphis, TN 38168-0883

Subj: Section VIII, Division 1 (1998 Edition, 1998 Addenda); U-1(e)(2)

Ref: ~~Your~~ letter of 5/25/99

Item: BC99-373

Dear Mr. Burrage:

Our understanding of the question in your inquiry and our response is as follows:

Question: Are the calculations for lifting of the vessel considered a part of Code, Section VIII, Division 1, design calculations and required to be made available to the Authorized Inspector?

Reply: No.

Very truly yours,

Alan Roby

Alan Roby
Secretary; Subcommittee on Pressure Vessels
(212) 591-8538

BCC:
R.M. Elliott
J.P. Swezy
K. Mokhtarian

T.P. Pastor
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Codes and Standards

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October 25, 1999

Mr. Ron Clarkson
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Carson, CA 90745

Subj: Section VIII, Division 1 (1998 Edition, 1998 Addenda); Fig. UW-13.1, Sketch (n), and UW-33(a)

Ref: Your letter of 4/7/99

Item: BC99-300

Dear Mr. Clarkson:

Our understanding of the question in your inquiry and our reply are as follows:

Question: May the center line misalignment of a Category B joint of materials of different thicknesses, as determined by Fig. UW-13.1, sketch (n) of Section VIII, Division 1, be increased by the amount of offset allowed for the applicable joint in Table UW-33?

Reply: Yes.

Very truly yours,

Alan Roby

Alan Roby
Secretary; Subcommittee on Pressure Vessels
(212) 591-8538

BCC: **T. P. Pastor**
K. Mokhtarian
R. W. Boyce
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Approval Date: July 17, 1998

See Numeric Index for expiration and any reaffirmation dates.

Case 2286
Alternative Rules for Determining Allowable Compressive Stresses for Cylinders, Cones, Spheres, and Formed Heads
Section VIII, Divisions 1 and 2

Inquiry: May alternative rules for determining allowable compressive stresses for cylinders, cones, spheres, and formed heads be used for the design of these components in lieu of the rules of Section VIII, Division 1, UG-23, UG-28, UG-29, UG-33, and Appendix 1-8, or Section VIII, Division 2, Article D-3?

Reply: It is the opinion of the Committee that cylinders, cones, spheres, and formed heads for pressure vessels otherwise designed and constructed in accordance with the rules of Section VIII, Divisions 1 or 2, may be designed using the following rules for calculation of allowable compressive stresses in lieu of the rules stated in the Inquiry above.

1 GENERAL

1.1 Scope

This Code Case provides alternative rules to those given in Section VIII, Division 1, UG-23(b), UG-28, UG-29, UG-33 and Appendix 1-8, or Section VIII, Division 2, Article D-3 for determining allowable compressive stresses for unstiffened and ring stiffened circular cylinders and cones, and for unstiffened spherical, ellipsoidal, and torispherical heads. The allowable stress equations are based upon theoretical buckling equations that have been reduced by knockdown factors and by plasticity reduction factors that were determined from tests on fabricated shells.

This Code Case expands the coverage of load conditions and shell geometries, and includes equations for combinations of loads not considered in the Code paragraphs referenced above. These alternative rules are applicable to D_o/t ratios not exceeding 2000, compared to the $D_o/t = 1000$ limit in Fig. G in Subpart 3 of Section II, Part D. Use of these alternative rules assumes the shell section to be axisymmetric with uniform thickness for unstiffened cylinders and formed heads. Stiffened cylinders and cones are also assumed to be of uniform thickness between stiffeners. Where nozzles with reinforcing plates or locally thickened shell sections exist, use the thinnest uniform thickness in the applicable unstiffened or stiffened shell section for calculation of allowable compressive stress. When used, this Code Case shall be made applicable to the entire vessel.

The reinforcement for openings in vessels that do not exceed 10% of the cylinder diameter or 80% of the ring spacing into which the opening is placed may be designed in accordance with the following rules. Openings in shells that exceed these limitations require a special design performed in accordance with the rules of Division 1, U-2(g) or Division 2, AD-100 as applicable.

Reinforcement for nozzle openings in vessels designed for external pressure alone shall be in accordance with the requirements of UG-37(d)(1) or AD-520(b), as applicable. The required thickness shall be determined in accordance with para. 3.3 or 4.2. The factor F used in UG-37(c) and AD-520(a) shall be 1.0.

For cases where the shell thickness is controlled by combinations other than external pressure alone, the opening(s) shall be reinforced to provide 100% of that required in UG-37(c) or AD-520(a), as applicable. The required thickness t or t_c (in the applicable unstiffened or stiffened shell section) shall be the thickness used for allowable stress calculation in the appropriate portion of para. 5.0. The factor F used in UG-37(c) and AD-520(a) shall be 1.0.

The maximum temperature permitted for use of this Code Case is shown in Table 1.

When applying this Code Case for Section VIII, Division 1, all references to Section VIII, Division 2 shall be ignored. Likewise, when using this Code Case for Section VIII, Division 2 applications, then all references to Section VIII, Division 1 shall be ignored.

Alternative equations for determination of allowable compressive stress due to loads specified in UG-22 or AD-110 are given for the following:

(a) Stiffened or unstiffened cylinders and cones subjected to:

(1) uniform axial or longitudinal compression;

(2) longitudinal compression from a bending moment applied over the entire cross-section;

TABLE 1

Table in Which Material is Listed		Max. Temperature, °F
Division 1	Division 2	
UCS-23.1	ACS-1	800
UNF-23.1	ANF-1.1	300
UNF-23.2	ANF-1.2	150
UNF-23.3	ANF-1.3	900
UNF-23.4	ANF-1.4	600
UNF-23.5	...	600
UHA-23	AHA-1	800
UHT-23	AQT-1	700

(3) circumferential compression from external pressure or other applied loads;

(4) loads that produce in-plane shear stresses; and

(5) combinations of (a), (b), (c), and (d) above.

(b) Unstiffened spherical shells and formed heads subjected to:

(1) equal biaxial compressive membrane stresses;

(2) unequal biaxial compressive membrane stresses; and

(3) unequal biaxial membrane stresses — one stress is tensile and the other is compressive.

1.2 Materials

The allowable stress equations apply directly to shells fabricated from carbon and low alloy steel materials listed in Table UCS-23 or Table ACS-1 of Section VIII at temperatures below the creep range. These equations can also be applied to other materials for which a chart or table is provided in Subpart 3 of Section II, Part D. The method for calculating the allowable stresses for shells constructed from these materials is determined by the following procedure.

Step 1. Calculate the value of factor A using the following equations. The terms F_{xe}, F_{he} and F_{ve} are defined in the Nomenclature (Section 2).

$$A = \frac{F_{xe}}{E} \qquad A = \frac{F_{he}}{E} \qquad A = \frac{F_{ve}}{E}$$

Step 2. Using the value of A calculated in Step 1, enter the applicable material chart in Subpart 3 of Section II, Part D for the material under consideration. Move vertically to an intersection with the material temperature line for the design temperature. Use interpolation for intermediate temperature values.

Step 3. From the intersection obtained in Step 2, move horizontally to the right to obtain the value of B. E_t is given by the following equation:

$$E_t = \frac{2B}{A}$$

When values of A fall to the left of the applicable material/temperature line in Step 2, $E_t = E$.

Step 4. Calculate the allowable stresses from the following equations:

$$F_{xa} = \frac{F_{xe}}{FS}\frac{E_t}{E} \quad F_{ba} = F_{xa} \quad F_{ha} = \frac{F_{he}}{FS}\frac{E_t}{E} \quad F_{va} = \frac{F_{ve}}{FS}\frac{E_t}{E}$$

1.3 Stress Reduction Factors

Allowable stresses in this Code Case are determined by applying a stress reduction factor, FS, to predicted buckling stresses calculated in this Code Case. The required values of FS are 2.0 when the buckling stress is elastic and $5/3$ (1.67) when the buckling stress equals yield stress at design temperature. A linear variation shall be used between these limits. The equations for FS are given below.

$$\begin{aligned} FS &= 2.0 && \text{if } F_{ic} \le 0.55F_y \\ FS &= 2.407\text{-}0.741F_{ic}/F_y && \text{if } 0.55\,F_y < F_{ic} < F_y \\ FS &= 1.667 && \text{if } F_{ic} \ge F_y \end{aligned}$$

F_{ic} is the predicted buckling stress, which is determined by letting $FS = 1$ in the allowable stress equations. For combinations of earthquake loading or wind loading with other load cases listed in UG-22 or AD-110, the allowable stresses may be increased as permitted by UG-23(c) or Table AD-150.1.

1.4 Capacity Reduction Factors (β)

Capacity reduction factors that account for shape imperfections are built into the allowable stress equations in this Code Case. These factors are in addition to the stress reduction factors in 1.3 above.

(a) For unstiffened or ring stiffened cylinders under axial compression:

$$\beta = 0.207 \qquad \text{for } \frac{D_o}{t} \geq 1247$$

$$\beta = \frac{338}{389 + \frac{D_o}{t}} \qquad \text{for } \frac{D_o}{t} < 1247$$

(b) Unstiffened and ring stiffened cylinders and cones under external pressure: $\beta = 0.8$

(c) Spherical, torispherical, and ellipsoidal heads under external pressure: $\beta = 0.124$

1.5 Stress Components for Stability Analysis and Design

Stress components which control the buckling of a cylindrical shell consist of longitudinal, circumferential, and in-plane shear membrane stresses.

1.6 Geometry

Allowable stress equations are given for the following geometries and load conditions:

(a) Unstiffened cylindrical, conical, and spherical shells;

(b) Ring stiffened cylindrical and conical shells; and

(c) Unstiffened spherical, ellipsoidal, and torispherical heads.

The cylinder and cone geometries are illustrated in Figs. 2.1 and 2.3 and the stiffener geometries in Fig. 2.4. The effective sections for ring stiffeners are shown in Fig. 2.2. The maximum cone angle α shall not exceed 60 deg.

1.7 Buckling Design Method

The buckling strength formulations presented in this Code Case are based upon classical linear theory which is modified by reduction factors that account for the effects of imperfections, boundary conditions, nonlinearity of material properties, and residual stresses. The reduction factors are determined from approximate lower bound values of test data of shells with initial imperfections representative of the tolerance limits specified in this Code Case.

Special consideration shall be given to ends of members (shell sections) or areas of load application where stress distribution may be nonlinear and localized stresses may exceed those predicted by linear theory. When the localized stresses extend over a distance equal to one half the buckling mode (approx. $1.2\sqrt{D_o t}$), the localized stresses should be considered as a uniform stress around the full circumference. Additional stiffening may be required.

1.8

This Case number shall be shown on the Manufacturer's Data Report.

2 NOMENCLATURE

NOTE: The terms not defined here are uniquely defined in the sections in which they are first used. The word "hoop" used in this Code Case is synonymous to the term "circumferential."

A = cross-sectional area of cylinder
$A = \pi(D_o - t)t$, in.^{2}

A_S = cross-sectional area of a ring stiffener, in.^{2}

A_F = cross-sectional area of a large ring stiffener that acts as a bulkhead, in.^{2}

D_i = inside diameter of cylinder, in.

D_o = outside diameter of cylinder, in.

D_e = outside diameter of assumed equivalent cylinder for design of cones or conical sections, in.

D_S = outside diameter at small end of cone, or conical section between lines of support, in.

D_L = outside diameter at large end of cone, or conical section between lines of support, in.

E = modulus of elasticity of material at design temperature, determined from the applicable material chart in Subpart 2 of Section II, Part D, ksi. The applicable material chart is given in Table 1A and 1B, or Table 2A and 2B, Subpart 1, Section II, Part D. Use linear interpolation for intermediate temperatures.

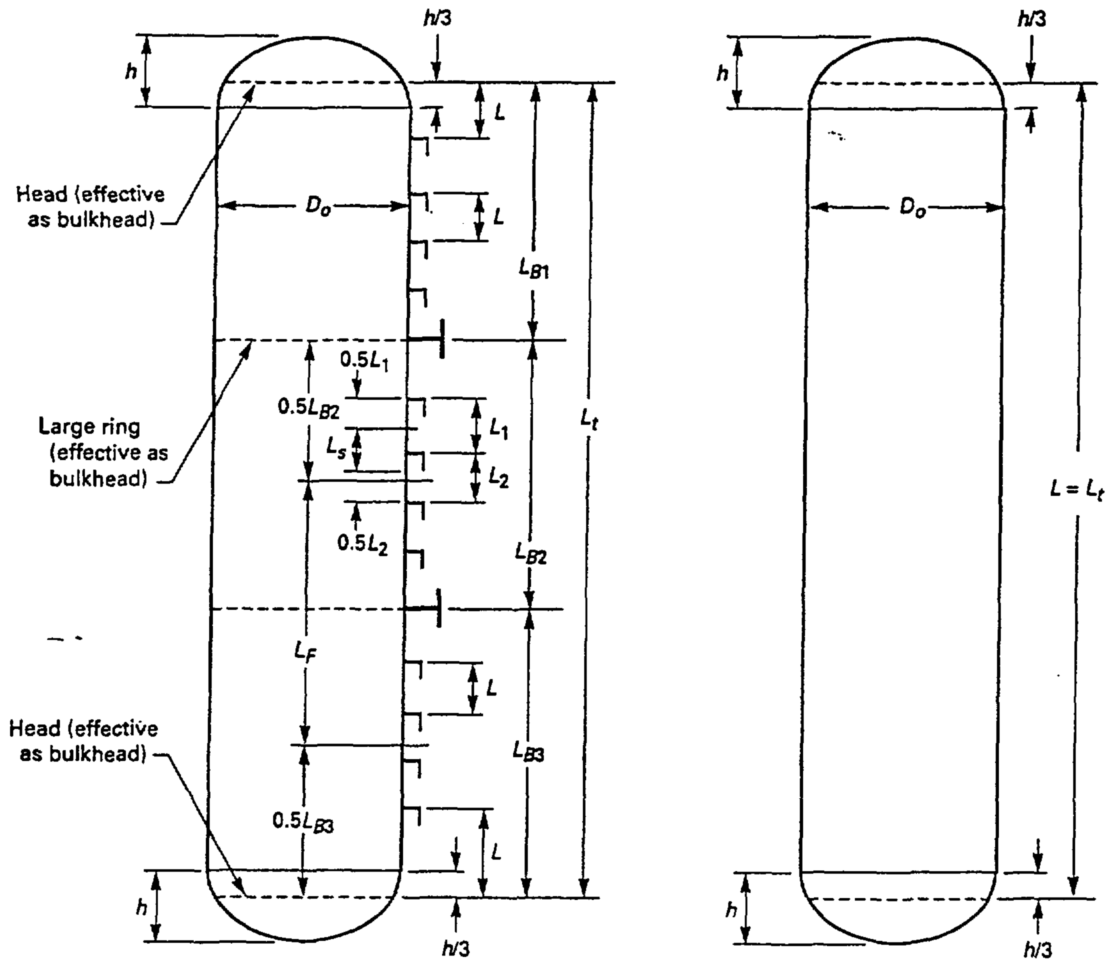
E_t = tangent modulus, ksi

f_a = axial (longitudinal) compressive membrane stress resulting from applied axial load, Q, ksi

f_b = axial (longitudinal) compressive membrane stress resulting from applied bending moment, M, ksi

f_h = hoop compressive membrane stress resulting from applied external pressure, P, ksi

f_q = axial (longitudinal) compressive membrane



(a) Ring Stiffened

(b) Unstiffened

FIG. 2.1 GEOMETRY OF CYLINDERS

stress resulting from pressure load, Q_p, on end of cylinder, ksi

f_v = shear stress from applied loads, ksi

$f_x = f_a + f_q$, ksi

F_{ba} = allowable axial compressive membrane stress of a cylinder due to bending moment, M, in the absence of other loads, ksi

F_{ca} = allowable compressive membrane stress of a cylinder due to axial compression load with $\lambda_c >$ 0.15, ksi

F_{bha} = allowable axial compressive membrane stress of a cylinder due to bending in the presence of hoop compression, ksi

F_{hba} = allowable hoop compressive membrane stress of a cylinder in the presence of longitudinal compression due to a bending moment, ksi

F_{he} = elastic hoop compressive membrane failure stress of a cylinder or formed head under external pressure alone, ksi

F_{ha} = allowable hoop compressive membrane stress of a cylinder or formed head under external pressure alone, ksi

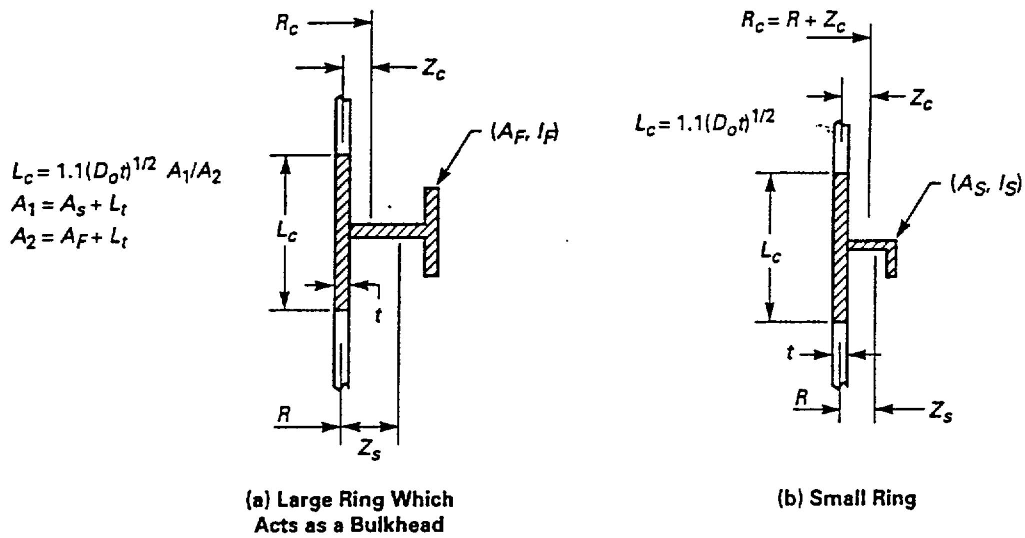


FIG. 2.2 SECTIONS THROUGH RINGS

F_{hva} = allowable hoop compressive membrane stress in the presence of shear stress, ksi

F_{hxa} = allowable hoop compressive membrane stress of a cylinder in the presence of axial compression, ksi

F_{ta} = allowable stress in tension, from applicable table in Subpart 1 of Section II, Part D, ksi

F_{va} = allowable shear stress of a cylinder subjected only to shear stress, ksi

F_{ve} = elastic shear buckling stress of a cylinder subjected only to shear stress, ksi

F_{vha} = allowable shear stress of a cylinder subjected to shear stress in the presence of hoop compression, ksi

F_{xa} = allowable compressive membrane stress of a cylinder due to axial compression load with $\lambda_c \leq 0.15$, ksi

F_{xc} = inelastic axial compressive membrane failure (local buckling) stress of a cylinder in the absence of other loads, ksi

F_{xe} = elastic axial compressive membrane failure (local buckling) stress of a cylinder in the absence of other loads, ksi

F_{xha} = allowable axial compressive membrane stress of a cylinder in the presence of hoop compression, ksi

F_y = yield strength of material at design metal temperature from applicable table in Subpart 1 of Section II, Part D, ksi. For values of F_y not provided in Section II, Part D, use UG-28(c)(2), Steps (3)(a) and (3)(b).

FS = stress reduction factor or design factor

I = moment of inertia of full cross-section, $I = \pi R^3 t$, in.[4]

c = distance from neutral axis of cross-section to point under consideration, in.

I_s' = moment of inertia of ring stiffener plus effective length of shell about centroidal axis of combined section, in.[4]

$$I_s' = I_s + A_s Z_s^2 \frac{L_e t}{A_s + L_e t} + \frac{L_e t^3}{12}$$

I_s = moment of inertia of ring stiffener about its centroidal axis, in.[4]

K = effective length factor for column buckling

$L, L_1, L_2, L\ldots$ = design length of unstiffened vessel section between lines of support or the total length of tube between tube sheets, in. A line of support is:

(a) a circumferential line on a head (excluding conical heads) at one-third the depth of the head from the head tangent line as shown on Fig. 2.1;

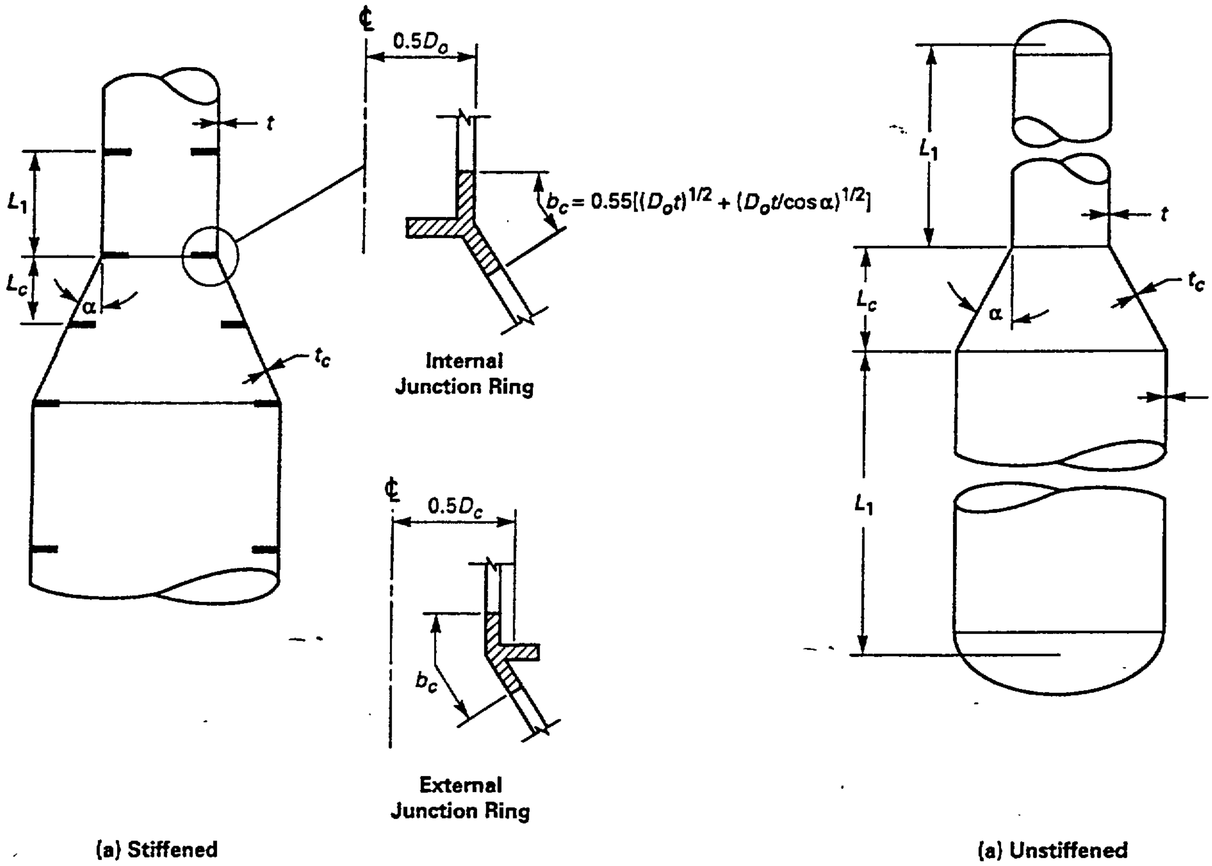


FIG. 2.3 GEOMETRY OF CONICAL SECTIONS

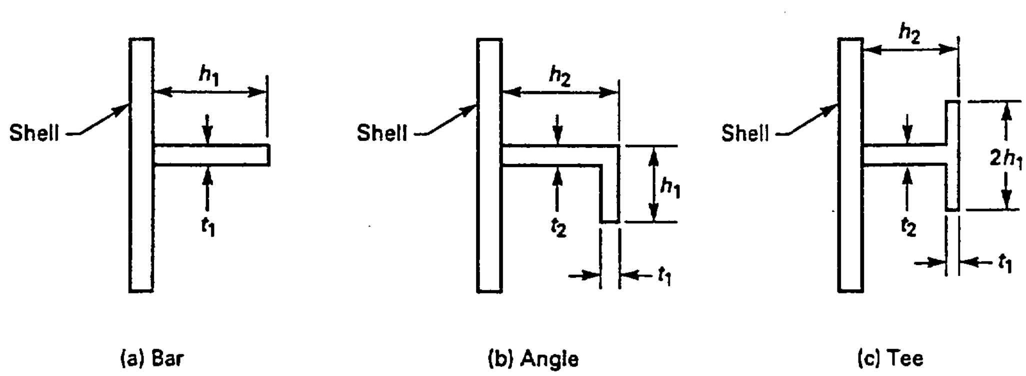


FIG. 2.4 STIFFENER GEOMETRY FOR EQS. 3-17 AND 3-18

(b) a stiffening ring that meets the requirements of Eq. (3-14) of para. 3.5.2;
(c) a tubesheet.

$L_B, L_{B1}, L_{B2}, L_{B...}$ = length of cylinder between bulkheads or large rings designed to act as bulkheads, in.

L_c = axial length of cone or conical section, in. (see Fig. 2.3)

L_u = laterally unbraced (laterally unsupported) length of a cylindrical member that is subject to column buckling, in. This applies to supports for pressure vessels or pedestal type vessels. Stiffening rings are not points of support unless they are externally supported.

L_e = effective length of shell, in. (see Fig. 2.2)

L_F = one-half of the sum of the distances, L_B, from the center line of a large ring to the next large ring or head line of support on either side of the large ring, in. (see Fig. 2.1)

L_s = one-half of the sum of the distances from the center line of a stiffening ring to the next line of support on either side of the ring, measured parallel to the axis of the cylinder, in. A line of support is described in the definition for L (see Fig. 2.1), in.

L_t = overall length of vessel as shown in Fig. 2.1, in.

M = applied bending moment across the vessel cross-section, in.-kips

$M_s = L_s/\sqrt{R_o t}$

$M_x = L/\sqrt{R_o t}$

P = applied external pressure, ksi

P_a = allowable external pressure in the absence of other loads, ksi

Q = applied axial compression load, kips

Q_p = axial compression load on end of cylinder resulting from applied external pressure, kips

r = radius of gyration of cylinder, in.

$$r = \frac{\left(D_o^2 + D_i^2\right)^{1/2}}{4}$$

R = radius to centerline of shell, in.

R_c = radius to centroid of combined ring stiffener and effective length of shell, in.

$$R_c = R + Z_c$$

R_o = radius to outside of shell, in.

S = elastic section modulus of full shell cross-section, in.^{3}

$$S = \frac{\pi(D_o^4 - D_i^4)}{32D_o}$$

V = shear force from applied loads at cross-section under consideration, kips

ϕ = angle measured around the circumference from the direction of applied shear force to the point under consideration.

t = thickness of shell, less corrosion allowance, in.

t_c = thickness of cone, less corrosion allowance, in.

Z_c = radial distance from centerline of shell to centroid of combined section of ring and effective length of shell, in.

$$Z_c = \frac{A_s Z_s}{A_s + L_e t}$$

Z_s = radial distance from center line of shell to centroid of ring stiffener, (positive for outside rings), in.

α = one-half of the apex angle of a conical section

β = capacity reduction factor to account for shape imperfections

λ_c = slenderness factor for column buckling

$$\lambda_c = \frac{KL_u}{\pi r}\sqrt{\frac{F_{xa}FS}{E}}$$

3 ALLOWABLE COMPRESSIVE STRESSES FOR CYLINDRICAL SHELLS

The maximum allowable stresses for cylindrical shells subjected to loads that produce compressive stresses are given by the following equations. For stress components acting alone, the maximum values shall be used. For combined stress components, the concurrent (coexisting) stress values shall be used.

In no case shall the allowable primary membrane compressive stresses exceed the maximum allowable tensile stress listed in Section II, Part D.

3.1 Uniform Axial Compression

Allowable longitudinal stress for a cylindrical shell under uniform axial compression is given by F_{xa} for values of $\lambda_c \leq 0.15$ and by F_{ca} for values of $\lambda_c > 0.15$.

$$\lambda_c = \frac{KL_u}{\pi r}\sqrt{\frac{F_{xa}\,FS}{E}}$$

where KL_u is the effective length. L_u is the unbraced length. Minimum values for K are:

(a) 2.1 for members with one end free and the other end fixed;

(b) 1.0 for members with both ends pinned;

(c) 0.8 for members with one end pinned and the other end fixed; and

(d) 0.65 for members with both ends fixed.

In this case, "member" is the unbraced cylindrical shell or cylindrical shell section as defined in the Nomenclature.

3.1.1 Local Buckling (for $\lambda_c \leq 0.15$). F_{xa} is the smaller of the values given by Eqs. (3-1) and (3-2).

$$F_{xa} = \frac{F_y}{FS} \quad \text{for } \frac{D_o}{t} \leq 135 \qquad (3\text{-}1a)$$

$$F_{xa} = \frac{466F_y}{\left(331 + \frac{D_o}{t}\right)FS} \quad \text{for } 135 < \frac{D_o}{t} < 600 \qquad (3\text{-}1b)$$

$$F_{xa} = \frac{0.5F_y}{FS} \quad \text{for } \frac{D_o}{t} \geq 600 \qquad (3\text{-}1c)$$

or

$$F_{xa} = \frac{F_{xe}}{FS} \qquad (3\text{-}2)$$

where

$$F_{xe} = \frac{C_x E t}{D_o} \qquad (3\text{-}3)$$

$$C_x = \frac{409\bar{c}}{389 + \frac{D_o}{t}} \text{ not to exceed 0.9} \quad \text{for } \frac{D_o}{t} < 1247$$

$$C_x = 0.25\bar{c} \quad \text{for } \frac{D_o}{t} \geq 1247$$

$$\bar{c} = 2.64 \quad \text{for } M_x \leq 1.5$$

$$\bar{c} = \frac{3.13}{M_x^{0.42}} \quad \text{for } 1.5 < M_x < 15$$

$$\bar{c} = 1.0 \quad \text{for } M_x \geq 15$$

$$M_x = \frac{L}{(R_o t)^{1/2}} \qquad (3\text{-}4)$$

3.1.2 Column Buckling ($\lambda_c > 0.15$ and $KL_u/r < 200$)

$$F_{ca} = F_{xa}[1 - 0.74(\lambda_c - 0.15)]^{0.3} \qquad (3\text{-}5a)$$

$$\text{for } 0.15 < \lambda_c < 1.147$$

$$F_{ca} = \frac{0.88\, F_{xa}}{\lambda_c^2} \text{ for } \lambda_c \geq 1.147 \qquad (3\text{-}5b)$$

3.2 Axial Compression Due to Bending Moment

Allowable longitudinal stress for a cylinder subjected to a bending moment acting across the full circular cross-section is given by F_{ba}.

$$F_{ba} = F_{xa} \text{ (see para. 3.1.1)} \quad \text{for } \frac{D_o}{t} \geq 135 \qquad (3\text{-}6a)$$

$$F_{ba} = \frac{466F_y}{FS\left(331 + \frac{D_o}{t}\right)} \quad \text{for } 100 \leq \frac{D_o}{t} < 135 \qquad (3\text{-}6b)$$

$$F_{ba} = \frac{1.081F_y}{FS} \quad \text{for } \frac{D_o}{t} < 100 \text{ and } \gamma \geq 0.11 \qquad (3\text{-}6c)$$

$$F_{ba} = \frac{(1.4 - 2.9\gamma)F_y}{FS} \quad \text{for } \frac{D_o}{t} < 100 \text{ and } \gamma < 0.11 \qquad (3\text{-}6d)$$

where

$$\gamma = \frac{F_y D_o}{E\, t}$$

3.3 External Pressure

The allowable circumferential compressive stress for a cylinder under external pressure is given by F_{ha} and the allowable external pressure is given by the following equation.

$$P_a = 2F_{ha}\frac{t}{D_o}$$

$$F_{ha} = \frac{F_y}{FS} \quad \text{for } \frac{F_{he}}{F_y} \geq 2.439 \qquad (3\text{-}7a)$$

$$F_{ha} = \frac{0.7F_y}{FS}\left(\frac{F_{he}}{F_y}\right)^{0.4} \quad \text{for } 0.552 < \frac{F_{he}}{F_y} < 2.439 \qquad (3\text{-}7b)$$

$$F_{ha} = \frac{F_{he}}{FS} \quad \text{for } \frac{F_{he}}{F_y} \leq 0.552 \qquad (3\text{-}7c)$$

where

$$F_{he} = 1.6C_hE\frac{t}{D_o} \tag{3-8}$$

$$C_h = 0.55\frac{t}{D_o} \quad \text{for } M_x \geq 2\left(\frac{D_o}{t}\right)^{0.94}$$

$$C_h = 1.12M_x^{-1.058} \quad \text{for } 13 < M_x < 2\left(\frac{D_o}{t}\right)^{0.94}$$

$$C_h = \frac{0.92}{M_x - 0.579} \quad \text{for } 1.5 < M_x \leq 13$$

$$C_h = 1.0 \quad \text{for } M_x \leq 1.5$$

3.4 Shear

Allowable in-plane shear stress for a cylindrical shell is given by F_{va}.

$$F_{va} = \frac{\eta_v F_{ve}}{FS} \tag{3-9}$$

where

$$F_{ve} = \alpha_v C_v E\frac{t}{D_o} \tag{3-10}$$

$$C_v = 4.454 \quad \text{for } M_x \leq 1.5 \tag{3-11a}$$

$$C_v = \left(\frac{9.64}{M_x^2}\right)(1 + 0.0239\,M_x^3)^{1/2} \text{ for } 1.5 < M_x < 26 \tag{3-11b}$$

$$C_v = \frac{1.492}{M_x^{1/2}} \quad \text{for } 26 \leq M_x < 4.347\frac{D_o}{t} \tag{3-11c}$$

$$C_v = 0.716\left(\frac{t}{D_o}\right)^{1/2} \quad \text{for } M_x \geq 4.347\frac{D_o}{t} \tag{3-11d}$$

$$\alpha_v = 0.8 \quad \text{for } \frac{D_o}{t} \leq 500$$

$$\alpha_v = 1.389 - 0.218\log_{10}\left(\frac{D_o}{t}\right) \quad \text{for } \frac{D_o}{t} > 500$$

$$\eta_v = 1.0 \quad \text{for } \frac{F_{ve}}{F_y} \leq 0.48$$

$$\eta_v = 0.43\frac{F_y}{F_{ve}} + 0.1 \quad \text{for } 0.48 < \frac{F_{ve}}{F_y} < 1.7$$

$$\eta_v = 0.6\frac{F_y}{F_{ve}} \quad \text{for } \frac{F_{ve}}{F_y} \geq 1.7$$

3.5 Sizing of Rings (General Instability)

3.5.1 Uniform Axial Compression and Axial Compression Due to Bending. When ring stiffeners are used to increase the allowable longitudinal compressive stress, the following equations must be satisfied. For a stiffener to be considered, M_x shall be less than 15.

$$A_s \geq \left[\frac{0.334}{M_s^{0.6}} - 0.063\right]L_st \quad \text{and} \quad A_s \geq 0.06L_st \tag{3-12}$$

$$\text{also } I_s' \geq \frac{5.33L_st^3}{M_s^{1.8}} \tag{3-13}$$

3.5.2 External Pressure

(a) Small rings

$$I_s' \geq \frac{1.5F_{he}L_sR_c^2t}{E(n^2 - 1)} \tag{3-14}$$

F_{he} = stress determined from Eq. (3-8) with $M_x = M_s$.

$n^2 = \dfrac{2D_o^{3/2}}{3L_Bt^{1/2}}$ use $n = 2$ for $_n{}^2 < 4$,
and $n = 10$ for $_n{}^2 > 100$

(b) Large rings which act as bulkheads

$$I_s' \geq I_F \tag{3-15}$$

where

$$I_F = \frac{F_{heF}L_FR_c^2t}{2E}$$

I_F = the value of I_s' which makes a large stiffener act as a bulkhead. The effective length of shell is $L_e = 1.1\sqrt{D_ot}(A_1/A_2)$

A_1 = cross-sectional area of small ring plus shell area equal to L_st, in.^{2}

A_2 = cross-sectional area of large ring plus shell area equal to L_st, in.^{2}

R_c = radius to centroid of combined large ring and effective width of shell, in.

F_{heF} = average value of the hoop buckling stresses, F_{he}, over length L_F where F_{he} is determined from Eq. (3-8), ksi

3.5.3 Shear

$$I_s' \geq 0.184\, C_v M_s^{0.8} t^3 L_s \qquad (3\text{-}16)$$

C_v = value determined from Eq. (3-11) with $M_x = M_s$.

3.5.4 Local Stiffener Geometry Requirements. Stiffener geometry requirements are as follows. See Fig. 2.4 for stiffener geometry and definition of terms.

(a) Flat bar stiffener, flange of a tee stiffener, and outstanding leg of an angle stiffener

$$\frac{h_1}{t_1} \leq 0.375\left(\frac{E}{F_y}\right)^{1/2} \qquad (3\text{-}17)$$

where h_1 is the full width of a flat bar stiffener or outstanding leg of an angle stiffener and one-half of the full width of the flange of a tee stiffener and t_1 is the thickness of the bar, leg of angle, or flange of tee.

(b) Web of tee stiffener or leg of angle stiffener attached to shell

$$\frac{h_2}{t_2} \leq 1.0\left(\frac{E}{F_y}\right)^{1/2} \qquad (3\text{-}18)$$

where h_2 is the full depth of a tee section or full width of an angle leg and t_2 is the thickness of the web or angle leg.

3.6 Tolerances for Cylindrical and Conical Shells

3.6.1 Shells Subjected to Uniform Axial Compression and Axial Compression Due to Bending Moment. Cylindrical and conical shells shall meet the out-of-roundness limitations specified in UG-80(a) or AF-130.1. Additionally, the local deviation from a straight line, e, measured along a meridian over a gauge length L_x shall not exceed the maximum permissible deviation e_x given below.

$e_x = 0.002R$

$L_x = 4\sqrt{Rt}$ but not greater than L for cylinders

$L_x = 4\sqrt{Rt/\cos\alpha}$ but not greater than $L_c/\cos\alpha$ for cones

$L_x = 25t$ across circumferential welds

Also L_x is not greater than 95% of the meridional distance between circumferential welds.

3.6.2 Shells Subjected to External Pressure. Cylindrical and conical shells shall meet the tolerances as specified herein. These tolerance requirements replace some portions of those specified in UG-80(b) or AF-130.2. All requirements of UG-80(a) or AF-130.1 are applicable. In place of the maximum deviation requirements specified in UG-80(b)(2) or AF-130.2(a), the following requirements apply.

The maximum plus or minus deviation from a true circular form, e, shall not exceed the value given by the following equations.

$$e = 0.0165t\,(M_x + 3.25)^{1.069} \qquad (3\text{-}19)$$

e need not be less than $0.2t$, and shall not exceed the smaller of 0.0242R or $2t$.

M_x and L are defined in the Nomenclature

Measurements to determine e shall be made from a segmental circular template having the design outside radius, and placed on outside of the shell. The chord length L_c is given by the following equation.

$$L_c = 2R\sin(\pi/2n) \qquad (3\text{-}20)$$

$$n = c\left(\frac{\sqrt{R/t}}{L/R}\right)^d \qquad 2 \leq n \leq 1.41(R/t)^{0.5} \qquad (3\text{-}21)$$

where

$$c = 2.28(R/t)^{0.54} \leq 2.80$$

$$d = 0.38(R/t)^{0.044} \leq 0.485$$

The requirements of UG-80(b)(3),(4),(6),(7),(8), and UG-80(b)(10) or AF-130.2(b) and AF-130.4 remain applicable.

3.6.3 Shells Subjected to Shear. Cylindrical and conical shells shall meet the tolerances specified in UG-80(a) or AF-130.1.

4 ALLOWABLE COMPRESSIVE STRESSES FOR CONES

Unstiffened conical transitions or cone sections between rings of stiffened cones with an angle $\alpha \leq 60°$ shall be designed for local buckling as an equivalent cylinder according to the following procedure. See Fig. 2.3 for cone geometry.

4.1 Uniform Axial Compression and Bending

4.1.1 Allowable Longitudinal and Bending Stresses. Assume an equivalent cylinder with diameter D_e equal to $D/\cos\alpha$, where D is the outside diameter at the cross-section under consideration. D_e is substituted for D_o in the equations given in paras. 3.1 and 3.2 to find F_{xa} and F_{ba} and L_c for L in Eq. (3-4), where L_c is the unbraced length of cone section. The allowable stress must be satisfied at all cross-sections along the length of the cone.

4.1.2 Unstiffened Cone-Cylinder Junctions. Cone-cylinder junctions are subject to unbalanced radial forces (due to axial load and bending moment) and to localized bending stresses caused by the angle change. The longitudinal and hoop stresses at the junction may be evaluated as follows:

(a) Longitudinal Stress. In lieu of detailed analysis, the localized bending stress at an unstiffened cone-cylinder junction may be estimated by the following equation.

$$f_b' = \frac{0.6t\sqrt{D(t+t_c)}}{t_e^2}(f_x + f_b)\tan\alpha \qquad (4\text{-}1)$$

where

D = outside diameter of cylinder at junction to cone
t = thickness of cylinder
t_c = thickness of cone
$t_e = t$ to find stress in cylinder section
$t_e = t_c$ to find stress in cone section
α = cone angle as defined in Fig. 2.3
f_x = uniform longitudinal stress in cylinder section at the cone-cylinder junction resulting from pressure and/or applied axial loads, see Nomenclature, Section 2.
f_b = longitudinal stress in cylinder section at the cone-cylinder junction resulting from bending moment

For strength requirements, the total stress ($f_x + f_b + f_b'$) in the cone and cylinder sections shall be limited to 3 times the allowable stress at temperature listed in Section II, Part D, Tables 1A and 1B (Division 1) or Tables 2A and 2B (Division 2). The combined stress ($f_x + f_b$) shall not exceed the allowable stress at temperature listed in Section II, Part D, Tables 1A and 1B (Division 1), or Tables 2A and 2B (Division 2).

(b) Hoop Stress. The hoop stress caused by the unbalanced radial line load may be estimated from:

$$f_h' = 0.45\sqrt{D/t}\,(f_x + f_b)\tan\alpha \qquad (4\text{-}2)$$

For hoop tension, f_h' shall be limited to 1.5 times the tensile allowable per (a) above. For Division 1 applications, the applicable joint efficiency factor (E) shall be included in the equation. For hoop compression, f_h' shall be limited to F_{ha} where F_{ha} is computed from Eq. (3-7) with

$$F_{he} = 0.4\,E(t/D)$$

A cone-cylinder junction that does not satisfy the above criteria may be strengthened either by increasing the cylinder and cone wall thickness at the junction, or by providing a stiffening ring at the junction.

4.1.3 Cone-Cylinder Junction Rings. If stiffening rings are required, the section properties shall satisfy the following requirements:

$$A_c \geq \frac{tD}{F_y}(f_x + f_b)\tan\alpha \qquad (4\text{-}3)$$

$$I_c \geq \frac{tD(D_c)^2}{8E}(f_x + f_b)\tan\alpha \qquad (4\text{-}4)$$

where

D = cylinder outside diameter at junction = D_L or D_s in Fig. UG-33.1 or Fig. AD-300.1
D_c = diameter to centroid of composite ring section for external rings
$D_c = D_i$ for internal rings
A_c = cross-sectional area of composite ring section
I_c = moment of inertia of composite ring section

In computing A_c and I_c, the effective length of shell wall acting as a flange for the composite ring section shall be computed from:

$$b_e = 0.55(\sqrt{Dt} + \sqrt{Dt_c/\cos\alpha}) \qquad (4\text{-}5)$$

4.2 External Pressure

4.2.1 Allowable Circumferential Compression Stresses. Assume an equivalent cylinder with diameter D_e equal to $0.5(D_L + D_S)/\cos\alpha$, $L_e = L_c/\cos\alpha$. The value D_e is substituted for D_o in the equations given in para. 3.3 to determine F_{ha}. The allowable stress must be satisfied at all cross-sections along the length of the cone.

4.2.2 Intermediate Stiffening Rings. If required, circumferential stiffening rings within cone transitions shall be sized using Eq. (3-14) with $R_c = D_L/2$, where D_L is the cone diameter at the ring, per Fig. UG-33.1 or Fig. AD-300.1, and t is the cone thickness.

4.2.3 Cone-Cylinder Junction Rings. A junction ring is not required for buckling due to external pressure if $f_h < F_{ha}$ where F_{ha} is determined from Eq. (3-7) with F_{he} computed using C_h equal to 0.55 (cos α) (t/D_o) in Eq. (3-8). D_o is the cylinder diameter at the junction. The hoop stress may be calculated from the following equation.

$$f_h = \frac{P\,D_o}{2\,t_c \cos\alpha}$$

Circumferential stiffening rings required at the cone-cylinder junctions shall be sized such that the moment of inertia of the composite ring section satisfies the following equation:

$$I_c > \frac{D^2}{16E}\left\{tL_1F_{he} + \frac{t_cL_cF_{hec}}{\cos^2\alpha}\right\} \qquad (4\text{-}6)$$

where
- D = cylinder outside diameter at junction
- L_c = distance to first stiffening ring in cone section along cone axis as shown in Fig. 2.3
- L_1 = distance to first stiffening ring in cylinder section or line of support
- F_{he} = elastic hoop buckling stress for cylinder [see Eq. (3-8)]
- F_{hec} = F_{he} for cone section treated as an equivalent cylinder
- t = cylinder thickness
- t_c = cone thickness

4.3 Shear

4.3.1 Allowable In-Plane Shear Stress. Assume an equivalent cylinder with a length equal to the short length between rings (L_c/cos α) and a diameter D_e is equal to D/cos α, where D is the outside diameter of the cone at the cross-section under consideration. $L_e = L$ or L_c also as defined in Fig. UG-33.1 or Fig. AD-300.1. This length and diameter shall be substituted into the equations given in para. 3.4 to determine F_{va}.

4.3.2 Intermediate Stiffening Rings. If required, circumferential stiffening rings within cone transition shall be sized using Eq. (3-16) where L_s is the average distance to adjacent rings along the cone axis.

4.4 Local Stiffener Geometry Requirements

To preclude local buckling of a stiffener, the requirements of para. 3.5.4 must be met.

4.5 Tolerances

The tolerances specified in para. 3.6 shall be met.

5 ALLOWABLE STRESS EQUATIONS FOR UNSTIFFENED AND RING-STIFFENED CYLINDERS AND CONES UNDER COMBINED LOADS

5.1 For Combination of Uniform Axial Compression and Hoop Compression

5.1.1 For $\lambda_c \le 0.15$ the allowable stress in the longitudinal direction is given by F_{xha} and the allowable stress in the circumferential direction is given by F_{hxa}.

$$F_{xha} = \left(\frac{1}{F_{xa}^2} - \frac{C_1}{C_2F_{xa}F_{ha}} + \frac{1}{C_2^2F_{ha}^2}\right)^{-0.5} \qquad (5\text{-}1)$$

where

$$C_1 = \frac{(F_{xa}\,FS + F_{ha}\,FS)}{F_y} - 1.0 \text{ and } C_2 = \frac{f_x}{f_h}$$

$$f_x = f_a + f_q = \frac{Q}{A} + \frac{Q_p}{A} \quad \text{and} \quad f_h = \frac{PD_o}{2t}$$

$F_{xa}\,FS$ is given by the smaller of Eqs. (3-1) or (3-2), and F_{ha} is given by Eq. (3-7).

$$F_{hxa} = \frac{F_{xha}}{C_2} \qquad (5\text{-}2)$$

5.1.2 For $0.15 < \lambda_c < 1.2$, F_{xha} shall be taken as the smaller of F_{ah1} and F_{ah2}, where $F_{ah1} = F_{xha}$ as given by Eq. (5-1) with $f_x = f_a$, F_{ah2} is given by the following equation.

$$F_{ah2} = F_{ca}\left(1 - \frac{f_q}{F_y}\right) \qquad (5\text{-}3)$$

F_{ca} is given by Equation (3-5).

5.2 For Combination of Axial Compression Due to Bending Moment, M, and Hoop Compression

The allowable stress in the longitudinal direction is given by F_{bha} and the allowable stress in the circumferential direction is given by F_{hba}.

$$F_{bha} = C_3 C_4 F_{ba} \tag{5-4}$$

where C_3 and C_4 are given by the following equations and F_{ba} is given by Eq. (3-6).

$$C_4 = \frac{f_b}{f_h}\frac{F_{ha}}{F_{ba}}$$

$$C_3^2\,(C_4^2 + 0.6C_4) + C_3^{2n} - 1 = 0 \tag{5-5}$$

$$f_b = \frac{Mc}{I} \qquad f_h = \frac{PD_o}{2t} \qquad n = 5 - 4\frac{F_{ha}\cdot FS}{F_y}$$

Solve for C_3 from Eq. (5-5) by iteration. F_{ha} is given by Eq. (3-7).

$$F_{hba} = F_{bha}\frac{f_h}{f_b} \tag{5-6}$$

5.3 For Combination of Hoop Compression and Shear

The allowable shear stress is given by F_{vha} and the allowable circumferential stress is given by F_{hva}.

$$F_{vha} = \left[\left(\frac{F_{va}^2}{2C_5F_{ha}}\right)^2 + F_{va}^2\right]^{1/2} - \frac{F_{va}^2}{2C_5F_{ha}} \tag{5-7}$$

where

$$C_5 = \frac{f_v}{f_h} \qquad \text{and } f_v = V\sin\phi/\pi Rt$$

F_{va} is given by Eq. (3-9) and F_{ha} is given by Eq. (3-7).

$$F_{hva} = \frac{F_{vha}}{C_5} \tag{5-8}$$

5.4 For Combination of Uniform Axial Compression, Axial Compression Due to Bending Moment, M, and Shear, in the Presence of Hoop Compression ($f_h \neq 0$)

$$\text{Let } K_s = 1 - \left(\frac{f_v}{F_{va}}\right)^2 \tag{5-9}$$

5.4.1 For $\lambda_c \le 0.15$

$$\left(\frac{f_a}{K_sF_{xha}}\right)^{1.7} + \frac{f_b}{K_sF_{bha}} \le 1.0 \tag{5-10}$$

F_{xha} is given by Eq. (5-1), F_{bha} is given by Eq. (5-4) and F_{va} is given by Eq. (3-9).

5.4.2 For $0.15 < \lambda_c < 1.2$

$$\frac{f_a}{K_sF_{xha}} + \frac{8}{9}\frac{\Delta f_b}{K_sF_{bha}} \le 1.0 \quad \text{for } \frac{f_a}{K_sF_{xha}} \ge 0.2 \tag{5-11}$$

$$\frac{f_a}{2K_sF_{xha}} + \frac{\Delta f_b}{K_sF_{bha}} \le 1.0 \quad \text{for } \frac{f_a}{K_sF_{xha}} < 0.2 \tag{5-12}$$

where

$$\Delta = \frac{C_m}{1 - \dfrac{f_a FS}{F_e}} \qquad F_e = \frac{\pi^2 E}{(KL_u/r)^2}$$

See para. 5.1.2 for F_{xha}. F_{bha} is given by Eq. (5-4). K is the effective length factor (see para. 3.1). FS is determined from equations in para. 1.3, where $F_{ic} = F_{xa}\ FS$ [See Eqs. (3-1) and (3-2)].

C_m = coefficient whose value shall be taken as follows:

(a) For compression members in frames subject to joint translation (side sway),

$$C_m = 0.85$$

(b) For rotationally restrained compression members in frames braced against joint translation and not subject to transverse loading between their supports in the plane of bending,

$$C_m = 0.6 - 0.4(M_1/M_2)$$

where M_1/M_2 is the ratio of the smaller to larger moments at the ends of that portion of the member unbraced in the plane of bending under consideration. M_1/M_2 is positive when the member is bent in reverse curvature and negative when bent in single curvature.

(c) For compression members in frames braced against joint translation and subjected to transverse loading between their supports:

(1) for members whose ends are restrained against rotation in the plane of bending,

$$C_m = 0.85$$

(2) for members whose ends are unrestrained against rotation in the plane of bending, for example, an unbraced skirt supported vessel,

$$C_m = 1.0$$

5.5 For Combination of Uniform Axial Compression, Axial Compression Due to Bending Moment, M, and Shear, in the Absence of Hoop Compression ($f_h = 0$)

5.5.1

For $\lambda_c \leq 0.15$

$$\left(\frac{f_a}{K_s F_{xa}}\right)^{1.7} + \frac{f_b}{K_s F_{ba}} \leq 1.0 \qquad (5\text{-}13)$$

F_{xa} is given by the smaller of Eq. (3-1) or (3-2), F_{ba} is given by Eq. (3-6) and K_s is given by Eq. (5-9).

5.5.2

For $0.15 < \lambda_c < 1.2$

$$\frac{f_a}{K_s F_{ca}} + \frac{8}{9}\frac{\Delta f_b}{K_s F_{ba}} \leq 1.0 \quad \text{for } \frac{f_a}{K_s F_{ca}} \geq 0.2 \qquad (5\text{-}14)$$

$$\frac{f_a}{2K_s F_{ca}} + \frac{\Delta f_b}{K_s F_{ba}} \leq 1.0 \quad \text{for } \frac{f_a}{K_s F_{ca}} < 0.2 \qquad (5\text{-}15)$$

F_{ca} is given by Eq. (3-5), F_{ba} is given by Eq. (5-4), and K_s is given by Eq. (5-9). See para. 5.4.2 for definition of Δ.

6. ALLOWABLE COMPRESSIVE STRESSES FOR SPHERICAL SHELLS AND FORMED HEADS, WITH PRESSURE ON CONVEX SIDE

6.1 Spherical Shells

6.1.1 With Equal Biaxial Stresses. The allowable compressive stress for a spherical shell under uniform external pressure is given by F_{ha} and the allowable external pressure is given by P_a.

$$F_{ha} = \frac{F_y}{FS} \qquad \text{for } \frac{F_{he}}{F_y} \geq 6.25 \qquad (6\text{-}1a)$$

$$F_{ha} = \frac{1.31\, F_y}{FS\left(1.15 + \frac{F_{he}}{F_y}\right)} \qquad \text{for } 1.6 < \frac{F_{he}}{F_y} < 6.25 \qquad (6\text{-}1b)$$

$$F_{ha} = \frac{0.18F_{he} + 0.45F_y}{FS} \qquad \text{for } 0.55 < \frac{F_{he}}{F_y} \leq 1.6 \qquad (6\text{-}1c)$$

$$F_{ha} = \frac{F_{he}}{FS} \qquad \text{for } \frac{F_{he}}{F_y} \leq 0.55 \qquad (6\text{-}1d)$$

$$F_{he} = 0.075E\frac{t}{R_o} \qquad (6\text{-}2)$$

$$P_a = 2F_{ha}\frac{t}{R_o}. \qquad (6\text{-}3)$$

where R_o is the radius to the outside of the spherical shell and F_{ha} is given by Eq. (6-1).

6.1.2 With Unequal Biaxial Stresses — Both Stresses are Compressive. The allowable compressive stresses for a spherical shell subjected to unequal biaxial stresses, σ_1 and σ_2, where both σ_1 and σ_2 are compression stresses resulting from applied loads, are given by the following equations.

$$F_{1a} = \frac{0.6}{1 - 0.4k}F_{ha} \qquad (6\text{-}4)$$

$$F_{2a} = kF_{1a} \qquad (6\text{-}5)$$

where $k = \sigma_2/\sigma_1$ and F_{ha} is given by Eq. (6-1). F_{1a} is the allowable stress in the direction of σ_1 and F_{2a} is the allowable stress in the direction of σ_2. σ_1 is the larger of the compression stresses.

6.1.3 With Unequal Biaxial Stresses — One Stress is Compressive and the Other is Tensile. The allowable compressive stress for a spherical shell subjected to unequal biaxial stresses σ_1 and σ_2, where σ_1 is a compression stress and σ_2 is a tensile stress, is given by F_{1a}.

F_{1a} is the value of F_{ha} determined from Eq. (6-1) with F_{he} given by Eq. (6-6).

TABLE 2
FACTOR K_o

Use Interpolation for Intermediate Values						
$D_o/2h_o$	...	3.0	2.8	2.6	2.4	2.2
K_o	...	1.36	1.27	1.18	1.08	0.99
$D_o/2h_o$	2.0	1.8	1.6	1.4	1.2	1.0
K_o	0.90	0.81	0.73	0.65	0.57	0.50

$$F_{he} = (C_o + C_p)E\frac{t}{R_o} \qquad (6\text{-}6)$$

$$C_o = \frac{102.2}{195 + R_o/t} \qquad \text{for } \frac{R_o}{t} < 622$$

$$C_o = 0.125 \qquad \text{for } \frac{R_o}{t} \geq 622$$

$$C_p = \frac{1.06}{3.24 + \frac{1}{\bar{p}}} \qquad \bar{p} = \frac{\sigma_2}{E}\frac{R_o}{t}$$

6.1.4 Shear. When shear is present, the principal stresses shall be calculated and used for σ_1 and σ_2.

6.2 Toroidal and Ellipsoidal Heads

The allowable compressive stresses for formed heads is determined by the equations given for spherical shells where R_o is defined below.

R_o = for torispherical heads, the outside radius of the crown portion of the head, in.

R_o = for ellipsoidal heads, the equivalent outside spherical radius taken as K_oD_o, in.

K_o = factor depending on the ellipsoidal head proportions $D_o/2h_o$ (see Table 2 above)

h_o = outside height of the ellipsoidal head measured from the tangent line (head-bend line), in.

6.3 Tolerances for Formed Heads

Formed heads shall meet the tolerances specified in UG-81 or AF-135. Additionally, the maximum local deviation from true circular form, e, for spherical shells and any spherical portion of a formed head designed for external pressure shall not exceed the shell thickness. Measurements to determine e shall be made with a gage or template with the chord length L_e given by the following equation:

$$L_e = 3.72\sqrt{Rt}$$

7 REFERENCES

API 2U (1987), API Bulletin 2U (BUL 2U), "Bulletin on Stability Design of Cylindrical Shells," prepared under the jurisdiction of the API Committee on Standardization of Offshore Structures, First Edition, May 1987.

Welding Research Council Bulletin 406, "Proposed Rules for Determining Allowable Compressive Stresses for Cylinders, Cones, Spheres, and Formed Heads," C.D. Miller and K. Mokhtarian, November 1995, ISSN 0043-2326.

Miller, C.D. and Mokhtarian, K., (1996), "A Comparison of Proposed Alternative Rules with ASME Code Rules for Determining Allowable Compressive Stresses," The Eighth International Conference on Pressure Vessel Technology, Montreal, Canada, July 25, 1996.

Miller, C.D. (1991), "ASME Code Case N-284: Metal Containment Shell Buckling Design Methods," Revision 1, May 1991.

Miller, C.D. and Saliklis, E.P. (1993), "Analysis of Cylindrical Shell Database and Validation of Design Formulations," API Project 90-56, October 1993.

Miller, C.D., "Experimental Study of the Buckling of Cylindrical Shells With Reinforced Openings," ASME/ANS Nuclear Engineering Conference, Portland, Oregon, July 1982.

Miller, C.D. and Mokhtarian, K., "Allowable Tolerances for External Pressure Design," PVRC Grant 94-28, Final Report, CBI Technical Services Co. To be published as Welding Research Council Bulletin ___

PROPOSED RULES FOR DETERMINING ALLOWABLE COMPRESSIVE STRESSES FOR CYLINDERS, CONES, SPHERES AND FORMED HEADS

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WELDING RESEARCH COUNCIL

REASONS FOR ALTERNATIVE RULES

1. MANY TEST PROGRAMS HAVE BEEN CONDUCTED SINCE THE CURRENT ASME SECTION VIII RULES WERE FORMULATED

2. THE MARGINS OF SAFETY BETWEEN THE BUCKLING STRESSES AND THE ALLOWABLE STRESSES VARY WITH THE TYPE OF SHELL AND LOAD (FROM 2 TO 4)

3. THE PRESENT ASME RULES MAKE NO PROVISIONS FOR COMBINED LOADS

ALTERNATIVE RULES

1. BUCKLING STRESS EQUATIONS ARE BASED UPON THEORETICAL BUCKLING EQUATIONS WHICH HAVE BEEN REDUCED BY KNOCKDOWN FACTORS AND PLASTICITY REDUCTION FACTORS DETERMINED FROM TESTS

2. THE ALLOWABLE STRESS EQUATIONS ARE OBTAINED BY APPLYING A STRESS FACTOR (FACTOR OF SAFETY) TO THE BUCKLING STRESS

3. EQUATIONS ARE GIVEN FOR:

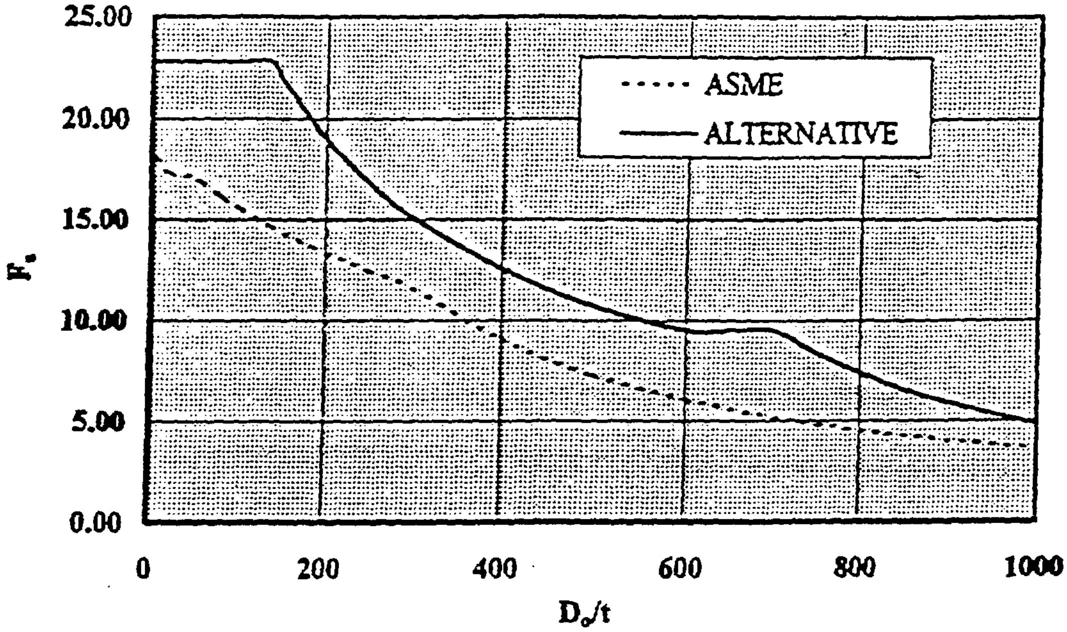
   a. COLUMN BUCKLING

   b. BENDING OVER FULL CROSS-SECTION FOR CYLINDERS AND CONES

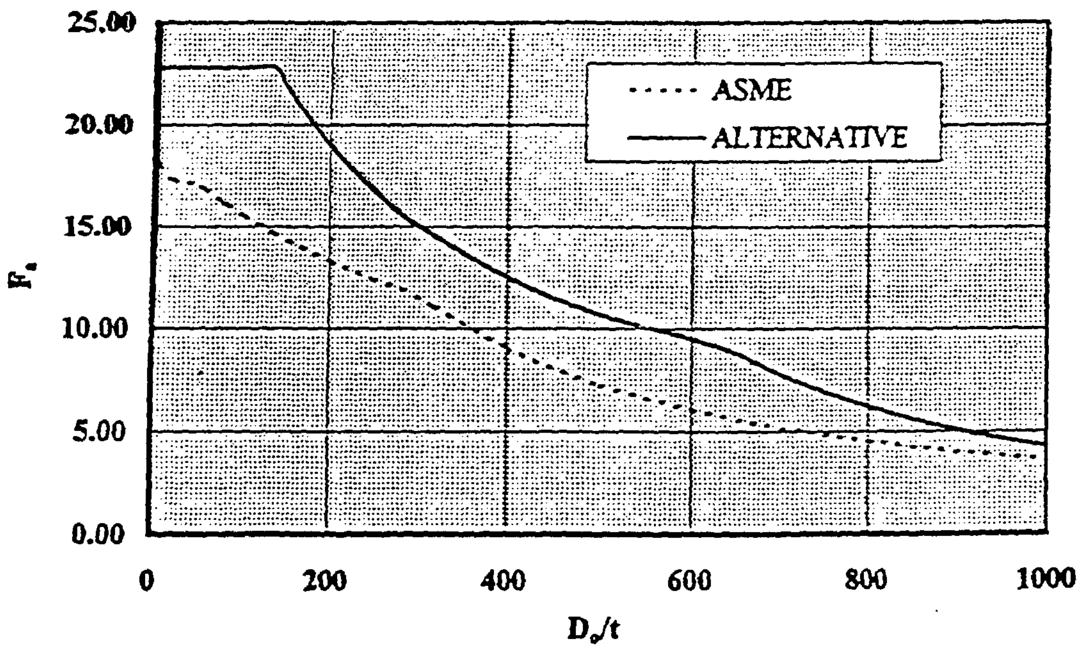
   c. RING STIFFENED CYLINDERS AND CONES SUBJECTED TO AXIAL COMPRESSION

   d. CYLINDERS AND CONES UNDER COMBINED LOADS

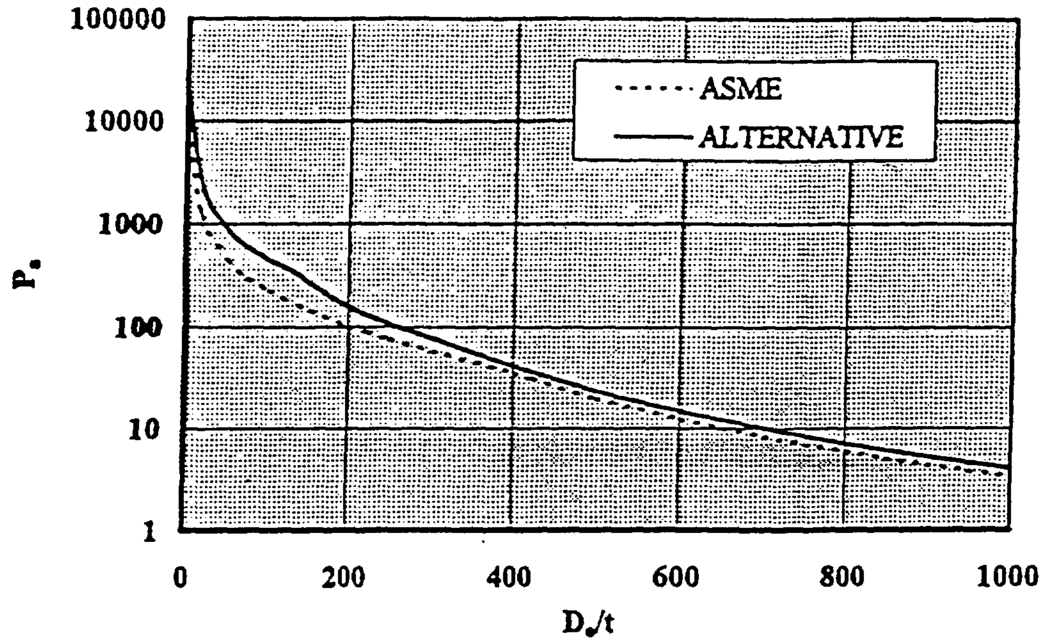
   e. FORMED HEADS UNDER UNEQUAL BIAXIAL COMPRESSION STRESSES



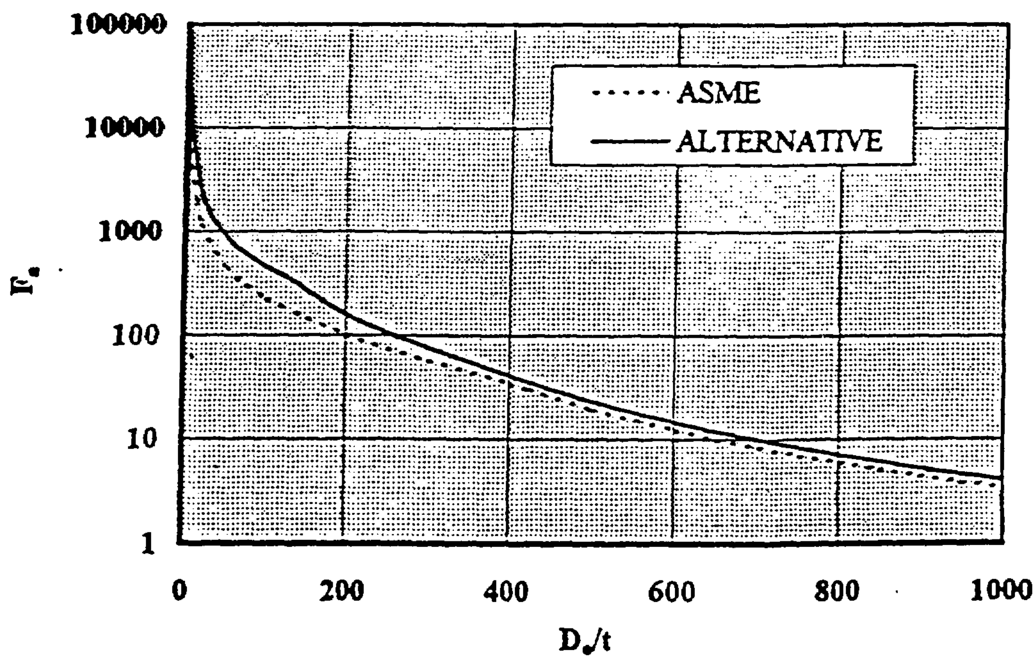
CYLINDERS UNDER AXIAL COMPRESSION $L/D_o = 0.25$



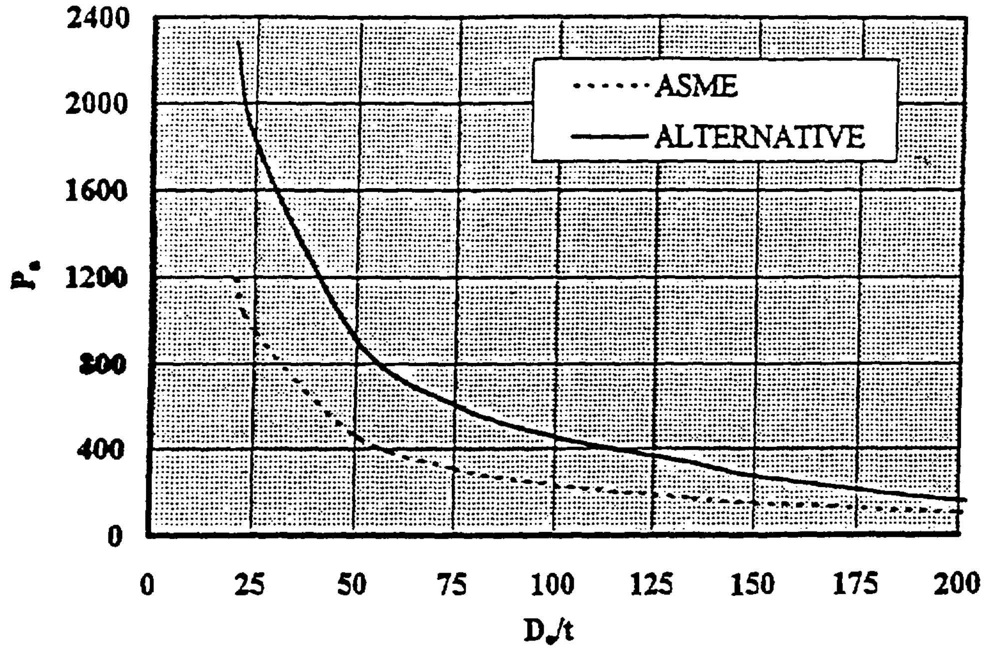
CYLINDERS UNDER AXIAL COMPRESSION $L/D_o = 10$



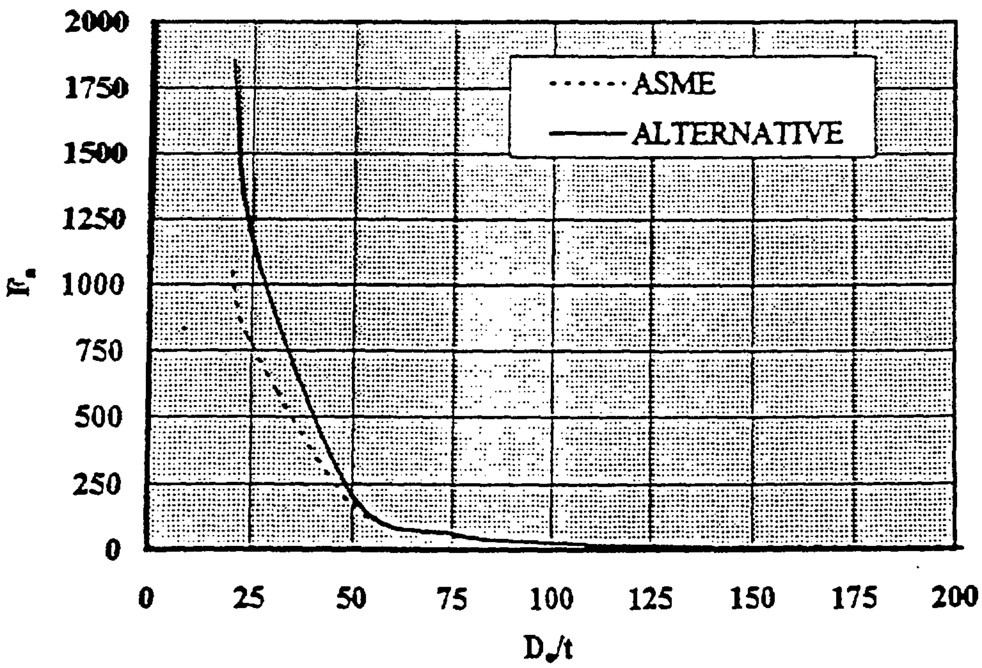
CYLINDERS UNDER EXTERNAL PRESSURE $L/D_o = 0.25$



CYLINDERS UNDER EXTERNAL PRESSURE $L/D_o = 10$



CYLINDERS UNDER EXTERNAL PRESSURE $L/D_o = 0.25$



CYLINDERS UNDER EXTERNAL PRESSURE $L/D_o = 10$

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THEORETICAL BACKGROUND AND LIMITATIONS

Basis: Shell Theory analysis by Prof. P.P. Bijlaard of Cornell University. Prof. Bijlaard employed a double Fourier series solution due to applied distributed loads over a rectangular region on the vessel.

Simplifying Assumptions:

1) Shallow Shell Theory used

2) Flexible, distributed loading surface assumed for cylindrical vessels

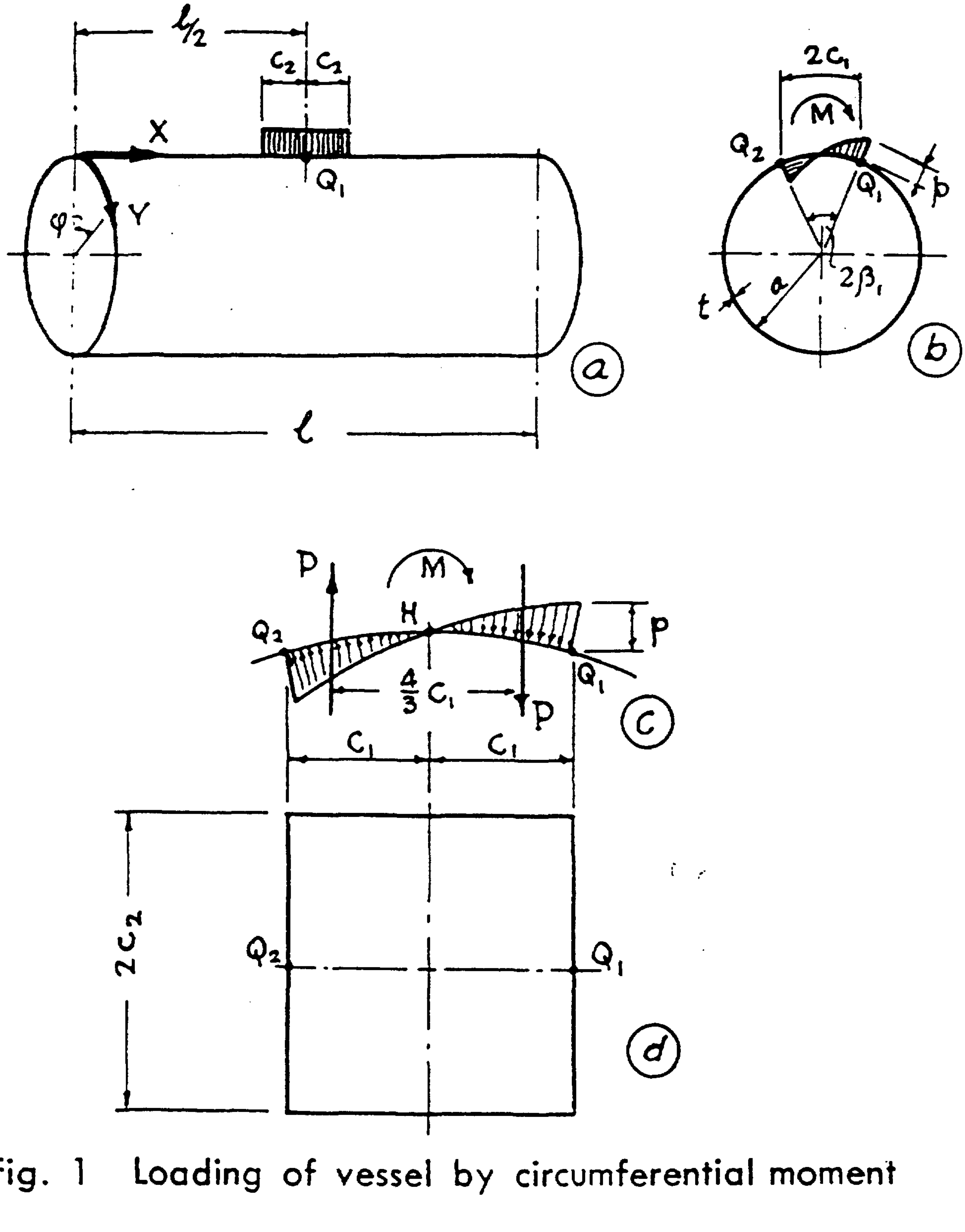

Fig. 1 Loading of vessel by circumferential moment

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Limitations:

Spherical Shells

$d/D \leq .33$

$U \leq 2.2$

$.25 \leq t/T \leq 10$

$5 \leq r_m/t \leq 50$

Cylindrical Shells

$d/D \leq .25$ (1)

$D/T \leq 600$

$L/D > 1.5$

$1/4 \leq C_1/C_2 \leq 4$

Where:

- d = Nozzle inside diameter
- D = Vessel inside diameter
- L = Vessel length
- C_1/C_2 = Half width and length of rectangular attachments

(1) Subsequent revisions have conservatively extended the theory to larger diameter ratios, approaching .6. However, the inaccuracies increase as the diameter ratio increases beyond ⅓.

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Moment loading in spheres
Comparison of Experimental and Theoretical Stresses

Geometry				Stresses (KSI)				
		$\rho =$	$\gamma =$	Bijlaard		Experimental [1]		Penny -
D_m/T	d_i/D_i	T/t	r_m/t	σ_y	σ_x	σ_y	σ_x	Lecki [2]
51.0	.5	2.0	25.5	2.72	2.13	4.81	5.03	3.64
51.0	.27	2.0	14.0	3.18	2.51	4.59	4.78	3.37
51.0	.27	4.0	27.5	2.40	.55	3.83	3.97	2.52
51.0	.129	4.0	13.9	2.11	.45	2.73	2.81	2.09

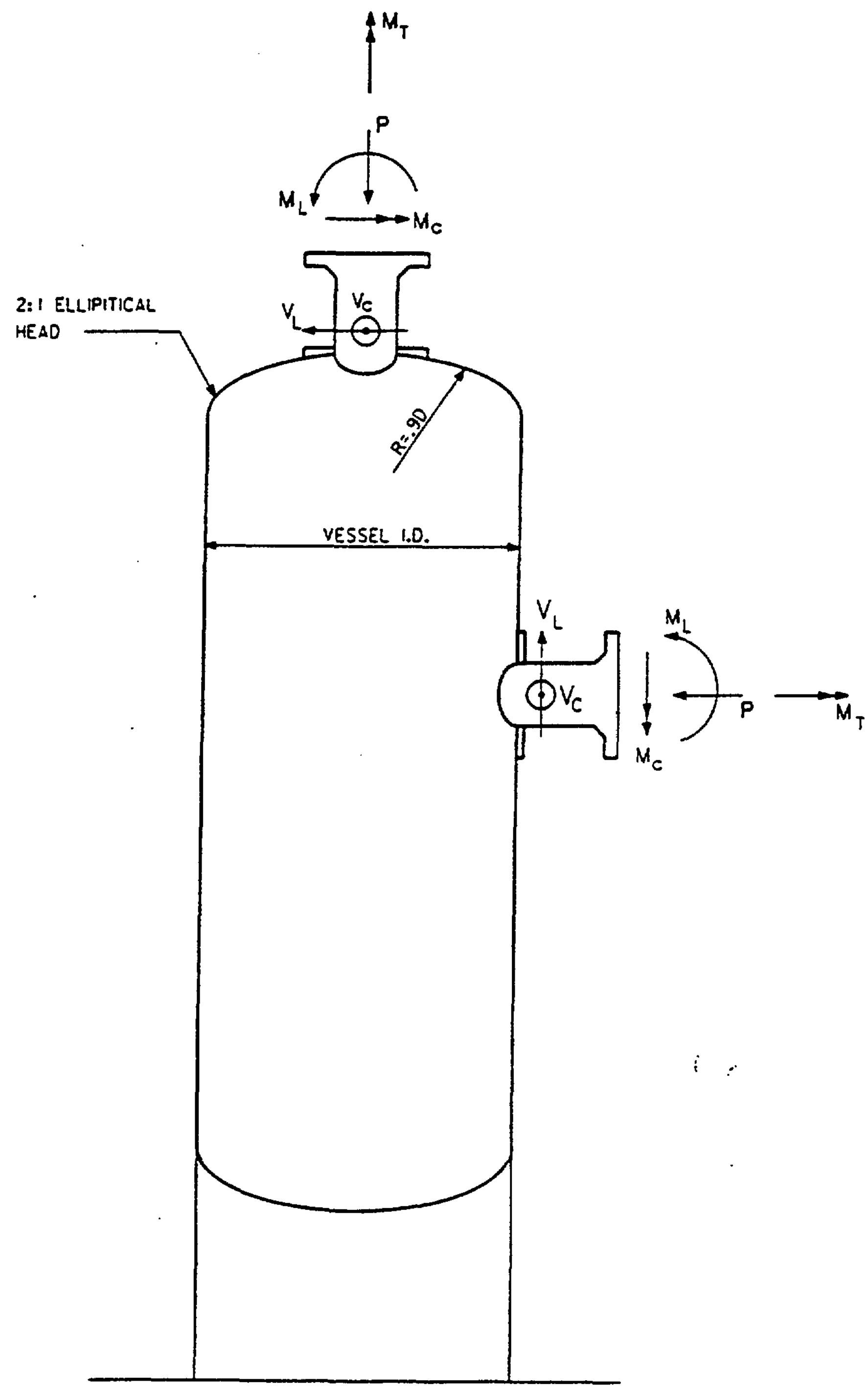
[1] Based on photoelastic models. Maximum stresses were in the nozzle.
[2] Based on Shell Theory without the shallow shell assumption.

Parameters

In Shell Theory, certain dimensionless parameters occur that, for convenience, allow you to plot the data with fewer variables.

Variables	**Parameters**
	Round Attachments in Spheres
r_o, r_m = Nozzle Outside, Mean Radius	$U = r_o / \sqrt{R_m T}$
R_m = Vessel Mean Radius	$\Upsilon = r_m/t$
t = Nozzle Thickness	$\rho = T/t$
T = Vessel Thickness	**Round attachments in Cylinders**
	$\gamma = R_m / T$
	$\beta = .875 \frac{r_o}{R_m}$
	Rectangles attachments in Cylinders
C_1 = Half width in circumferential direction	$\gamma = R_m / T$
	$\beta = C_i/R_m$ (Square)
C_2 = Half width in longitudinal direction	$\beta = f\left(\frac{C_1}{R_m}, \frac{C_2}{R_m}\right)$ (Rectangular)

EXTERNAL LOADS ON NOZZLES



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WRC 107
Sphere with a Hollow Attachment

Example: **10′ I.D. vessel with a 2:1 elliptical head and a 25.25″ O.D. nozzle located on the vessel centerline. Vessel thickness is 1″ and the nozzle thickness is .25″. Loading is a negative radial (outward) load P with combined overturning moment M, where $M = \sqrt{M_X^2 + M_Y^2}$, a Torsional Moment, M_T and Transverse Shear, V.**

$P = -20{,}000$ lbs.

$M = 500{,}000$ in-lbs

$M_T = 100{,}000$ in-lbs

$V = -10{,}000$ lbs

Where M and V act in the directions shown in the figure for M_L and V_L respectively. (Note: Dimensions selected for convenience in reading curves)

Spherical radius $R_i = .9(120) = 108''$

$R_m = 108.625$

$\gamma = r_m/t = 12.5 / .25 = 50.0$

$\rho = T/t = 4$

$U = r_o/\sqrt{R_m T} = 12.625/\sqrt{108.625} = 1.21$

Figure SP-9 for Radial Load, P

$\dfrac{N_y T}{P} = .207 \qquad \dfrac{M_y}{P} = .0197$

$\dfrac{N_x T}{P} = .0109 \qquad \dfrac{M_x}{P} = .00929$

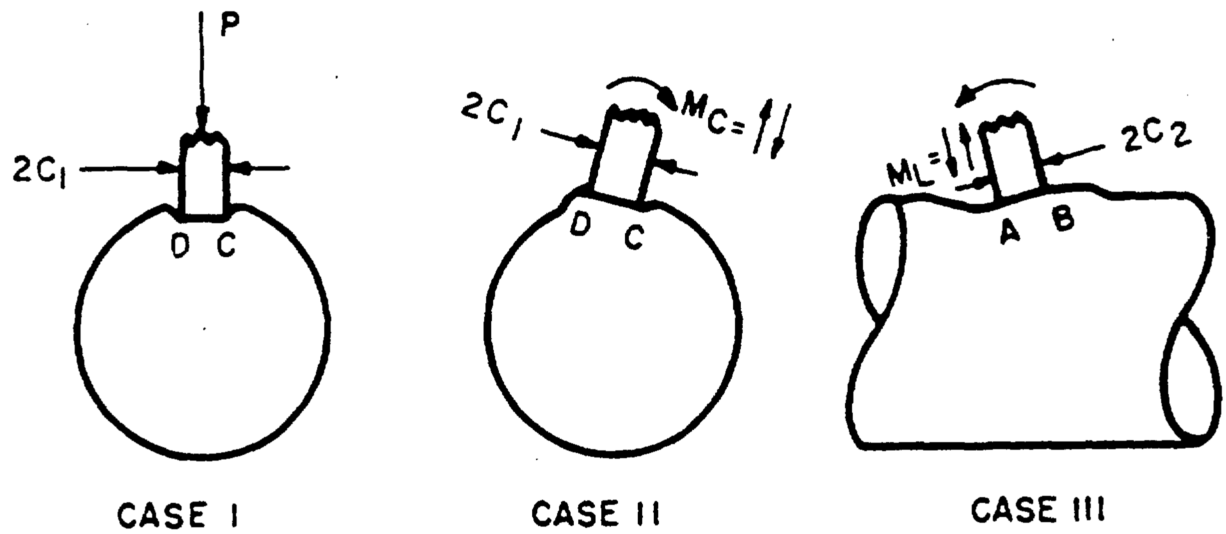
Figure SM-9 for External Moment, Cos (Θ) = 1.0

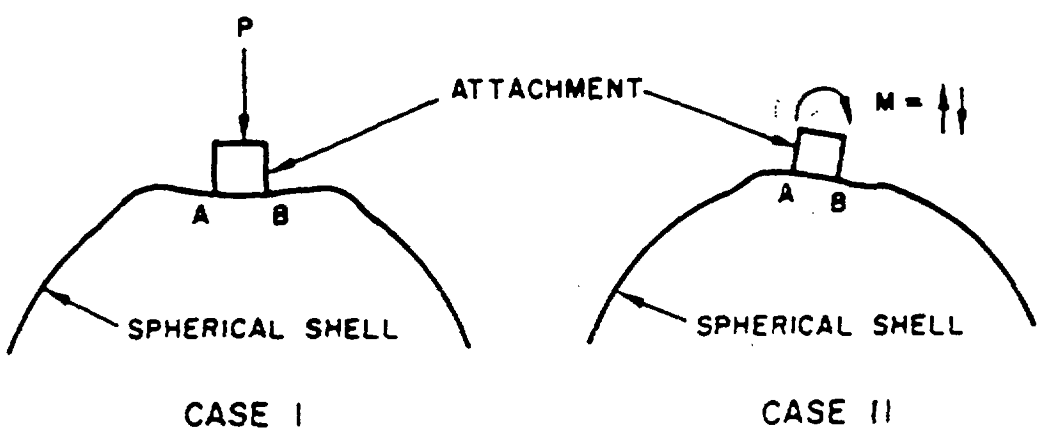
$N_x T\sqrt{R_m T} / M = .0169 \qquad M_x\sqrt{R_m T} / M = .0153$

$N_y T\sqrt{R_m T} / M = .226 \qquad M_y\sqrt{R_m T} / M = .0384$

Sign Convention

A knowledge of the shell deflections resulting from various modes of loading permits one to predict whether resulting stresses will be tensile (+) or compressive (−).





Component Stresses at Maximum Stress Location B_u
Radial Stresses, σ_x

Local Membrane Stress due to P

$$\sigma_x = -\left(\frac{N_xT}{P}\right)\frac{P}{T^2} = -.0109\left(\frac{-20000}{1}\right) = 216\ \text{PSI}$$

Local Membrane Stress due to M

$$\sigma_x = \left(\frac{N_xT\sqrt{R_mT}}{M}\right)\frac{M}{T^2\sqrt{R_mT}} = .0169\left(\frac{500000}{10.422}\right) = \underline{812}\ \text{PSI}$$

Total Local Radial Membrane Stress σ_x = 1028 PSI

Bending Stress Due to P

$$\sigma_x = -\left(\frac{M_x}{P}\right)\frac{6P}{T^2} = -.00929\left(\frac{-120000}{1}\right) = 1115\ \text{PSI}$$

Bending Stress Due to M

$$\sigma_x = \left(\frac{N_xT\sqrt{R_mT}}{M}\right)\frac{6M}{T^2\sqrt{R_mT}} = .0153\left(\frac{500000}{1.737}\right) = \underline{4394}\ \text{PSI}$$

Total Radial Bending Stress σ_x = 5509 PSI

Total Radial Membrane & Bending Stress σ_x = 6537 PSI

Tangential Stresses, σ_y

Local Membrane Stress due to P

$$\sigma_y = -\left(\frac{N_yT}{P}\right)\frac{P}{T^2} = -.207\left(\frac{-20000}{1}\right) = 4149 \text{ PSI}$$

Local Membrane Stress due to M

$$\sigma_y = \left(\frac{N_yT\sqrt{R_mT}}{M}\right)\frac{M}{T^2\sqrt{R_mT}} = .226\left(\frac{500000}{10.422}\right) = \underline{10{,}824} \text{ PSI}$$

Total Local Tangential Membrane Stress $\sigma_y = 14{,}973$ PSI

Bending Stress Due to P

$$\sigma_y = -\left(\frac{M_y}{P}\right)\frac{6P}{T^2} = -.0197\left(\frac{-20000}{1}\right) = 2359 \text{ PSI}$$

Bending Stress Due to M

$$\sigma_y = \left(\frac{M_y\sqrt{R_mT}}{M}\right)\frac{6M}{T^2\sqrt{R_mT}} = .0389\left(\frac{500000}{1.737}\right) = \underline{11{,}041} \text{ PSI}$$

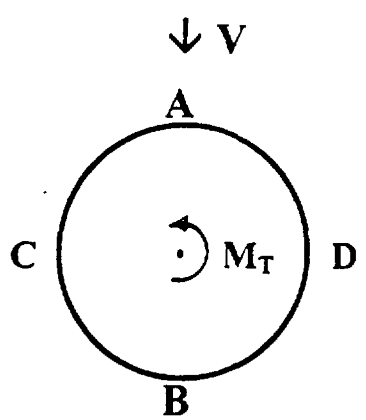
Total Bending Stress σ_y = 13,400 PSI

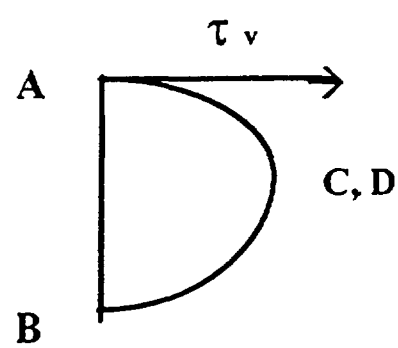
Total Tangential Membrane & Bending Stress $\sigma_y = 28{,}373$ PSI

Transverse Shear V

WRC 107 handles a transverse shear load the same as developed from Beam Theory. The transverse shear should be the shear force at the location of the nozzle to shell intersection and the shear stress is assumed to vary in a parabolic manner.

For the figure shown below the transverse shear is zero at A and B and a maximum at C and D. Since it acts in one direction it adds to the torsional shear at one location and subtracts at another.





The torsional shear is assumed constant around the perimeter acting in the direction of M_T. The torsion and transverse shears add at Location C and Subtract at Location D.

Transversal Shear Stress at Location C_u

$$\tau_v = \frac{-V}{\pi r_o T} = \frac{-10{,}000}{\pi(12.625)(1)} = 252 \text{ PSI}$$

The Transverse Shear at Location B_u = 0

Shear Stress Due to Torsion at Location B_u

$$\tau_T = \frac{M_T}{2\pi r_o^2 T} = \frac{-100{,}000}{2\pi(12.625)^2(1)} = 100 \text{ PSI}$$

Total Shear Stress, at Location B_u, τ = 226 PSI

Combined Stress Intensity is the Greater of:

$$S = \tfrac{1}{2}\ (\sigma_x + \sigma_y \pm \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2)}$$

or

$$S = \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2)}$$

Combined Membrane plus Bending Stress Intensity, at B_u:

$$S = \tfrac{1}{2}\ (6537 + 28373 \pm \sqrt{(6537 - 28373)^2 + 4(100)^2)}$$

$$= \tfrac{1}{2}\ (34910 \pm 21837)$$

$$= 28373 \text{ PSI}$$

Local Membrane Stress Intensity, at B_u:

$$S = \tfrac{1}{2}\ (14973 + 1028 \pm \sqrt{(1028 - 14973)^2 + 4(100)^2)}$$

$$= \tfrac{1}{2}\ (16001 \pm 13946)$$

$$= 14974 \text{ PSI}$$

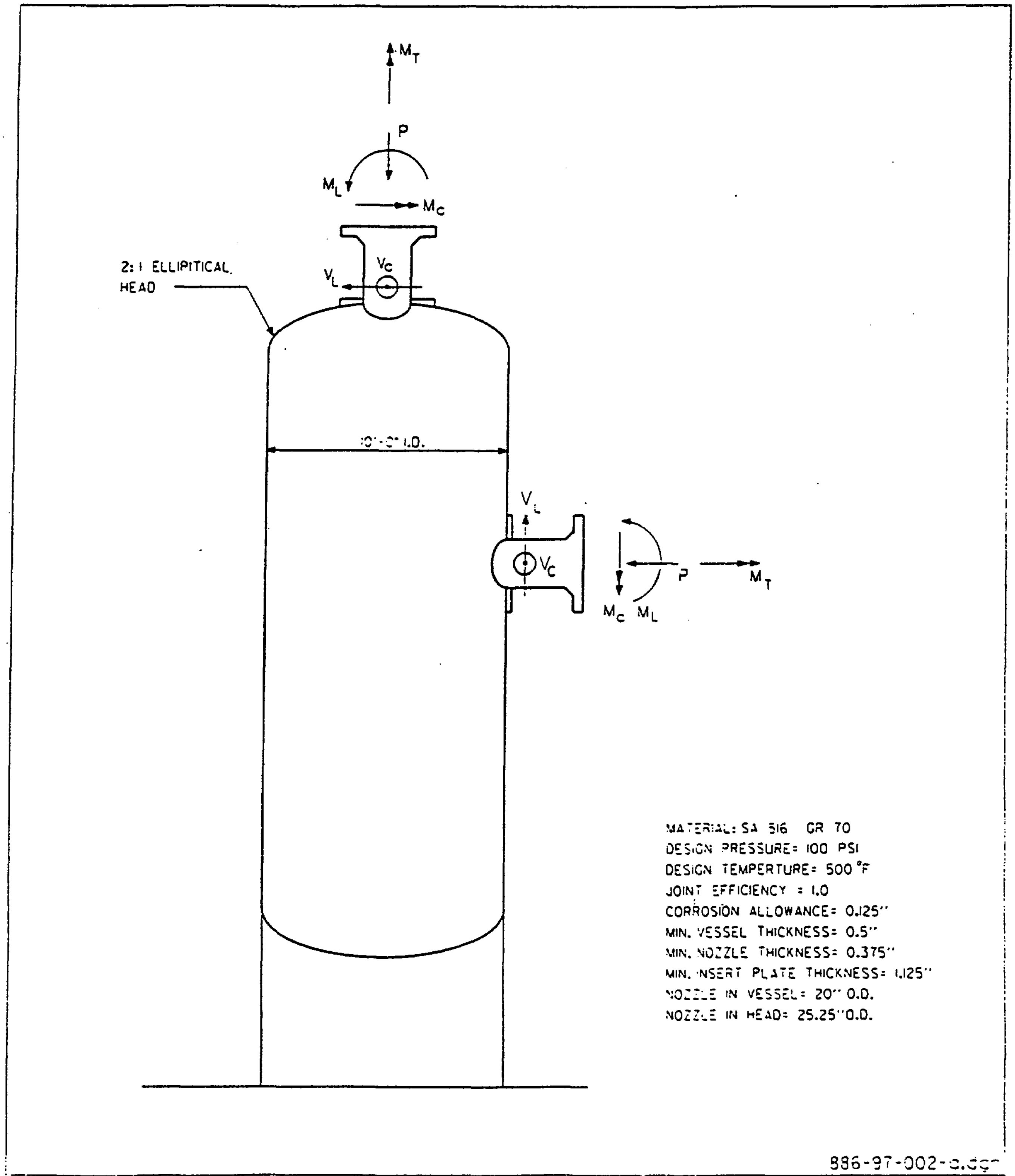
WRC 107

Nozzle in a Cylindrical Shell

Example: 10' I.D. Vessel with a 20.0" O.D. Nozzle. The vessel thickness is 1" and the nozzle thickness is .25".

Loading:

Radial Load, P	=	**-20,000 lbs**
Circumferential Moment, M_c	=	**353,553 in-lbs**
Longitudinal Moment, M_L	=	**353,553 in-lbs**
(Note: $M = \sqrt{M_c^2 + M_L^2}$	=	**500,000 in-lbs)**
Transverse Shear, V_c	=	**- 7,070 lbs**
V_L	=	**- 7,070 lbs**
Torsional Moment, M_T	=	**100,000 in-lbs**



M_T
P
M_L
M_C
2:1 ELLIPITICAL HEAD
V_C
V_L
10'-0" I.D.
V_L
V_C
P
M_T
M_C M_L
MATERIAL: SA 516 GR 70
DESIGN PRESSURE= 100 PSI
DESIGN TEMPERTURE= 500 °F
JOINT EFFICIENCY = 1.0
CORROSION ALLOWANCE= 0.125"
MIN. VESSEL THICKNESS= 0.5"
MIN. NOZZLE THICKNESS= 0.375"
MIN. INSERT PLATE THICKNESS= 0.125"
NOZZLE IN VESSEL= 20" O.D.
NOZZLE IN HEAD= 25.25" O.D.
886-97-002-

Vessel Design Example

ASME Section VIII Division 1 Allowable Stress - 17500 PSI

Vessel Shell Required Thickness:

$$t = \frac{PR}{SE-.6P} = \frac{100\,(60.125)}{17500(1.0)-.6(100)}$$

$$\begin{aligned} t &= .345'' \\ &\underline{+.125''} \quad \text{Corrosion} \\ &.470'' \quad \text{Minimum required} \end{aligned}$$

Use $t = .5''$

Vessel Head (2:1 Elliptical) Required Thickness:

$$t = \frac{PD}{2SE-.2P} = \frac{100\,(120.25)}{2(17500)-.2(100)}$$

$$\begin{aligned} t &= .344'' \\ &\underline{+.125''} \quad \text{Corrosion} \\ &.469'' \quad \text{Minimum Required} \end{aligned}$$

Use $.5''$

Use an insert plate for reinforcement
with a thickness = 1.125".

Calculate Parameters

- Shell Parameter: $\gamma = \frac{Rm}{T} = \frac{60.625}{1} = 60.625$

- Attachment Parameter: $\beta = .875 \frac{r_0}{Rm} = .875 \frac{9.875}{60.625}$

$\beta = .143$

Determine the Dimensionless Membrane Forces and Bending Moments

For Radial Load P

Determine the membrane forces at the transverse and longitudinal axis locations from Figures 3C and 4C.

Fig. 3C:	$N\phi/(P/R_m)$ Transverse Axis	=	6.35
Fig. 3C:	$N_x/(P/R_m)$ Longitudinal Axis	=	6.35
Fig. 4C:	$N_x/(P/R_m)$ Transverse Axis	=	9.00
Fig. 4C:	$N\phi/(P/R_m)$ Longitudinal Axis	=	9.00

From Figures 1C, 1C-1, 2C, 2C-1 Determine the Dimensionless Bending Moments.

Fig 1C:	$M\phi/P$ Transverse Axis	=	.078
Fig 1C-1:	M_x/P Longitudinal Axis	=	.080
Fig 2C:	M_x/P Transverse Axis	=	.044
Fig 2C-1:	$M\phi/P$ Longitudinal Axis	=	.044

Bending Moment M_L

Membrane Forces on Longitudinal Plane:

Fig 3B	$N_\phi/(M_L/R_m^2\beta)$	= 6.52
Fig 4B	$N_x/(M_L/R_m^2\beta)$	= 2.38

Bending Moments on Longitudinal Plane:

Fig 2B	$M_x/(M_L/R_m\beta)$	= .041
Fig 1B	$M_\phi/(M_L/R_m\beta)$	= .029

Maximum Bending Moments:

Fig 2B-1	$M_x/(M_L/R_m\beta)$	= .042
Fig 1B-1	$M_\phi/(M_L/R_m\beta)$	= .029

Bending Moment M_c

Membrane Forces on Transverse Plane:

Fig 3A	$N_\phi/(M_c/R_m^2\beta)$	= 2.410
Fig 4A	$N_x/(M_c/R_m^2\beta)$	= 4.25

Bending Moments on Transverse Plane:

Fig 1A	$M_\phi/(M_c/R_m\beta)$	= .079
Fig 2A	$M_x/(M_c/R_m\beta)$	= .039

Stress Multipliers

$P/R_mT = -20{,}000/60.625\,(1) = -20{,}000/60.625 = -330$

$M_c/R_m^2\beta T = 353{,}553/60.625^2\,(.143)(1) = 353{,}553/525.6 = 672.7$

$M_L/R_m^2\beta T = 353{,}553/60.625^2\,(.143)(1) = 353{,}553/525.6 = 672.7$

$6P/T^2 = -6(20000)/1^2 = -6\,(20000) = -120{,}000$

$6M_c/R_m\beta T^2 = 6(353{,}553)/60.625\,(.143)(1)^2 = .6921(353{,}553) = 244{,}690$

$6M_L/R_m\beta T^2 = 6(353{,}553)/60.625\,(.143)(1)^2 = .6921(353{,}553) = 244{,}690$

Stress Calculations

Vessel Circumferential Stress at Location Bu

Stress Type (Membrane/ Bending)	Loading	Circumferential Stress $\sigma\phi$(PSI)		
M	P	$-[N\phi/(P/Rm)]\,[P/RmT] = -9.00\ (-330)$	=	2970
M	Mc	$[N\phi/(Mc/Rm^2\beta)]\left[\dfrac{Mc}{Rm^2\beta T}\right]$	=	--
M	ML	$[N\phi/(ML/Rm^2\beta)]\left[\dfrac{ML}{Rm^2\beta T}\right] = 6.5\ (672.7)$	=	4386
		The Circumferential Membrane Stress at Bu	=	7356
B	P	$-[M\phi/P]\,[6P/T^2] = .044\ (-120{,}000)$	=	5280
B	Mc	$[M\phi/(Mc/Rm\beta)]\,[6Mc/(Rm\beta T^2)]$	=	--
B	ML	$[M\phi/(ML/Rm\beta)]\,[6ML/(Rm\beta T^2)] = .029(244690)$	=	7096 (1)
		Total Circumferential Bending Stress at Bu =		12376
		Total Membrane & Bending Stress at Bu =		19732

Notes: (1) Maximum Stress

Vessel Longitudinal Stress at Location Bu

Stress Type (Membrane/ Bending)	Loading	Longitudinal Stress σx (PSI)	
M	P	$[Nx/(P/Rm)]\ [P/RmT] = -6.35\ (-330)$	= 2096
M	Mc	$[Nx/(Mc/Rm^2\beta)] \left[\dfrac{Mc}{Rm^2\beta T}\right]$	= --
M	M_L	$[Nx/(M_L/Rm^2\beta)] \left[\dfrac{M_L}{Rm^2\beta T}\right] = 2.38(627.7)$	= 1601
	Total Longitudinal Membrane Stress at Bu		= 3697
B	P	$[Mx/P]\ [6P/T^2] = -.08\ (-120{,}000)$	= 9600
B	Mc	$[Mx/(Mc/Rm\beta)]\ [6Mc/(Rm\beta T^2)]$	= --
B	M_L	$[Mx/(M_L/Rm\beta)]\ [6M_L(Rm\beta T^2)] = .042(244690)$	= 10277 (1)
	Total Longitudinal Bending Stress at Bu		= 19877
Total Longitudinal Membrane & Bending Stress at Bu			= <u>23574</u>

Notes: (1) Maximum Stress

Combined Stresses

The Radial loads, overturning moments, transverse shears and torsion may have any direction or magnitude. The procedure in WRC 107 considers the stresses at 8 locations, however there is no assurance that the maximum stress does occur at one of the 8 locations.

It has been found in some cases that certain of the stress components may peak at points slightly removed from the attachment in spherical shells. When that occurs, the maximum stresses are represented by a dashed line in the figures for the stress components in question.

The maximum stress in cylindrical shells often occurs at a location off of the longitudinal of symmetry of the vessel.

The work of Eringen and Van Dyke illustrates the behavior in the following figure. When determining the stresses due to a longitudinal moment, WRC 107 provides both curves for stresses on the plane of symmetry and the maximum stress that occurs at some other angle. When evaluating radial thrust loads, both the longitudinal and transverse planes must be considered, and curves are provided.

REVISED BULLETIN 297 SEPTEMBER 1987

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Welding Research Council
WRC
bulletin

LOCAL STRESSES IN CYLINDRICAL SHELLS DUE TO EXTERNAL LOADINGS ON NOZZLES— SUPPLEMENT TO WRC BULLETIN NO. 107— (Revision I)

by J. L. Mershon, K. Mokhtarian, G. V. Ranjan and E. C. Rodabaugh

These Bulletins contain final Reports from projects sponsored by the Welding Research Council, important papers presented before engineering societies and other reports of current interest.

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WELDING RESEARCH COUNCIL
UNITED ENGINEERING CENTER
345 EAST 47th STREET, NEW YORK, N.Y. 10017

Welding Research Council Bulletin 297

Basis

The design curves in WRC 297 were generated by the FAST 2 Computer Code developed by Prof. Steele. Flugge - Conrad and Sanders - Simmonds shell theory solutions were utilized to model the nozzle-shell intersection. Bessel function solutions of the shallow shell equations were computed assuming that the opening in the vessel is a circle and the edge of the nozzle is flat.

WRC 297 is a Supplement to WRC 107

- The effects of nozzle neck thickness is accounted for
- Stresses in the nozzle neck may be computed
- Data for larger D/T ratios is provided
- Nozzle Flexibilities can be computed
- Revision 1 changes relate to the stiffness factors in figures 59 and 60 of the Bulletin.

WRC 297

Limitations

- **Based on shallow thin Shell Theory, by Prof. Steele.**
- **Nozzle axis is normal to the vessel.**
- **$d/t \geq 20$ (However, reasonable results are obtained for any d/t)**

  $20 \leq D/T \leq 2500$

  $d/T \geq 5$

  $d/D \leq .5$ (Depends on D/T ratio)
- **Nozzle-Shell junction must be a distance $> 2\sqrt{DT}$ on the vessel and $> 2\sqrt{dt}$ on the nozzle away from a structural discontinuity.**

Note: Although the above limits are stated in the bulletin, recent work indicates that the results may be in question when the diameter ratio exceeds approximately ⅓.

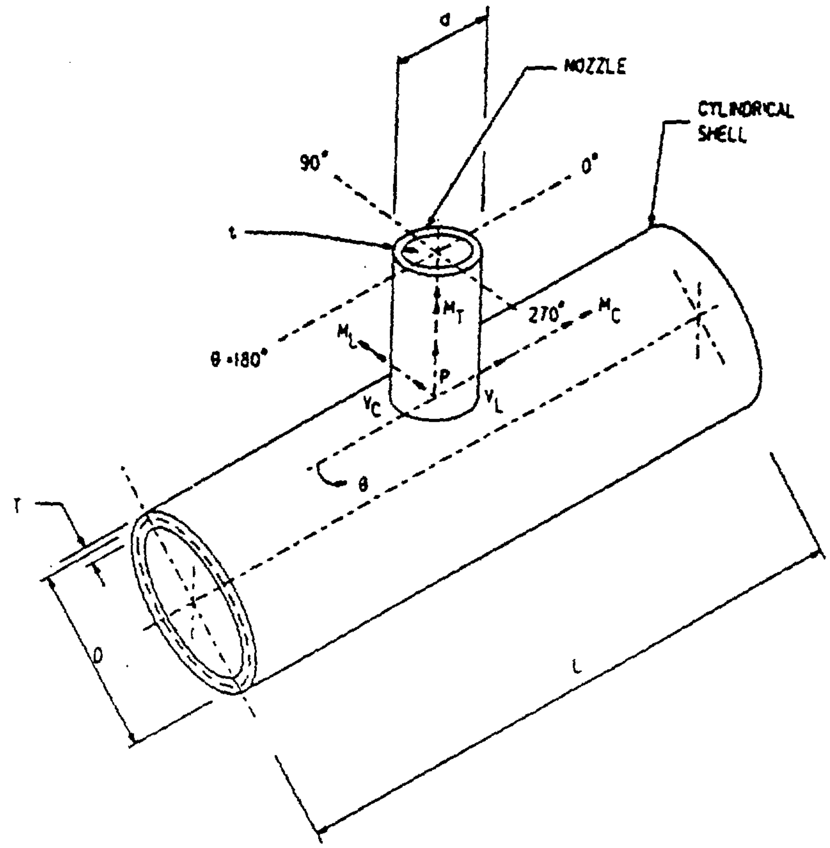

Fig. 1—Positive directions for external loads on radial nozzle in cylindrical shell and definition of geometry. Loads V_L, V_c, P, M_c, M_L, and M_T act at nozzle-to-shell intersection and form right-hand system.

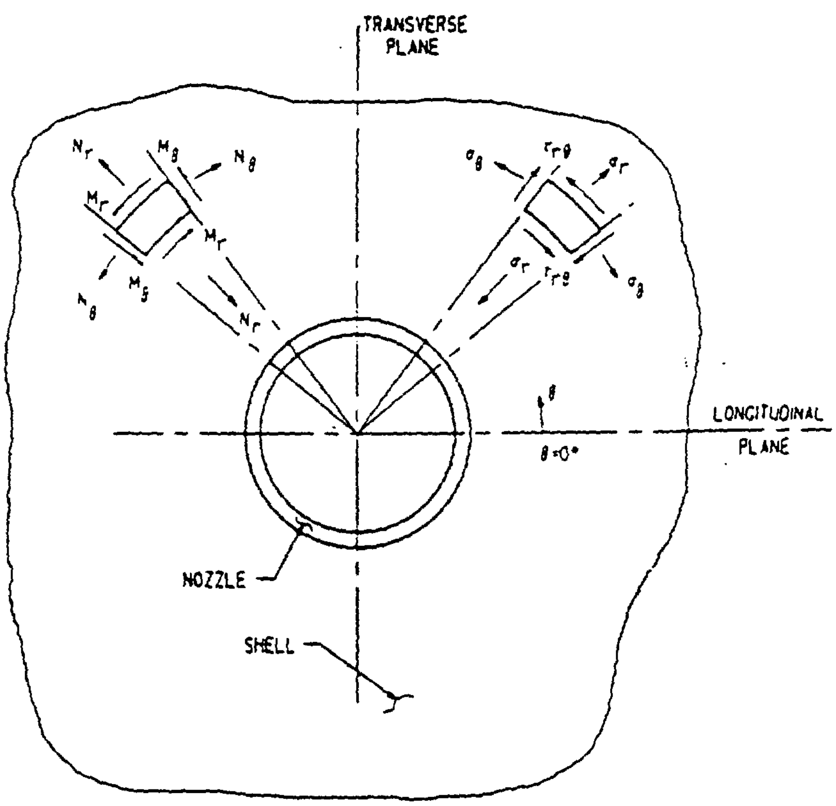

Fig. 2—Positive directions for stresses, internal membrane forces and internal bending moments in shell

WRC 297

Using the Nomenclature of WRC 297:

D	=	Mean Diameter of Vessel	= 121.25″
d	=	Outside Diameter of Nozzle	= 20
T	=	Vessel Thickness	= 1″
t	=	Nozzle Thickness	= .25″

Loading Sign Convention:

P = 20000 lbs
Vc = -7070 lbs
VL = -7070 lbs
Mc = -353553 in-lbs
ML = 353553 in-lbs
MT = 100000 in-lbs

$$\lambda = (d/D)(D/T)^{1/2} = \left(\frac{20}{121.75}\right)\left(\frac{121.75}{1}\right)^{1/2} = 1.816$$

d/t = 20/.25 = 80
T/t = 1/.25 = 4

Fig. 6/7 $\frac{Mr}{P} = .052$

Fig. 11/12 $\frac{NrT}{P} = .032$

Fig. 16/17 $\frac{M_\theta}{P} = .061$

Fig. 21/22 $N_\theta T/P = .232$
Fig. 25/26 Mrd/Mc = .116
Fig. 30/31 NrTd/Mc = .067
Fig. 34/35 $M_\theta d/Mc = .179$
Fig. 39/40 $N_\theta Td/Mc = .362$
Fig. 43/44 Mrd/ML = .071

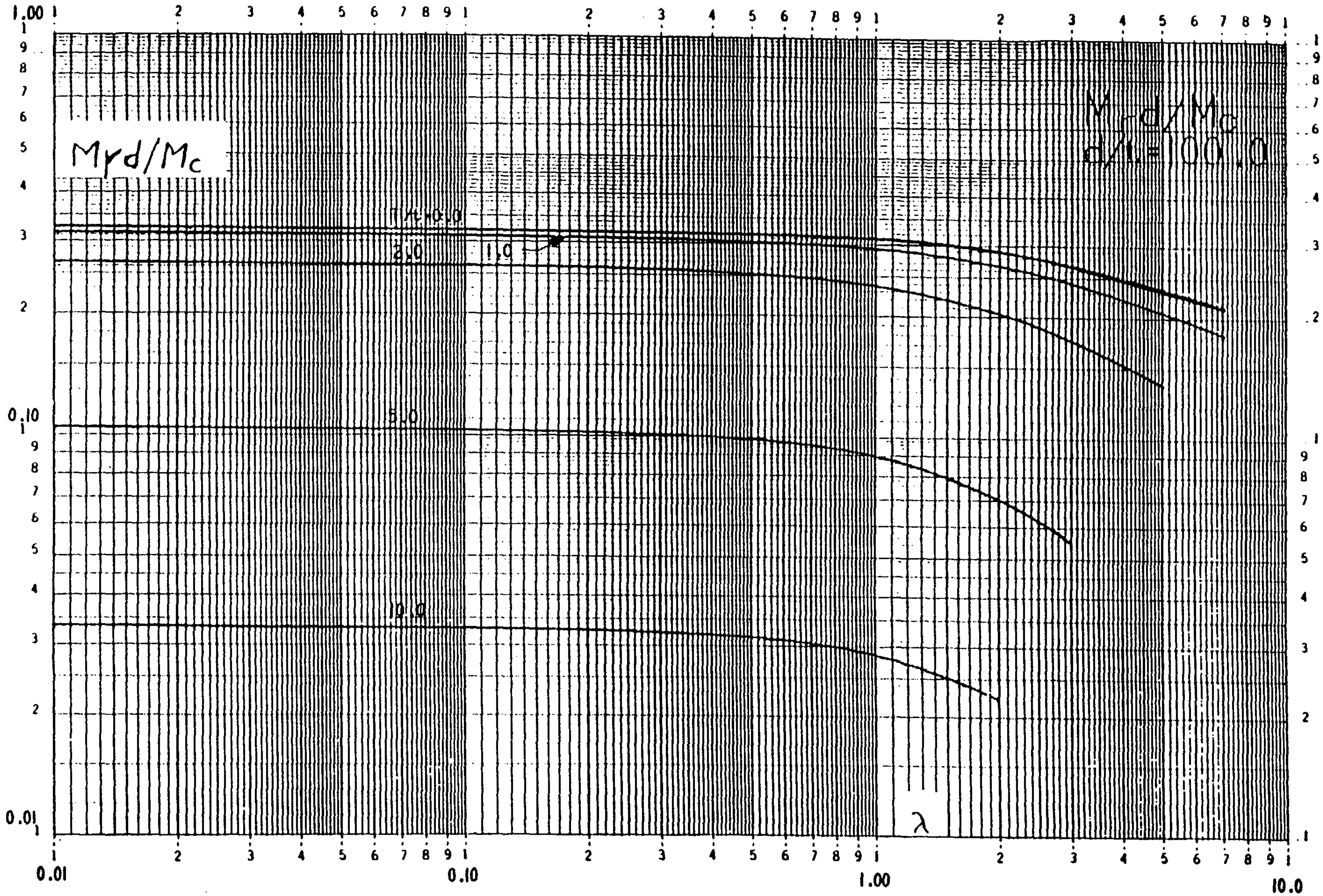

Fig. 26—internal moment stress resultants m_y due to M_c-load—(d/t = 100.0)

Stresses in Cylindrical Shells

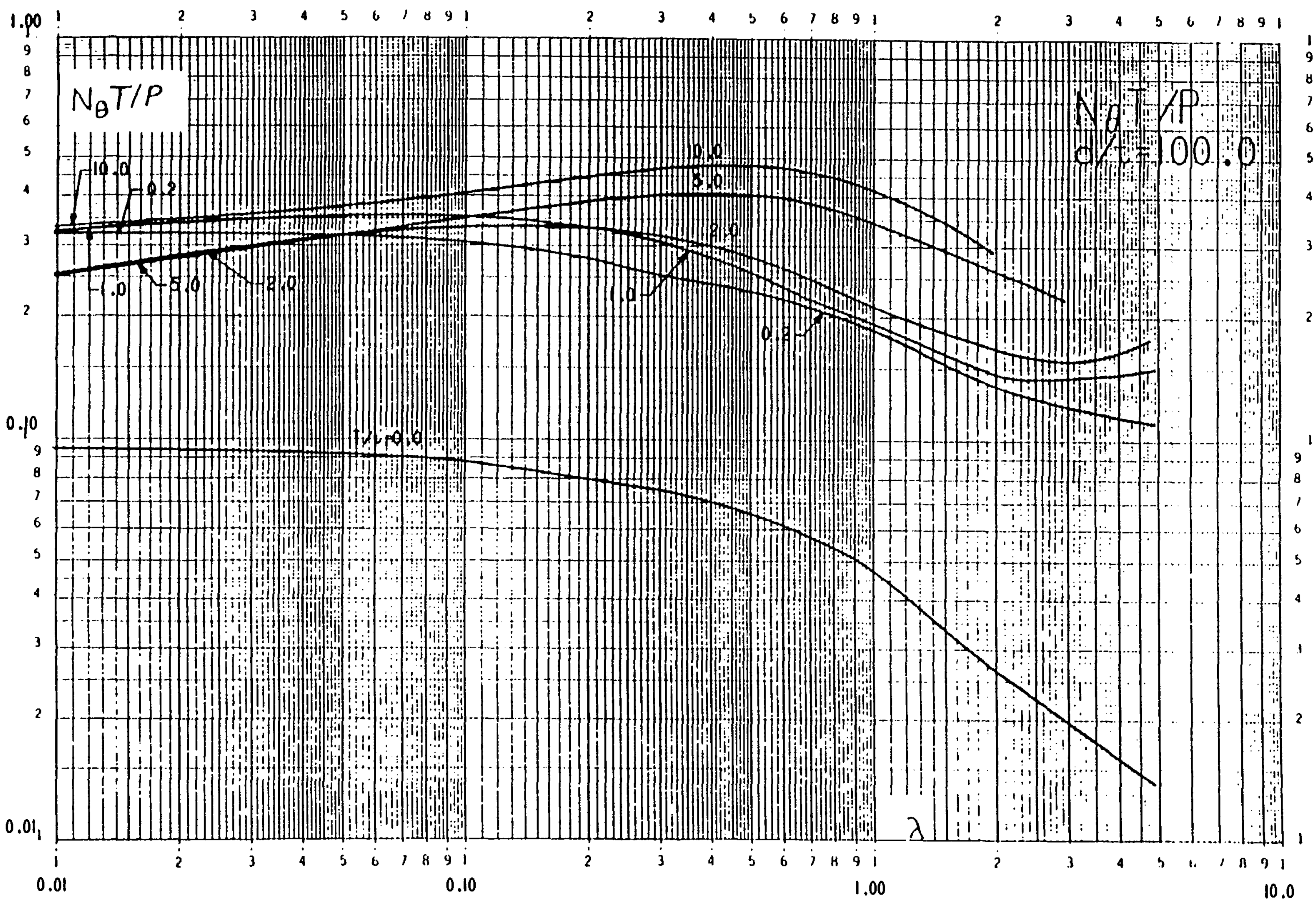

Fig. 22—Internal force stress resultants n_θ due to P-load—(d/t = 100.0)

WRC 297
(Continued)

Fig. 48/49 $N_rTd/M_L = .045$
Fig. 52/53 $M_\theta d/M_L = .089$
Fig. 57/58 $N_\theta Td/M_L = .505$

Nozzle Stiffness

For a Length L = 220″ (or ~ 2 Diameters)

$$\Lambda = \frac{L}{\sqrt{DT}} = \frac{220}{\sqrt{121.75(1)}} = 20$$

Fig 59 $\alpha = 3.85$

Fig 60 $\frac{M_L}{ET^3(ROT)} = 5.6$

Stress Multipliers

$$\frac{P}{T^2} = \frac{20000}{T^2} = 20000$$

$$\frac{Mc}{T^2d} = \frac{353553}{1^2(20)} = 17680$$

$$\frac{M_L}{T^2d} = \frac{353553}{1^2(20)} = 17680$$

$$\frac{6Mc}{T^2d} = 6\ \frac{353553}{1^2(20)} = 106070$$

$$\frac{6M_L}{T^2d} = 6\ \frac{353553}{1^2(20)} = 106070$$

Radial Stresses in Vessel at Location Θ=180, Outside Surface

Membrane Stress (PSI)

Loading

P	$(NrT/P)\ (P/T^2)$ =	.032 (20000)	=	649
M_c	$(NrTd/M_c)\ (M_c/T^2d)$		=	--
M_L	$(NrTd/M_L)\ (M_L/T^2d)$ =	.045 (17680)	=	799
				1448

Bending Stress (PSI)

P	$(Mr/P)\ (6P/T^2)$ =	.052 (120000)	=	6252
M_c	$(Mrd/M_c)\ (6M_c/dT^2)$		=	--
M_L	$(Mrd/M_L)\ (6M_L/dT^2)$ =	.071 (106070)	=	7499
				13751

Total Radial Membrane + Bending Stress = 15199

Tangential Stresses in Vessel at Location Θ=180, Outside Surface

Membrane Stress (PSI)

Loading

P	$(N\Theta T/P)\ (P/T^2)$ =	.232 (20000)	=	4639
M_c	$(N\Theta Td/M_c)\ (M_c/T^2 d)$		=	--
M_L	$(N\Theta Td/M_L)\ (M_L/T^2 d)$	= .505 (17680)	=	8934
				13573

Bending Stress (PSI)

P	$(M\Theta/P)\ (6P/T^2)$ =	.061 (120000)	=	7337
M_c	$(M\Theta d/M_c)\ (6M_c/dT^2)$		=	--
M_L	$(M\Theta d/M_L)\ (6M_L/dT^2)$	= .089 (106070)	=	9446
				16783

Total Tangential Membrane + Bending Stress = 30356

Outplane Shear, V_c:

τ_v	=	$-2V_c/(\pi dT) = -2(-7070)/\pi(20)(1)$	=	225
τ_T	=	$2M_T/(\pi d^2 T) = 2(100000)/\pi\, 20^2(1)$	=	<u>159</u>
		Total Shear (PSI)	=	384

<u>Surface Stress Intensity</u>

$$S = \tfrac{1}{2}\left(\sigma_r + \sigma_\Theta \pm \sqrt{(\sigma r - \sigma_\Theta)^2 + 4\tau^2}\right)$$

$$S = \tfrac{1}{2}\left(15199 + 30356 \pm \sqrt{(15199 - 30356) + 4(384)^2}\right)$$

$$S = 30365 \text{ PSI}$$

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STRESSES IN INTERSECTING CYLINDERS SUBJECTED TO PRESSURE

K. Mokhtarian, J. S. Endicott

These Bulletins contain final Reports from projects sponsored by the Welding Research Council, important papers presented before engineering societies and other reports of current interest.

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WRC 368

Stresses in Intersecting Cylinders
Subjected to Pressure

Basis

Parametric correlation of stresses predicted by the FAST 2 (Prof. C.R. Steele) Computer Code. This is the same code as used in the development of WRC 297.

Limitations:

$$10 < D/T < 1000$$
$$4 < d/t < 1000$$
$$.1 < t/T < 3$$
$$.03 < d/D < .5$$
$$.3 < Dt/dT < 6$$
$$.3 < d/\sqrt{Dt} < 6.5$$

WRC 368 Parametric Representation

vessel

$$\sigma_v = [\bar{\sigma}]\left[\frac{d}{\sqrt{Dt}}\frac{PD}{2T}\right]$$

nozzle

$$\sigma_n = [\bar{\sigma}]\left[\frac{D}{\sqrt{dT}}\frac{Pd}{2t}\right]$$

where

$$\bar{\sigma} = a_0 + a_1\left(\frac{D}{d}\right)^{P_1}\left(\frac{D}{T}\right)^{P_2}\left(\frac{t}{T}\right)^{P_3} + a_2\left(\frac{D}{d}\right)^{P_4}\left(\frac{D}{T}\right)^{P_5}\left(\frac{t}{T}\right)^{P_6}$$

Design Equations

Maximum membrane stress intensity on vessel:

$$\sigma_v = [0.5315 - 0.06342\,(D/d)^{1.25}\,(D/T)^{-.25} \cdot (t/T)^{-.75} + 0.4372\,(D/d)(D/T)^{-.25}(t/T)^{-.25}] \cdot [(d/\sqrt{Dt}) \times (PD/2T)] \quad (1)$$

Maximum surface stress intensity on vessel

$$\sigma_v = [1.0048 - 0.01427\,(t/T)^{-1.5} + 0.8605\,(D/d)^{1.25}\,(D/T)^{-0.5}\,(t/T)^{0.5}] \cdot [(d/\sqrt{Dt}) \times (PD/2T)] \quad (2)$$

Maximum membrane stress intensity on nozzle

$$\sigma_n = [0.2728 - .04706\,(D/d)^{0.25}\,(t/T)^{-0.50} + 0.9551 \cdot (D/T)^{-0.25}\,(t/T)^{0.50}][(D/\sqrt{dT}) \times (Pd/2t)] \quad (3)$$

Maximum surface stress intensity on nozzle

$$\sigma_n = [0.3377 - 0.5272\,(D/d)^{-.50}\,(t/T)^{-.75} + 1.4229\,(D/d)^{-.50}\,(t/T)^{-.25}][(D/\sqrt{dT}) \times (Pd/2t)] \quad (4)$$

WRC 368 Example

D	=	Vessel Mean Diameter	=	121.25″
d	=	Nozzle Mean Diameter	=	19.75″
T	=	Vessel Thickness	=	1.0″
t	=	Nozzle Thickness	=	.25″

Design Pressure = 100 PSIG

Vessel

Membrane Stress	=	25075 PSI
Membrane + Bending Stress	=	27580 PSI

Nozzle

Membrane Stress	=	28940 PSI
Membrane + Bending Stress	=	59060 PSI

Note: Since the shears are small, the Stress Intensity and maximum principal stresses are approximately equal.

FLANGES WITH RING TYPE GASKETS

- Appendix 2 has rules for design of flanges with gaskets that are entirely within the bolt circle and no contact outside of this circle.

- Based on old Taylor Forge method and similar to Div. 2. (new rules being developed).

- Rules based on following concept:
    - provide adequate bolt load to seat gasket and preclude leak
    - provide adequate bolt load to resist internal pressure
    - using the greater of above, analyze flange

- See 2 - 3 for nomenclature. Hand calc. tedious. Use spreadsheet or Table similar to that attached. Tabulate all design info.

- Base all calcs on corroded condition. If the gasket is contained in a groove, the depth of groove must be added to required flange thickness.

- Keep dimension R to a min., to reduce moment arm hD.

- Place gasket close to bolt holes, to reduce bolt up moment.

- Generally, integral design is more economical than loose design for pressures greater than about 100 psi and for sizes up to 36".

- Determine diminution G, diameter at location of gasket reaction and b_o, basic gasket seating width. See Table 2 – 5.1

- Determine H_p, the force that must be exerted on the gasket to maintain a tight join =2b*3.14 GmP. Get m from Table 2 – 5.1.

- Calculate H, the total hydrostatic end load = $0.785\ G^2P$.

- Calculate W_{m2}, the force required to seat the gasket = 3.14 bGy. Get y from Table 2 – 5.1 .

- Calculate W_{m1} ,the required bolt load for operating condition = H $+H_p$.

- Determine the number and size of bolts, based on the larger of W_{m1} and W_{m2}.

- Bolt spacing should be sufficient to maintain tightness between bolt holes.

-- Calculate bolt areas at the root of thread.

- Bolt spacing to allow for wrench clearance.

- For raised face flanges, check for gasket crushing.

- Initially assume g_1 = 1 to 1.5 times g_0.

- The min. bolt circle diameter is $B+2g_1+2R$, unless more needed to accommodate bolts.

- Flange O.D. A=C+2E, E being dependent on bolt size.

- Calculate the moment for design condition $M_{01} = M_d + M_G + M_T$

- Calculate the moment for bolt up $M_{02} = W(C-G)/2$, Where, $W = (A_m + A_b)\ S_a/2$. The average of required area and actual area is used to protect against overloading by wrenching.

- Assume values of g_1 and t.

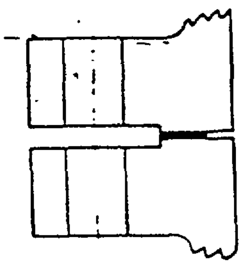
- Calculate flange moments.

- Calculate flange stresses. Compare with allowables and adjust design, if needed.

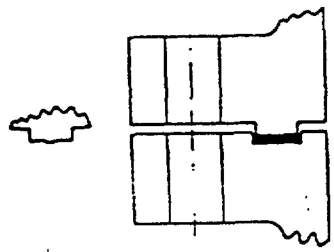
Face and Gasket

Two general categories of seals. First category seals by bolt force sqeezing gasket and includes raised face, tongue and groove, ring type joint, male and female, and lap joint. Gasket materials for this category include sheet stocks of rubber , cork, asbestos, spiral wound metallic elastomers, and solid metal rings. The second category is called self energizing or pressure actuated. The initial sealing is achieved by the gasket-facing geometry without significant bolt load, and the gasket seating force increases with pressure. O-rings of elastomer materials most widely used. Metallic self-energizing seals such as delta, ring double cone, lens joint, and metallic o-rings are also used.

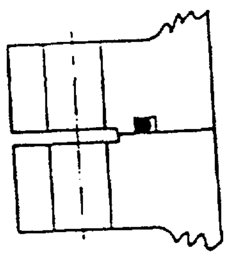
These require very fine surface finishes, tight tolerances, and care in assembly.



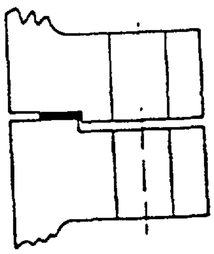
RAISED FACE



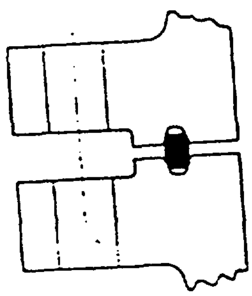
TONGUE & GROOVE



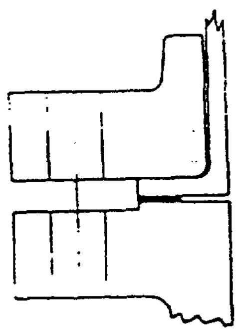
O RING



MALE & FEMALE



RING TYPE JOINT



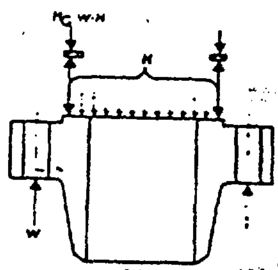
LAP JOINT

FACTORS m,y,b

Three factors are needed for design:

m = factor needed to hold a seal under internal pressure, dimensionless.
y = unit seating load, psi
b = effective gasket width, in.

The Code m and y values are old and based on judgement and experience. Code committee working on new rules and new factors. Will include a leak tightness factor.



Equilibrium equation:

$$W_{m1} = \pi/4\, G^2 P + \pi\, 2b\, G m p$$

W_{m1} is the operating bolt load.

G is the effective gasket diameter.

P is the internal pressure.

The first term on the right is the hydrostatic pressure load acting on the effective gasket diameter, and the second is the gasket reaction over an annular area 2b wide on the same G. sealing theory requires that the sealing load be resisted by the gasket when pressure equals P. But the gasket also resists the total load. When pressure P is applied, the hydrostatic portion of the total load is removed and the sealing load remains as the theoretical load required to hold a tight joint.

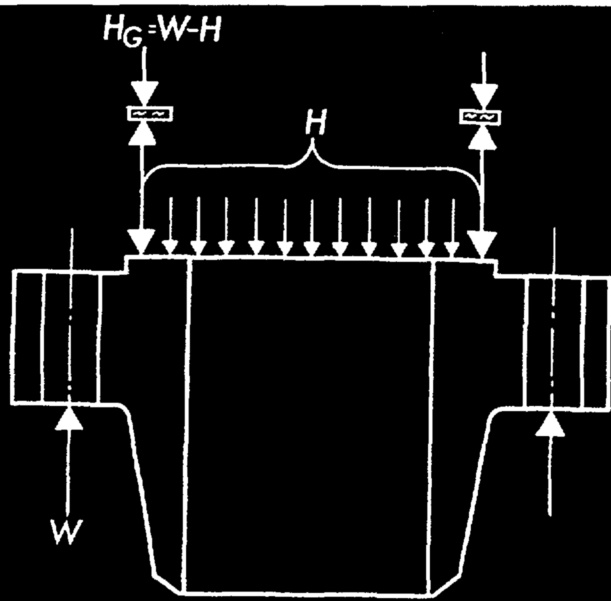


Fig. 4

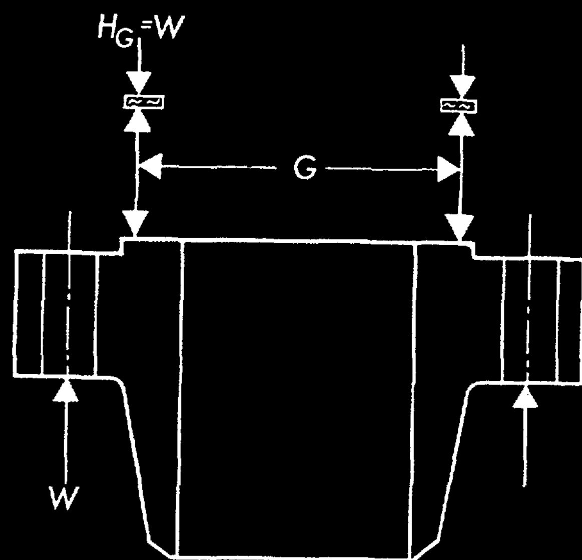


Fig. 5

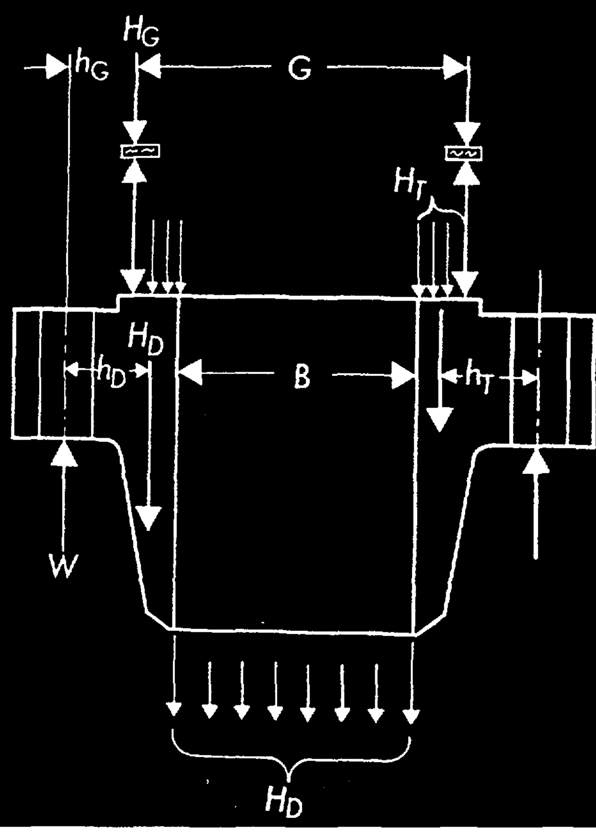


Fig. 6

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

SHORT COURSE ON

ASME BOILER & PRESSURE VESSEL CODE

SECTION VIII, DIVISION 1
PRESSURE VESSELS

EXAMPLE PROBLEMS

NOTICE

All statements and text given by the author represent his opinion alone and do not necessarily represent the position of the American Society of Mechanical Engineers nor the ASME Boiler and Pressure Vessel Committee.

For official Interpretations of the ASME Code, write, phone, or FAX the Secretary of the ASME Boiler & Pressure Vessel Committee

Section VIII, Division 1

Pressure Vessels

Example No. 1

Problem

Determine the required thickness of a cylindrical shell and hemispherical heads of a horizontal pressure vessel which contains a fluid during operation at 72 lbs./ft.^{3} The internal design pressure is 45 psi at a design temperature of 650°F. There is no corrosion allowance. The seamless heads are welded to a cylindrical shell which contains a longitudinal butt weld joint. All butt welds are assumed to be Type (1) with full RT (E = 1.0). The shell is 10 ft. inside diameter and 30 ft. long, tan-to-tan. The material is SA-515 Gr. 60. Also, determine the required thickness of the same vessel with an external design pressure of 15 psi (full vacuum) without stiffening rings and with stiffening rings which are close-spaced to permit use of the shell thickness for internal pressure.

Solution

(1) From Table 1A of II-D, for SA-515 Gr. 60 at 650°F, the allowable stress value is 15.8 ksi and the external pressure chart is Fig. CS-2 of Subpart 3 of II-D.

(2) Determine the fluid head during operation:
For contents: (10)(72)/(144) = 5.0 psi
Therefore, design the vessel for 45 + 5 = 50 psi

(3) For internal pressure of the cylinder using UG-27(c)(1) with E = 1.0, minimum required thickness of the shell is:

$t_r = (PR)/(SE\text{-}0.6P)$

$t_r = [(50)(60)]/[(15800)(1.0)\text{-}0.6(50)] = 0.190$ in.

Check for applicability of using UG-27(c)(1) or 1-2:

Is $t \leq R/2$? $0.190 < 30$

Is $P \leq 0.385SE$? $50 < .385(15800)(1) = 6{,}080$

(4) For internal pressure of the hemispherical head using UG-32(f), minimum required thickness of the head is:

$t_h = (PL)/(2SE\text{-}0.2P)$

$t_h = [(50)(60)]/[2(15800)(1.0)\text{-}0.2(50)] = 0.095$ in.

Check for applicability of using UG-32(f) or 1-3:

Is $t \leq 0.365L$? $0.095 < .365(60) = 21.4$

Is $P \leq 0.665SE$? $50 < .665(15800)(1) = 10{,}500$

(5) For external pressure on the cylinder, use UG-28 and Part 3 of II-D with E = 1.0 for butt joints in compression:

Determine the effective length without stiffening rings:

L = 1/3 each head depth + straight

$L = 2(1/3)(60) + 360 = 400$ in.

Assume: $t = 0.190$ in. (for internal pressure)

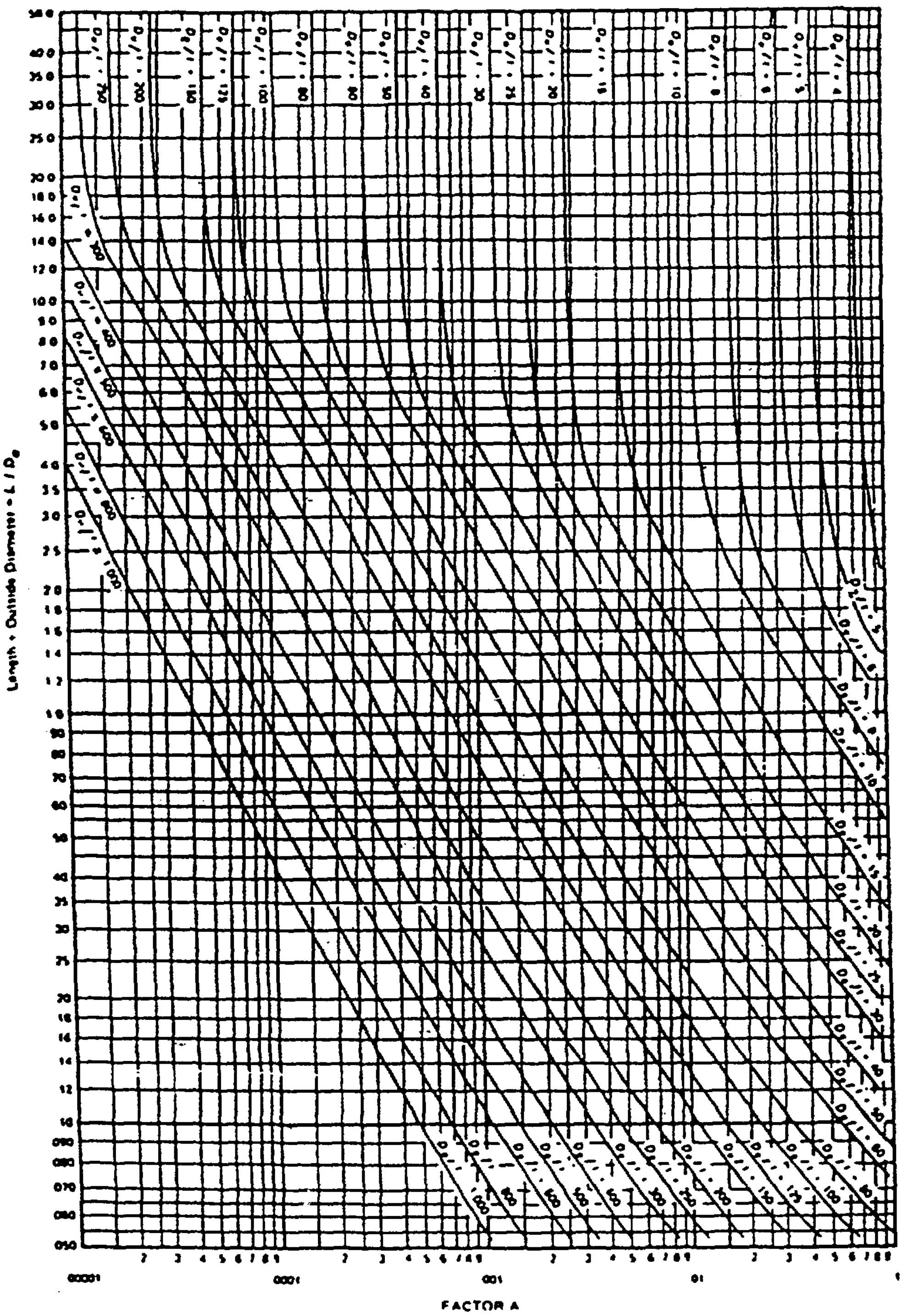
$D_o = 120 + 2(.19) = 120.38$ in.

Then: $L/D_o = 400/120.38 = 3.32$

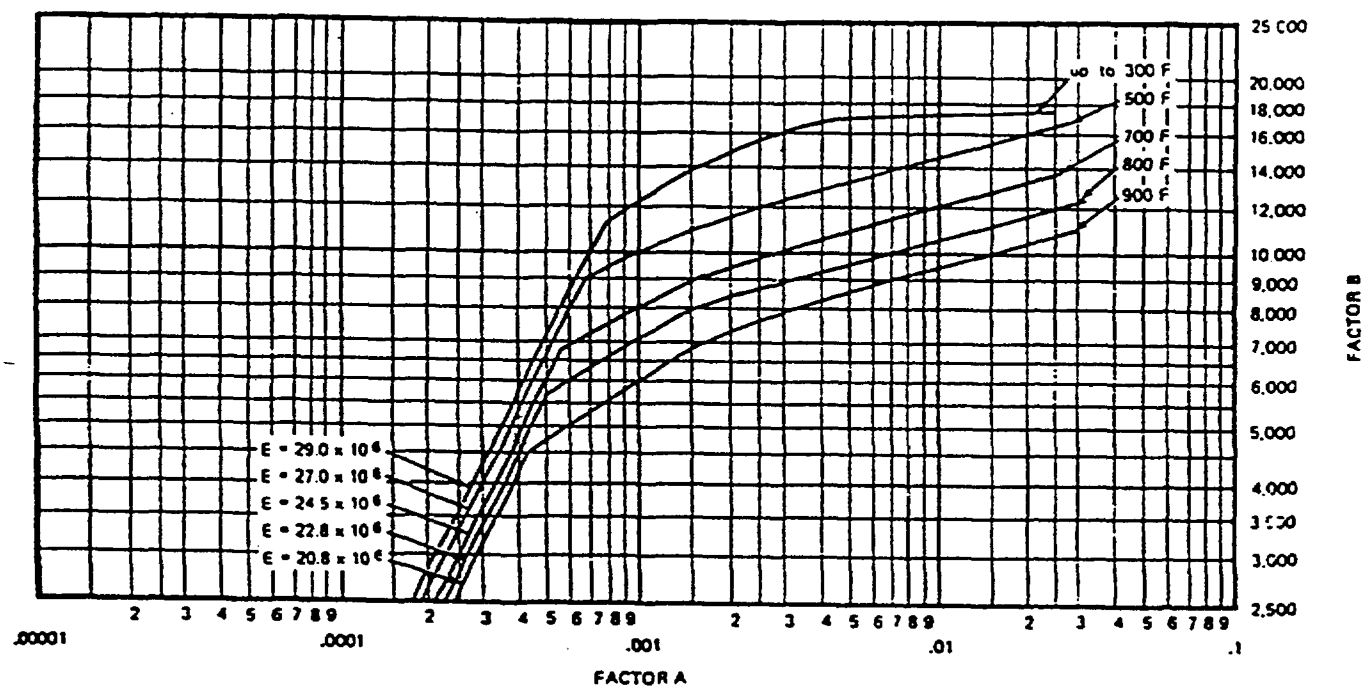
$D_o/t = 120.38/0.19 = 634$

(a) Enter Fig. G of II-D with $L/D_o = 3.32$ and read across to the sloping line of $D_o/t = 634$. Read down to *Factor A* = 0.000024

Geometry chart for vessels under external pressure



Materials chart for vessels under external pressure - carbon and low-alloy steels
(Y.S. ≥ 30 ksi)



(b) Enter Fig. CS-2 with *Factor A* = 0.000024, which is off to the left side and can not be read. Follow Step (7) of UG-28(c) and use equation as follows:

$P_a = (2AE)/[3(D_o/t)]$

$P_a = [2(.000024)(25.125\text{x}10^6)]/[(3)(634)]$

$P_a = 0.6$ psi $<$ 15 psi → Must increase thickness!

Assume: $t = 5/8$ in. $= 0.625$ in.

$D_o = 120 + 2(.625) = 121.25$ in.

Then: $L/D_o = 400/121.25 = 3.30$

$D_o/t = 121.25/.625 = 194$

(a) From Fig. G, *Factor A* = 0.00014

(b) Recalculating P_a,

$P_a = [2(.00014)(25.125\text{x}10^6)]/(3)(194)$

$P_a = 12.1$ psi $<$ 15 psi → Must increase thickness!

Assume: $t = 11/16$ in. $= 0.6875$ in.

$D_o = 120 + 2(.6875) = 121.375$ in.

$L/D_o = 400/121.375 = 3.30$

$D_o/t = 121.375/.6875 = 177$

(a) From Fig. G, *Factor A* = 0.00017

(b) Recalculating P_a,

$P_a = [2(.00017)(25.125\text{x}10^6)]/(3)(177)$

$P_a = 16.1$ psi $>$ 15 psi → $t = 11/16$ in. o.k.

Further calculations show $t_{min} = 0.671$ in. for 15 psi

(6) For external pressure on the hemispherical head, use UG-33(c), UG-28(d), and Subpart 3 of II-D. First assume, a thickness for internal pressure of $t = 0.095$ in.

Assume: $t = 0.095$in. & $R_o = .5(120 + 2x.095) = 60.095$ in.

(a) Calculate *Factor A*:

$Factor\ A = 0.125/(R_o/t) = 0.125/(60.095/.095) =$ 0.00020

(b) With $Factor\ A = 0.00020$, $Factor\ B = 2{,}450$

(c) Determine P_a:

$P_a = B/(R_o/t) = 2450/(60.095/.095)$

$P_a = 3.9$ psi < 15 psi $\rightarrow$ Must increase thickness!

Assume: $t = 0.190$ in. & $R_o = .5(120 + 2x.19) = 60.19$ in.

(a) Calculate *Factor A*:

$Factor\ A = 0.125/(60.19/.19) = 0.00039$

(b) Which gives $Factor\ B = 4{,}800$

(c) Determine P_a:

$P_a = 4800/(60.19/.19)$

$P_a = 15.1$ psi > 15 psi $\rightarrow t = 0.200$ in. o.k.

Further calculations show $t_{min} = 0.189$ in. for 15 psi

Of interest is the fact that for 45 psi internal pressure, the minimum required thickness of the cylinder is 0.190 in. while for 15.0 psi external pressure, the minimum required thickness is 0.671 in. For the head, the minimum required thickness for internal pressure is only 0.095 in. while for external pressure of 15.0 psi, the minimum required thickness is 0.189 in.

If a lesser thickness than 0.671 in. is desired for the cylinder, stiffening rings are required on the cylinder to obtain a smaller value of L to use in the calculation of P_a.

By "trial-and-error," the maximum stiffening ring spacing with the minimum thickness required for internal pressure of 0.190 in. is L = 19.25 in. as follows:

Assume: $t = 0.190$ in.

$L/D_o = 19.25/120.38 = 0.16$

$D_o/t = 120.38/.19 = 634$

(a) Enter Fig. G, *Factor A* = 0.00057

(b) Determine P_a:

$P_a = [2(.00057)(25.125\text{x}10^6)/(3)(634)]$

$P_a = 15.1 \text{ psi} > 15 \text{ psi} \rightarrow t = 0.190$ in. o.k.

This indicates that the optimum design would be one where the shell was thickened above 0.190 in. with stiffening rings being placed at a spacing larger than 19.25 in. c-c. The optimum design would be obtained by "trial-and-error" and consider the longitudinal tension stress when the support details are set. After the **best** thickness and stiffening ring spacing is determined, the design of the stiffening ring shall be developed according to UG-29.

Example No. 2

Problem

For the vessel in Example No. 1, all butt weld joint efficiencies were assumed to be E = 1.0. What degree of inspection is required at the butt weld joints for internal pressure if the thicknesses are set by the thickness required for the external pressure without stiffening rings? Using reduced butt joint efficiencies for lesser NDE requirements if applicable, calculate thicknesses and stiffening ring requirements for internal/external pressure. No stress from support considered.

Solution

(1) Determine the required longitudinal butt joint efficiency for internal pressure in the cylindrical shell with t = 0.688 in. by solving UG-27(c)(1) in terms of E by rearranging terms as follows:

$$E = [P/(S)(t)](R + 0.6t)$$

$$E = [(50)/(15800)(.688)][60 + .6(.688)] = 0.27$$

(2) Determine the required butt joint efficiency for internal pressure in the hemispherical heads with t_h = 0.25 in. based on the head-to-shell butt joint by solving UG-32(f) in terms of E by rearranging terms as follows:

$$E = [P/2(S)(t)](R + 0.2t)$$

$$E = [(50)/2(15800)(.25)][60 + .2(.25)] = 0.38$$

(3) Determine the required circumferential butt joint efficiency for internal pressure in the cylindrical shell with t = 0.688 in. by solving UG-27(c)(2) in terms of E by rearranging terms as follows:

$E = [P/2(S)(t)](R - 0.4t)$

$E = [(50)/2(15800)(.688)][60 - .4(.688)] = 0.14$

(4) Based on (1), (2), and (3), the assumed thicknesses for external pressure are acceptable for a butt weld joint efficiency of E = 0.27 for the cylindrical shell based on the circumferential stress; E = 0.38 for the hemispherical head membrane stress; and E = 0.14 for the cylindrical shell based on the longitudinal stress. From Table UW-12, Col. (C), a Type (1) butt weld joint efficiency with visual only is E = 0.70 and a Type (2) butt weld joint efficiency with visual only is E = 0.65. Consequently, with the assumed thicknesses, **visual examination only of the butt joints is required** for the internal design pressure of 45 psi.

(5) Based on visual examination only, determine the minimum required thicknesses for internal pressure. Place stiffening rings at a spacing which makes these thicknesses acceptable for an external pressure of 15 psi.

(a) Determine the minimum required thickness of the cylindrical shell using UG-27(c)(1) with E = 0.70:

$t_r = (PR)/(SE\text{-}0.6P)$

$t_r = [(50)(60)]/[(15800)(.7)\text{-}0.6(50)]$

$t_r = 0.272$ in.→ Use 5/16 in.

(b) Determine the adequacy of an arrangement with six stiffening rings spaced at 400/7 = 57.1 in., a shell thickness of 5/16 in., and an external pressure of 15 psi:

$D_o = 120 + 2(5/16) = 120.625$ in.

$L = 57.1$ in.

$t = 0.3125$ in.

$L/D_o = 57.1/120.625 = 0.473$

$D_o/t = 120.625/.3125 = 386$

From the chart, *Factor A* = 0.00040 and *Factor B* = 4,700

$P = 4B/3(D_o/t) = [4(4700)] \div [3(386)]$

$P = 16.2$ psi > 15.0 psi; $t = 5/16$ in. o.k.

This design is acceptable with 5/16 in. shell plate using six stiffening rings with an arrangement of approximate spacing of end rings at 37 in. from the head-to-shell joint along the shell and five intermediate spacings of 57 in.

(c) Determine the minimum required thickness of the hemispherical head using UG-32(f) with E = 0.65:

$t_h = (PL)/(2SE\text{-}0.2P)$

$t_h = [(50)(60)]/[2(15800)(.65)\text{-}0.2(50)]$

$t_h = 0.146$ in.

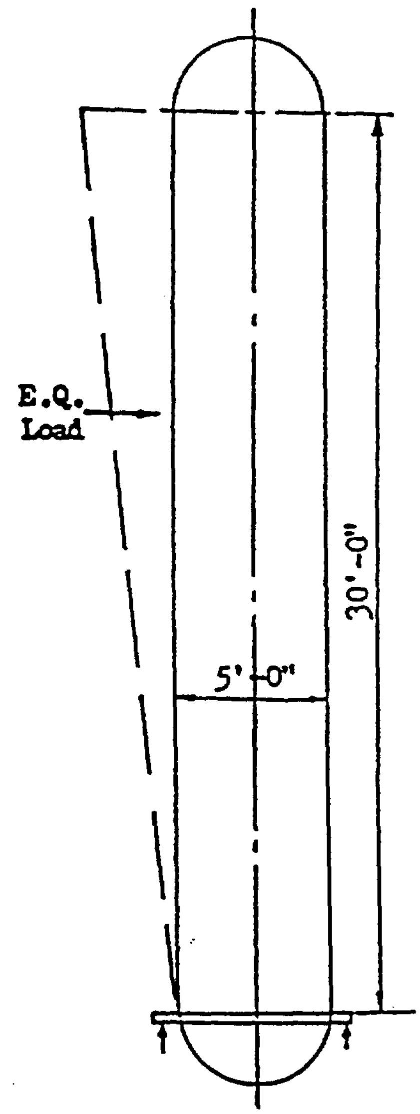
(d) From Example No. 1, for 15 psi external pressure, $t_{min.} = 0.189$ in.

This design is acceptable with 1/4 in. head thickness.

Example No. 3

Problem

Determine the required thickness of a cylindrical shell and hemispherical heads of a pressure vessel with internal design pressure of 225 psi at a design temperature of 650°F. There is no corrosion allowance and E = 1.0 for all butt weld joints. Material is SA-515 Gr. 60. The shell is 5 ft.-0 in. I.D. and 30 ft.-0 in. tan-to-tan. The vessel is installed vertically and supported at the lower head-to-shell tangent line. It is designed for 0.1g horizontal E.Q. force. When the d/t is constant, force is applied at the centroid of an inverse triangle. The contents weighs 35 lbs./ft.^{3} and vessel will be hydrostatically tested in the vertical position, if possible.



Solution

(1) From Table 1A of II-D for SA-515 Gr. 60 at 650°F, the allowable stress value is 15.8 ksi and the external pressure chart is Fig. CS-2 of Subpart 3 of II-D. When earthquake is considered in addition to the sustained

loadings, according to UG-23(d) the allowable stress for the total calculated stress may be increased to 1.2 S_a.

(2) Determine fluid head during operation and during hydrostatic test:

(a) At lower tangent line, H = 30 + 2.5 = 32.5 ft.

For contents: (32.5)(35)/(144) = 7.9 psi

For hydro. test: (32.5)(62.4)/(144) = 14.1 psi

(b) At bottom of lower head, H = 30 + 5 = 35 ft.

For contents: (35)(35)/(144) = 8.5 psi

For hydro. test: (35)(62.4)/(144) = 15.2 psi

(3) For internal pressure plus fluid head at the lower tangent line of the cylindrical shell using UG-27(c)(1) with E = 1.0, the minimum required thickness of the shell is:

$t_r = (PR)/(SE\text{-}0.6P)$

$t_r = [(225 + 7.9)(30)] \div$

$[(15800)(1.0)\text{-}0.6(225 + 7.9)]$

$t_r = 0.446$ in.

(4) For internal pressure plus fluid head at the center of the lower hemispherical head using UG-32(f), the minimum required thickness of the head is:

$t_h = (PL)/(2SE\text{-}0.2P)$

$t_h = [(225 + 8.5)(30)] \div$

$[2(15800)(1.0)\text{-}0.2(225 + 8.5)]$

$t_h = 0.222$ in.

(5) Assume t = 0.50 in. for shell and heads

(6) Determine the maximum allowable axial compressive stress on the cylinder according to UG-23(b) assuming $t = 0.5$ in.:

Factor A = $0.125/(R_o/t)$ = 0.125/[(30.5)(.5) = 0.0020 and from Fig. CS-2, *Factor B* = 10,000 psi, the allowable axial compressive stress. When EQ is considered, the allowable stress is 1.2 S_a = 1.2(10000). = 12,000 psi.

(7) Determine the weight of vessel and contents:

Metal Wt.

Shell = $\pi[(30.5)^2-(30)^2](360)(490/1728)$ = 9,700 lbs.

Heads = $(4/3)\pi[(30.5)^3-(30)^3](490/1728)$ = 1,630 lbs.

Total metal wt. = 11,330 lbs.

Fluid Wt.

Shell = $\pi(30)^2(360)(35/1728)$ = 20,620 lbs.

Heads = $(4/3)\pi(30)^3(35/1728)$ = 2,290 lbs.

Total fluid wt. = 22,910 lbs.

(8) Determine the earthquake moment at support line:

$$M_E = 0.1(11330+22910)(2/3)(360)$$

$$M_E = 821{,}760 \text{ in.-lbs.}$$

(9) Stress due to internal pressure following UG-27(c)(2):

$$S = (P/2E)[(R/t)-0.4]$$

$$S = [(225)/(2)(1)][(30/.5)-.4] = 6{,}710 \text{ psi}$$

(10) Stress due to dead load of shell and upper head:

$$S = W/\pi D_m t$$

$$S = (9700+815)/\pi(60.5)(.5) = 110 \text{ psi}$$

(11) Stress due to dead load of lower head and vessel contents:

$S = W/\pi D_m t$

$S = (815+22910)/\pi(60.5)(.5) = 250$ psi

(12) Stress due to earthquake moment, M_E:

$S = M_E/\pi R_m^2 t$

$S = (821760)/\pi(30.25)^2(.5) = 570$ psi

(13) Total stress above support with I.P. and D.L.:

$S_T = +6710 - 110$

$S_T =$ 6,600 psi tension < 15,800 psi allow.

(14) Total stress above support with I.P., D.L., & +E.Q.:

$S_T = +6710 - 110 + 570$

$S_T =$ 7,170 psi tension < 18,960 psi allow.

(15) Total stress above support with D.L. & -E.Q. and no internal pressure load:

$S_T = -110 - 570$

$S_T =$ 680 psi compression < 12,000 psi allow.

(16) Total stress above support with I.P., D.L., & -E.Q.:

$S_T = -570 - 110 + 6710$

$S_T =$ 6,030 psi tension < 18,960 psi allow.

(17) Total stress below the support with I.P. and D.L.:

$S_T = +6710 + 250$

$S_T =$ 6,960 psi tension < 15,800 psi allow.

In all cases, the total stress is less than the allowable stress; and, consequently, the circumferential stress controlled and set the thickness. Local stresses at the supports shall be considered separately and may require an increase in shell thickness in the local region.

(18) The minimum required thickness of the cylindrical shell based on the hydrostatic test pressure of (1.3P + fluid head) and with an allowable stress of yield strength* at test temperature of 32,000 psi is:

$$t_r = (PR)/(SE - 0.6P)$$

$$t_r = \{[(1.3)(225) + 14.1](30)\} \div \{(32000)(1) - 0.6[(1.3)(225) + 14.1]\}$$

$$t_r = 0.289 \text{ in.} < 0.5 \text{ in. ordered thickness.}$$

Since minimum required thickness of shell based on hydro. test is less than ordered thickness, shell is satisfactory for hydro. test in vertical position.

(19) The minimum required thickness of the lower head based on the hydro. test pressure of (1.3P + fluid head) and yield strength* of 32,000 psi using UG-32(f) is:

$$t_h = (PL)/(2SE - 0.2P)$$

$$t_h = \{[(1.3)(225) + 15.2](30) \div \{2(32000)(1) - 0.2[(1.3)(225) + 15.2)]\}$$

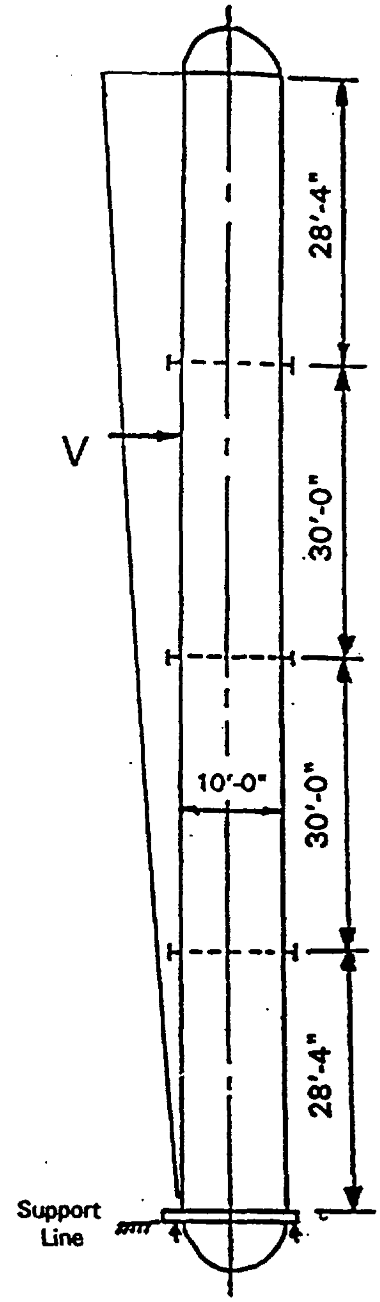
$$t_h = 0.144 \text{ in.} < 0.50 \text{ in. ordered thickness}$$

Since minimum required thickness of lower head based on hydro. test is less than ordered thickness, lower head is satisfactory for hydro. test in vertical position.

* Yield strength is assumed based on "no visible permanent distortion" being permitted as given in UG-99(d).

Example No. 4

Problem



Determine the required thickness of a cylindrical shell and hemispherical heads of a pressure vessel with external design pressure of 15 psi (full vacuum) at a design temperature of 500°F. There is no corrosion allowance. Material is SA-516 Gr. 60. The shell is 10 ft.-0 in. inside diameter and 116 ft.-8 in. tan-to-tan. The vessel will be installed vertically and supported at the lower head-to-shell tangent line. It is designed for a horizontal earthquake force according to the ASCE 7-95 Standard applied at the centroid of an inverse triangle. The vessel will contain fluid at 50 lbs./ft.^{3} and will be hydrostatically tested in the vertical position, if possible. Three stiffening rings are used with the two end ones spaced at 28 ft.-4 in. from each head-to-shell tangent line and the third at the center with 30 ft.-0 in. between each end and the center ring and between the 1/3-head depth line and each end ring.

Solution

(1) The external pressure chart is Fig. CS-2 of Subpart 3 of II-D. When earthquake is considered in addition to the sustained loadings, the allowable stress for total calculated stress may be increased to $1.2S_a$.

(2) VIII-1, does not contain rules for the design of vessels with combined external pressure and axial compressive loading which are applied at the same time (beam column); consequently, U-2(g) applies. For this example problem, a method given in ASME Paper No. 54-A-104, entitled *The Design of Vertical Pressure Vessels Subjected to Applied Forces by Dr. E.O. Bergman*, is used. Any method which satisfies U-2(g) is acceptable.

(3) Determine a tentative shell thickness based on external pressure alone using UG-28(c)(1) making allowance for the fact that the load combination of external pressure and axial compression is more severe than the external pressure alone. The method of superposition of loads and stresses is not applicable. Assume $t = 0.625$ in., $D_o/t = 121.25/.625 = 194$, and $L/D_o = 360/121.25 = 2.97$. From Fig. G, *Factor A* = .00017. On Fig. CS-2, *Factor A* is off to the left. From UG-28(c)(1) Step 7:

$$P_a = [2AE]/[3(D_o/t)] = 2(.00017)(27\text{x}10^6)/3(194)$$

$$P_a = 15.77 \text{ psi.} \rightarrow \text{Continue to use } t = 0.625 \text{ in.}$$

(4) Determine the weight of vessel and contents:

Metal Wt.

Shell = $\pi[(60.625)^2-(60)^2](1400)(490/1728) = 94{,}030$ lbs.

Heads = $(4/3)\pi[(60.625)^3-(60)^3](490/1728)$ = <u>8,100</u> lbs.

Total metal wt. = 102,130 lbs.

<u>Fluid Wt.</u>

Shell = $\pi(60)^2(1400)(50/1728)$ = 458,150 lbs.

Heads = $(4/3)\pi(60)^3(50/1728)$ = <u>26,180</u> lbs.

Total fluid wt. = 484,330 lbs.

(5) Determine the earthquake loading:

Following 9.2.7 of the ASCE 7-95 Standard, the seismic base shear or horizontal force, V, is determined by:

$V = 0.6C_aW$ for $T < 0.06$ *sec.*

and by 9.2.3.2 for other values of T as follows:

$V = C_sW$, where $C_s = 1.2C_v/RT^{2/3} \leq 2.5C_a/R$

For this problem, assume that calculations of these equations gives $V = 0.1\ W$ applied at centroid of inverse triangle, where W equals the weight of the vessel.

$V = 0.1(102130 + 484330) = 58{,}650$ lb.

(6) Determine the earthquake moment at support line:

$M_E = (58650)(2/3)(1400)$

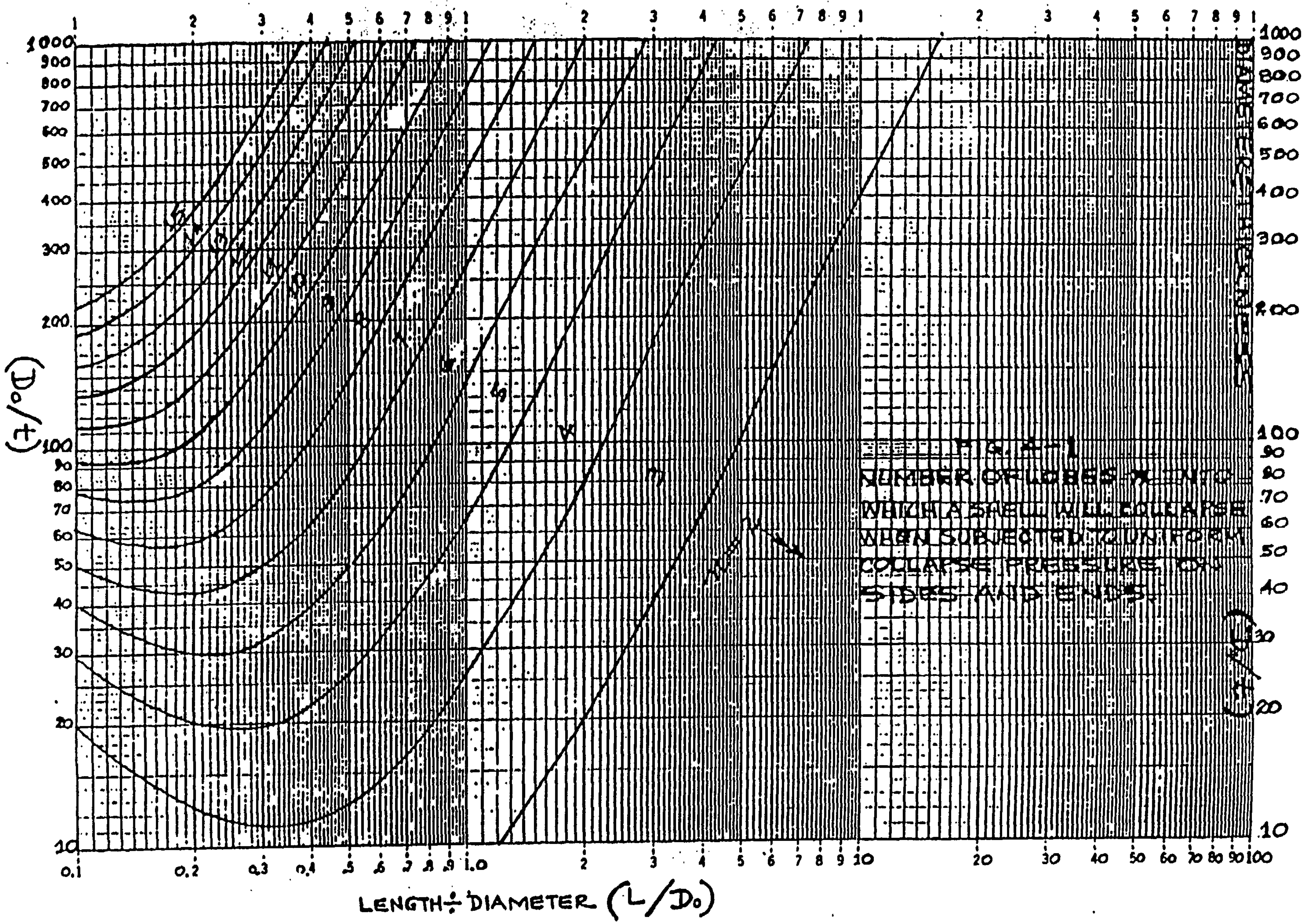
$M_E = 54{,}740{,}000$ in.-lbs.

(7) Determine the maximum allowable axial compressive stress on the cylinder according to UG-23(b) assuming $t = 0.625"$

A $= 0.125/(R_o/t) = 0.125/97 = 0.0013$

and from Fig. CS-2, *Factor B* = 10,500 psi, the allowable axial compressive stress.

(8) Using the Bergman procedure, determine the maximum compressive loading for dead load only (lb./in.), without



LENGTH ÷ DIAMETER (L/Do)
(Do/t)

earthquake load:

$P = W/\pi D_m$

$P = (94030+4050)/\pi(120.625) = 258.8$ lb./in.

$P_o = 15.0$ psi

$a = P/P_oD_o = (258.8)/(15)(121.25) = 0.142$

$m = 1.23/(L/D_o)^2 = 1.23/(2.97)^2 = 0.139$

$n = 3.0$ from Fig. 4-1

$P_o' = [(n^2 - 1 + m + am)/(n^2 - 1 + m)]P_o$

$P_o' = \{[(3)^2\text{-}1+.139+(.142)(.139)] \div$
$[(3)^2\text{-}1+.139]\}(15)$

$P_o' = 15.03$ psi $<$ 15.77 psi, o.k.

(9) Maximum compressive loading for dead load plus EQ load:

$P = W/\pi D_m + 4M_E/\pi D_m^2$

$P = 258.8 + 4(54740000)/\pi(120.625)^2$

$P = 258.8 + 4790.1 = 5{,}048.9$ lbs./in.

$P_o = 15.0$ psi

$a = P/P_oD_o = (5048.9)/(15)(121.25) = 2.776$

$m = 0.139$ and $n = 3$ as determined in (8), above

$P_o' = (15.0)\{[(3)^2\text{-}1+.139+(2.776)(.139)] \div$
$[(3)^2\text{-}1+.139]\}$

$P_o' = 15.7$ psi $<$ 15.77 psi, o.k.

(10) Since $a > 1.0$, it is necessary to check the vessel as a cantilever beam supported at the lower end.

(a) Axial unit stress due to P_o is $P_oD_o/4$

$= (15.7)(120.625)/4 = 473.5$ lb./in.

(b) Compressive stress due to P_o = unit stress/t

$= 473.5/.625 = 760$ psi

(c) Axial unit stress due to $P_o + EQ$

$= 473.5 + 5048.9 = 5522.4$ lb./in.

(d) Compressive stress due to $P_o + EQ$

$= 5522.4/.625 = 8{,}840$ psi

(e) Allowable stress with DL + EQ = 1.2(10500)

= 12,600 psi

Since 8,840 psi < 12,600 psi, compressive stress is o.k. with $t = 0.625$ in.

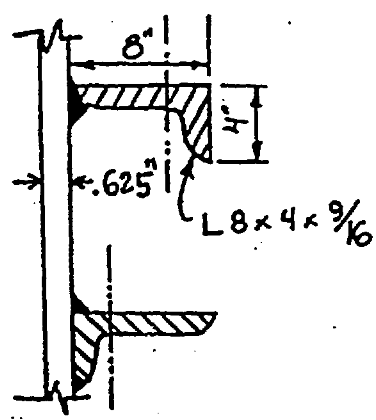
Example No. 5

Problem

Using the rules of UG-29, determine the stiffening ring sizes for the vessel in Example No. 4. Rings are at 500°F, the same temperature as shell, and material is SA-36. External pressure chart is same as shell, Fig. CS-2 of II-D. What size of stiffening rings are required when the ring alone is used? What size of stiffening ring-shell combination is required?

Solution

(1) For the stiffening ring alone:



(a) Assume an 8 in.x4 in.x9/16 in. angle with the 8 in. leg out.

A_s = 6.43 in.^{2}; I = 42.8 in.^{4};

D_o = 121.25 in.; L_s = 360 in.;

t = 0.625 in.; P_o' = 15.7 psi

(b) From UG-29(a) and Fig. CS-2:

$B = (3/4)[(P_o')(D_o)]/[t+(A_s/L_s)]$

$B = (3/4)[(15.7)(121.25)]/[.625+(6.43/360)] = 2220$

$A = 2B/E = 2(2220)/27.0\text{x}10^6 = 0.000165$

$I_s = \{D_o^2 L_s [t + (A_s/L_s)]A\}/14$

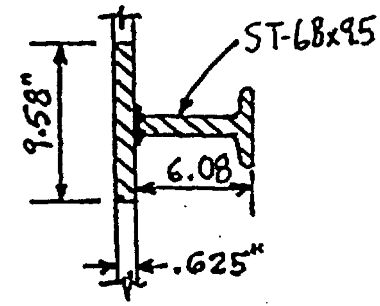
$I_s = \{(121.25)^2(360)[.625+(6.43/360)](.000165)\}/14$

I_s = 40.10 in.^{4} < 42.8 in.^{4} actual

Therefore, 8 in.x 4 in.x 9/16 in. angle is good!

(2) <u>For stiffening ring-shell combination:</u>

(a) Assume a ST-6Bx9.5 tee integrally attached to the shell wall so that the combination acts together. For the tee: A_s = 2.81 in.^{2}; I = 10.2 in.^{4}.



(b) Determine the length of shell acting with the tee where the effective length is $1.1(D_o t)^{1/2}$ = $1.1(121.25\text{x}.625)^{1/2}$ = 9.58 in. and 0.625 in. thick. The combined moment of inertia of the tee-shell combination is determined as:

	Area	Arm	Area x Arm
.625x9.58 =	5.99	6.39	38.28
	2.81	1.67	4.69
	8.80		42.97

y = 42.97/8.80 = 4.88 in.

I_{xx}	Area	Arm	Ad^2
0.19	5.99	1.51	13.66
10.20	2.81	3.21	28.95
10.39			42.61

$I' = I_{xx} + Ad^2 = 10.39 + 42.61 = 53.00$ in.^{4}

(c) From UG-29(a) and Fig. CS-2:

$B = (3/4)[(P_o')(D_o)]/[t+(A_s/L_s)]$

$B = (3/4)[(15.7)(121.25)]/[.625+(2.81/360)] = 2256$

$A = 2B/E = 2(2256)/27.0 \times 10^6 = 0.000167$

$I_s = \{D_o^2 L_s [t + (A_s L_s)]A\}/10.9$

$I_s = \{(121.25)^2(360)[.625 + (2.81/360)](.000167)\}$
$/(10.9)$

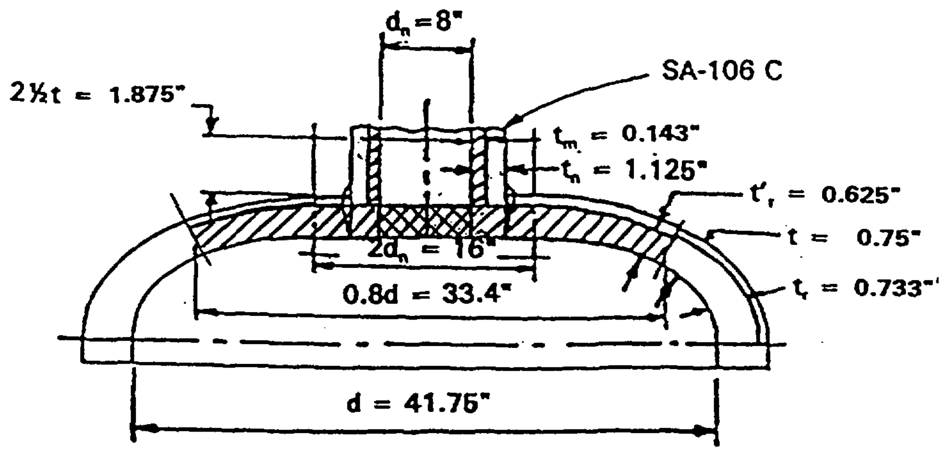
$I_s = 51.31 \text{ in.}^4 < 53.00 \text{ in.}^4$ actual

Therefore, ST-6Bx9.5 tee is good!

Example No. 6

Problem

Determine the reinforcement requirements of an 8 in. I.D. nozzle which is centrally located in a 2:1 ellipsoidal head. The nozzle is inserted through the head and attached by a full penetration weld. The inside diameter of the head skirt is 41.75 in. The head material is SA-516 Gr.70 and the nozzle material is SA-106 Gr.C. The internal design pressure is 700 psi at a design temperature of 400°F. There is no corrosion allowance and all butt weld joint efficiencies are E = 1.0.



Solution

(1) The allowable tensile stress for both SA-516 Gr.70 and SA-106 Gr.C at 400°F is 20.0 ksi. Therefore, $f_r = 20.0/20.0 = 1.0$.

(2) Using UG-32(d), minimum required thickness of 2:1

ellipsoidal head is:

t_r = $(PD)/(2SE - 0.2P)$

t_r = (700x41.75)/(2x20000x1.0 - 0.2x700)

t_r = 0.733 in.

Nominal thickness used is t = 0.75 in.

(3) According to Rule (3) of t_r in UG-37(a), when an opening and its reinforcement are in an ellipsoidal head and are located entirely within a circle the center of which coincides with the head and the diameter is equal to 80% of the shell diameter, t_r is the thickness required for a seamless sphere of radius K_1D, where D is the shell I.D. and K_1 is 0.9 from Table UG-37. For this head, the opening and its reinforcement shall be within a circle with a diameter of $0.8D$ = (.8)(41.75) = 33.4 in.

(4) Following (3), above, the radius $R = K_1D$ = 0.9(41.75) = 37.575 in. is used in UG-32(f) to determine the t_r for reinforcement calculations is:

t_r = $(PR)/(2SE - 0.2P)$

t_r = [(700)(37.575)]/[2(20000)(1.0)-.2(700)]

t_r = 0.625 in.

(5) Using UG-27(c)(1), minimum required thickness of nozzle:

t_{rn} = $(PR_n)/(SE - 0.6P)$

t_{rn} = [(700)(4)]/[(20000)(1.0)-.6(700)] = 0.143 in.

Nominal thickness used is t_n = 1.125 in.

Check UG-45: UG-45(a) is o.k. UG-45(b)(1) = 0.733 in.; UG-45(b)(4) = .365(.875) = 0.319 in.; & UG-45(b) is o.k.

(6) Limit parallel to head surface:

$X = d$ or $(0.5d+t+t_n)$, whichever is larger.

$X = 8$in. or$(4+.75+1.125=5.875$ in.),use $X = 8$ in.

(7) Limit perpendicular to head surface:

$Y = 2.5t$ or $2.5t_n$, whichever is smaller.

$Y = 2.5(.75) = 1.875$ or $2.5(1.125)=2.812$ in.,

use $Y = 1.875$ in.

(8) Limits of $2X = 2(8) = 16$ in. < 33.4 in., therefore, provision to use spherical head rule is valid.

(9) Reinforcement area required according to UG-37(c), is:

$$A_r = dt_r F + 2t_n t_r F(1 - f_r) = (8)(.625)(1) + 0$$

$$A_r = 5.00 \text{ in.}^2$$

(10) Reinforcement area available in head is:

$$A_1 = d(Et - Ft_r) - 2t_n(Et - Ft_r)(1 - f_r)$$

$$A_1 = d(t - t_r)\text{-}0 = (8)(.75\text{-}.625)\text{-}\ 0 = 1.000 \text{ in.}^2$$

(11) Reinforcement area available in nozzle is:

$$A_2 = 2Y(t_n - t_{rn}) = (2)(1.875)(1.125\text{-}.143)$$

$$A_2 = 3.682 \text{ in.}^2$$

(12) Reinforcement area available in fillet welds is:

$$A_4 = 2(.5)t_w^2 = 2(.5)(.75)^2 = 0.562 \text{ in.}^2$$

(13) Total reinforcement area available in head, nozzle, and welds is:

$$A_T = A_1+A_2+A_4 = 1.0+3.682+.562 = 5.244 \text{ in.}^2$$

Area available of 5.244 in.^{2} > area required of 5.00 in.^{2}

(14) Determination of weld strength and load paths:
UW-15(b) states that weld strength and load path calculations for pressure loading are not required for this nozzle which is like that shown in Fig. UW-16.1(c).

Example No. 7

Problem

Determine the reinforcement requirements for a 12 in. x 16 in. elliptical manway opening. The manway forging is inserted through the vessel wall and attached by a full penetration weld. The 12 in. dimension lies along the longitudinal axis of the vessel. The manway cover seals against the outside surface of the manway forging. The I.D. of the shell is 41.875 in. The shell material is SA-516 Gr.70 and the manway forging is SA-105. The internal design pressure is 700 psi at a design temperature of 400°F. There is no corrosion allowance and all butt weld joint efficiencies are E = 1.0.

Solution

(1) The allowable tensile stress for both SA-516 Gr.70 and SA-105 at 400°F is 20.0 ksi. Therefore, $f_r = 1.0$.

(2) Using UG-27(c)(1), minimum required thickness of shell is:

$$t_r = (PR)/(SE\text{-}0.6P)$$

$$t_r = [(700)(20.9375)]/[(20000)(1.0) -.6(700)]$$

$$t_r = 0.749 \text{ in.}$$

Nominal thickness used is t = 0.75 in.

(3) Since the manway forging is elliptical-shaped, there are no rules in VIII-1 for determining the minimum required thickness. Therefore, U-2(g) applies. For an elliptical

shell, the minimum required thickness is given below. The maximum value of minimum required thickness is used for calculations on all planes. This equation for minimum required thickness is:

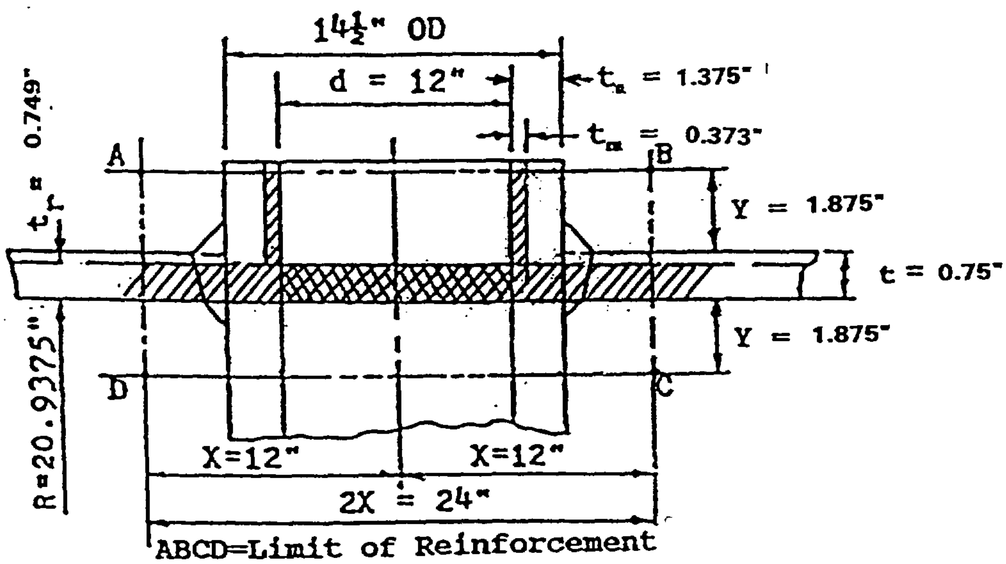
$$t_{rn} = P\,a^2\,b^2 \,/\, SE\,(a^2 \sin^2\Phi + b^2 \cos^2\Phi)^{3/2}$$

$$t_{rn} = [(700)(8)^2(6)^2]/\{(17500)(1.0)[(8)^2(0)^2 + (6)^2(1)^2]^{3/2}\} = 0.373 \text{ in.}$$

Nominal thickness used is $t_n = 1.375$ in.

UG-45(a) is o.k. UG-45(b) is not required.

(4) Examination of the longitudinal plane:



(a) Limit parallel to shell surface:

$X = d$ or $(0.5d + t + t_n)$, whichever is larger.

$X = 12$ or $(6 + .75 + 1.375 = 8.125)$, use $X = 12$ in.

(b) Limit perpendicular to shell surface:

$Y = 2.5t$ or $2.5t_n$ whichever is smaller.

$Y = 2.5(.75) = 1.875$ or $2.5(1.375) = 3.437$,

use $Y = 1.875$ in.

(c) Reinforcement area required following UG-37(c) is:

$A_r = dt_r F + 2t_n t_r F(1 - f_r) = (12)(.858)(1.0) + 0$

$A_r = 8.988$ in.^{2} when $f_r = 1.0$.

(d) Reinforcement area available in shell is:

$A_1 = d(E_1 t - Ft_r) - 2t_n(E_1 t - Ft_r)(1 - f_r)$

when $f_r = 1.0$, second term becomes zero; therefore,

$A_1 = (12)(.75 - .749) = 0.012$ in.^{2}

(e) Reinforcement area available in nozzle is:

Outward:

$A_2 = 2Y(t_n - t_{rn})$

$A_2 = 2(1.875)(1.375 - .373) = 3.758$ in.^{2}

Inward:

$A_3 = 2Y(t_n) = 2(1.875)(1.375) = 5.156$ in.^{2}

(f) Reinforcement area available in fillet welds is:

$A_4 = 2(.5)t_w^2 = 2(.5)(.75)^2 = 0.562$ in.^{2}

(g) Total reinforcement area available from shell, nozzle, and welds is:

$A_T = A_1 + A_2 + A_3 + A_4$

$A_T = 0.012 + 3.758 + 5.156 + .562 = 9.488$ in.^{2}

Area available of 9.488 in.^{2} > area required of 8.988 in.^{2}

(5) Examination of the circumferential plane:

The opening has a 16 in. dimension in this plane, but $F = 0.5$ and $f_r = 1.0$

(a) Reinforcement area required following UG-37(c) is:

$A_r = dt_r F = (16)(.749)(0.5) = 5.992 \text{ in.}^2$

(b) Total reinforcement area available from shell and nozzle without recalculating for 16 in. opening is:

$A_T = 9.488 \text{ in.}^2$

Area available of $9.488 \text{ in.}^2 >$ area required of 5.922 in.^2

Since A_T based on the longitudinal plane is larger than A_r without any consideration of an increase in the limit parallel to the shell surface, X, which is permitted due to the increase of d from 12 in. to 16 in., the design is satisfactory.

(6) Determination of weld strength and load paths:

UW-15(b) states that weld strength and load path calculations for pressure loading are not required for this nozzle which is like that shown in Fig. UW-16.1(c).

Example No. 8

Problem

Determine the reinforcement requirements for a 5.625 in. I.D. nozzle which is located on a *hillside* or non-radial position on the circumferential plane. The nozzle wall is abutting the vessel wall and attached by a full penetration weld. The I.D. of the shell is 41.875 in. The shell material is SA-516 Gr.70 and the nozzle material is SA-106 Gr.B. The internal design pressure is 700 psi at a design temperature of 400°F. There is no corrosion allowance and all butt weld joint efficiencies are E = 1.0.

Solution

(1) The allowable tensile stress @ 400°F for SA-516 Gr.70 is 20.0 ksi and for SA-106 Gr.B is 17.1 ksi.

f_r = 17.1/20.0 = 0.855

(2) Using UG-27(c)(1), minimum required thickness of shell is:

$$t_r = (PR)/(SE - 0.6P)$$

$$t_r = [(700)(20.9375)]/[(20000)(1.0) - .6(700)]$$

$$t_r = 0.749 \text{ in.}$$

Nominal thickness used is t = 0.75 in.

(3) Using UG-27(c)(1), minimum required thickness of nozzle is:

$$t_{rn} = (PR_n)/(SE - 0.6P)$$

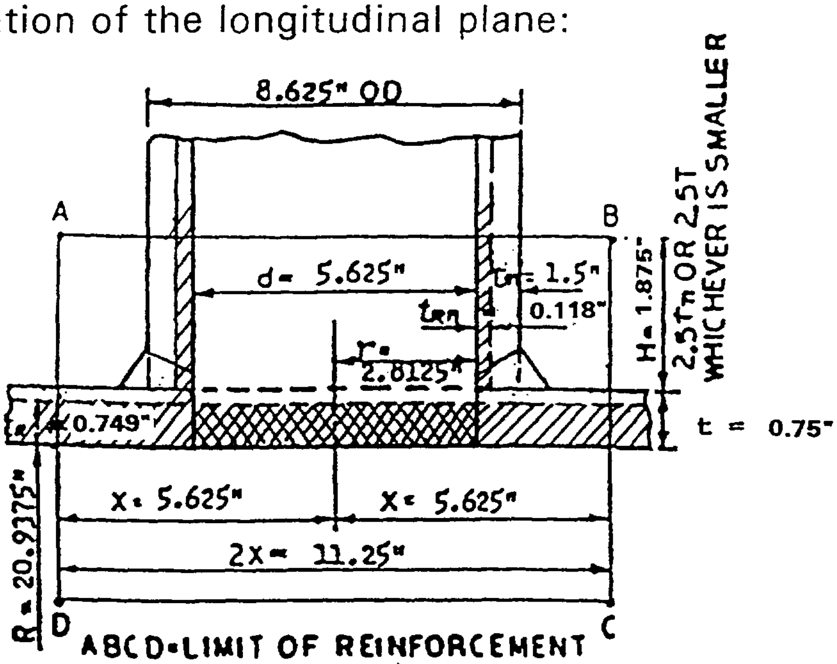
t_{rn} = [(700)(2.8125)]/[(17100)(1.0) - .6(700)]

t_{rn} = 0.118 in.

Nominal thickness used is t_n = 1.5 in.

Check UG-45: UG-45(a) is o.k. UG-45(b)(1) = 0.749 in.; UG-45(b)(4) = .322(.875) = 0.282 in.; & UG-45(b) is o.k.

(4) Examination of the longitudinal plane:



(a) Although allowable stresses of the shell and nozzle are different, since the nozzle weld is a full penetration weld abutting the shell wall, f_r = 1.0 for both A_r & A_1 and f_r = 0.855 for A_2.

(b) Limit parallel to shell surface:

$X = d$ or $(0.5d + t + t_n)$, whichever is larger.

X = 5.625 or (2.812 + .75 + 1.5 = 5.062),

use X = 5.625 in.

(c) Limit perpendicular to shell surface:

$Y = 2.5t$ or $2.5t_n$, whichever is smaller.

$Y = 2.5(.75) = 1.875$ in. or $2.5(1.5) = 3.75$ in., use $Y = 1.875$ in.

(d) Reinforcement area required following UG-37(c) is:

$A_r = dt_r F = (5.625)(.749)(1.0) = 4.213$ in.^{2}

(e) Reinforcement area available in shell is:

$A_1 = (2X - d)(t - t_r) = (11.25 - 5.625)(.75 - .749)$

$A_1 = 0.005$ in.^{2}

(f) Reinforcement area available in nozzle is:

$A_2 = 2Y(t_n - t_{rn})(f_r) = 2(1.875)(1.5 - .118)(.855)$

$A_2 = 4.431$ in.^{2}

(g) Reinforcement area available in fillet welds is:

$A_4 = 2(.5)t_w^2 = 2(.5)(.75)^2 = 0.562$ in.^{2}

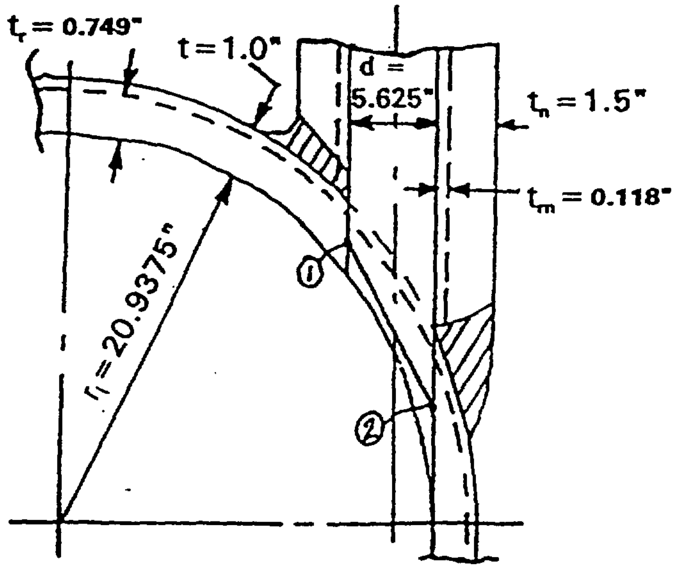
(h) Total reinforcement area available from shell, nozzle, and welds is:

$A_T = A_1 + A_2 + A_4 = 0.005 + 4.431 + .562 = 4.998$in.^{2}

Area available of 4.998 in.^{2} > area required of 4.213 in.^{2}

(5) Examination of circumferential plane:

(a) Since this is a non-radial plane, it is necessary to determine the chord length measured diagonally across the opening, chord length 1-2, based on the mid-point of the minimum required thickness of the shell, $t_r = 0.749$ in. Based upon the geometry, chord length 1-2 = 12.217 in. Therefore, $d' = 12.217$ in. and $F = 0.5$ on the circumferential plane.



(b) Reinforcement area required using the chord length 1-2 is:

$A_r' = d' t_r F = (12.217)(.749)(0.5) = 4.575 \text{ in.}^2$

(c) Total reinforcement area available from shell and head is:

$A_T = 4.998 \text{ in.}^2$ based on the longitudinal plane.

Area available of $4.998 \text{ in.}^2 >$ area required of 4.575 in.^2

Since A_T based on the longitudinal plane is larger than A_r' without any consideration of an increase in the limit parallel to the shell surface due to the increased d', the design is satisfactory.

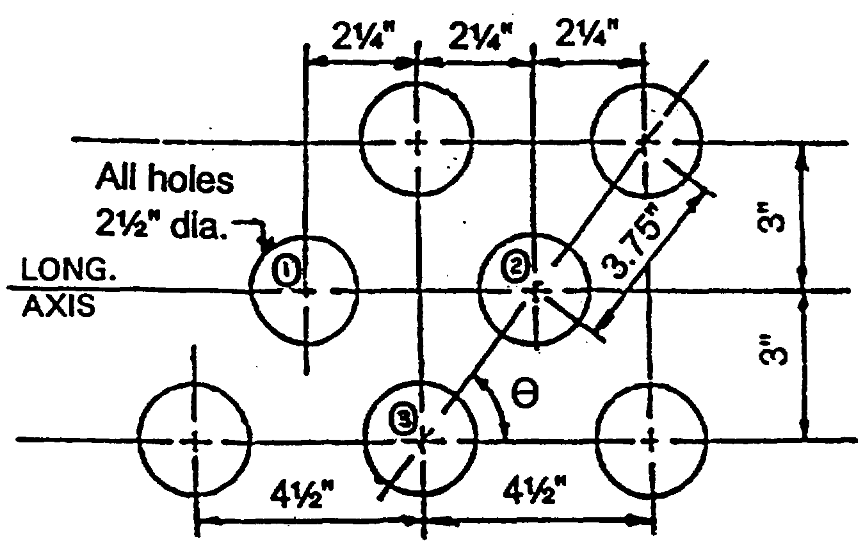
(6) Determination of weld strength and load paths:

UW-15(b) states that weld strength and load path calculations for pressure loading are not required for this nozzle which is like that shown in Fig. UW-16.1(a).

Example No. 9

Problem

Determine the minimum required thickness of a 36 in. I.D. cylindrical shell based upon reinforcement requirements given in UG-37 through UG-42 for a series of openings in a pattern. The openings are 2.50 in. diameter on a staggered pattern of three longitudinal rows on 3.0 in. circumferential spacing and 4.50 in. longitudinal spacing. The internal design pressure is 600 psi at a design temperature of 500°F. The shell material is SA-516 Gr.70 and the nozzle material is SA-210 Gr.C. There is no corrosion allowance. The openings are not located in or near any butt weld joint.



Solution

(1) The allowable tensile stress for both SA-516 Gr.70 and SA-210 Gr.C at 500°F is 20.0 ksi. Therefore, $f_r = 1.0$

(2) Using UG-27(c)(1), minimum required thickness of shell for reinforcement calculations assuming a seamless shell with E = 1.0 is:

t_r = *(PR)/(SE - 0.6P)*

t_r = [(600)(18)]/[(20000)(1.0) - .6(600)]

t_r = 0.550 in.

(3) For comparison with the ligament efficiency method, determine the reinforcement requirements based only on the shell thickness (without consideration of the nozzle thickness). Since the reinforcement area available comes only from the shell, the shell thickness will have to be increased. A trial thickness will be assumed and verified. Try t= $2t_r$= 2(.550) = 1.100 in.

Nominal thickness used is *t* = 1.25 in.

(4) Limit parallel to shell surface:

X = *d* or *(.5d + t)*, whichever is larger.

X = 2.50 or (1.25 + 1.25 = 2.50), use *X* = 2.50 in.

(5) Examine the longitudinal plane, 1-2:

(a) With an actual center-to-center spacing of 4.5 in., the reinforcing limits of *2X* = 2(2.5) = 5.0 in. exceeds the actual spacing of 4.5 in. Therefore, the reinforcement limits overlap and the rules of UG-42 apply. Those limits state that no reinforcement area shall be used more than once.

(b) Reinforcement area required is:

$A_r = dt_rF$ = (2.5)(.550)(1.0) = 1.375 in.^{2}

(c) Reinforcement area available in shell is:

A_1 = *(spacing - d)(t - t_r)*

A_1 = (4.5-2.5)(1.25-.550) = 1.400 in.^{2}

(6) Examine the diagonal plane, 2-3:

(a) With a circumferential spacing of 3 in. and a longitudinal spacing of 4.5 in., the diagonal center-to-center spacing is p' = $[(3)^2+(2.25)^2]^{½}$ = 3.75 in. And, $\Theta=\tan^{-1}(3/2.25)$; $\Theta = 53.13°$. With a diagonal spacing of 3.75 in., the reinforcement limits of 5.0 in. exceed the spacing. Therefore, the limits overlap and the rules of UG-42 apply. From VIII-1, Fig. UG-37, F=0.68 for Θ=53.13°.

(b) Reinforcement area required is:

$A_r = dt_rF$ = (2.5)(.550)(.68) = 0.935 in.^{2}

(c) Reinforcement area available in shell is:

A_1 = *(spacing - d)(t - Ft_r)*

A_1 = (3.75 - 2.5)(1.25 - .68x.550) = 1.095 in.^{2}

(7) Both the longitudinal and diagonal planes are satisfactory with t = 1.25 in. and the nozzle not considered.

(8) As an alternative, the reinforcement requirements may be based on a combination of shell area and nozzle area.

(9) Minimum required thickness of nozzle as determined from UG-27(c)(1) with E = 1.0 is:

t_{rn} = *(PR)/(SE - 0.6P)*

t_{rn} = [(600)(1.25)] ÷ (20000)(1.0)-.6(600) = 0.038 in.

(10) With the spacing that close, it is doubtful that very much thickness over the minimum required thickness could be added and be able to weld the tubes. Therefore, most, if not all, of the reinforcement area will come from the shell.

Example No. 10

Problem

Determine the minimum required thickness of the shell in Example No. 9 using the ligament efficiency rules of UG-53 instead of the reinforcement rules of UG-36 thru UG-42.

Solution

(1) Determine the minimum longitudinal ligament efficiency or equivalent longitudinal efficiency and compare it with the longitudinal butt joint efficiency. Use lesser efficiency to calculated the minimum required thickness of the shell.

(2) Determine the longitudinal ligament efficiency based on the longitudinal spacing of 4.5 in. as follows:

$$E = (p - d)/p = (4.5 - 2.5)/(4.5) = 0.44$$

(3) Determine the equivalent longitudinal ligament efficiency from the diagonal efficiency using Fig. UG-53.6 as follows:

$p' = p = 3.75$ in.; $\Theta = 53.13°$; $d = 2.5$ in.;

$p/d = 1.5$ which gives E = 0.38.

(4) Using the minimum efficiency of E = .38*, calculate t_r from UG-27(c)(1) as follows:

$$t_r = (PR)/(SE - 0.6P)$$

$$t_r = (600)(18)/(20000)(.38)-.6(600) = 1.492 \text{ in.}$$

Nominal thickness used is $t = 1.5$ in.

* Assumes ligament efficiency < butt joint efficiency.

Example No. 11

Problem

Determine the reinforcement requirements for a 6 in. I.D. nozzle which is located in the center of a flat head. The diameter of the head is 30 in. and the C = 0.33. The head material is SA-516 Gr.70 and the nozzle material is SA-105. The internal design pressure is 800 psi at a design temperature of 400°F. There is no corrosion allowance and there are no butt weld joints within the head.

Discussion

The rules of UG-39 give two alternative methods for determining the required thickness of a flat head with an opening of $d/D \leq 0.5$. One is based on the reinforced opening method and the other is based on adjusting the value of C used for determining the head thickness without an opening. When the C-value method is used, all of the replacement area is obtained from the increase in head thickness only. Consequently, the first solution using the reinforced opening method will assume that a very small amount of replacement area is obtained from the nozzle with most replacement area coming from the thickened head. This will provide a comparison of required head thickness using the two methods. A second reinforced opening solution will assume that most of the replacement area is obtained from excess nozzle wall thickness with a minimum amount coming from the head.

Solution

(1) The allowable tensile stress for both SA-516 Gr.70 and SA-105 at 400°F is 20.0 ksi. Therefore, $f_r = 1.0$.

(2) Using UG-34(c)(2), the minimum required thickness of a flat head is:

$$t_r = d(CP/SE)^{1/2} = (30)[(.33)(800)/(20000)(1.0)]^{1/2}$$

$$t_r = 3.447 \text{ in.}$$

(3) Using UG-27(c)(1), the minimum required thickness of the nozzle is:

$$t_{rn} = (PR)/(SE - 0.6P)$$

$$t_{rn} = [(800)(3)]/[(20000)(1.0) - .6(800)]$$

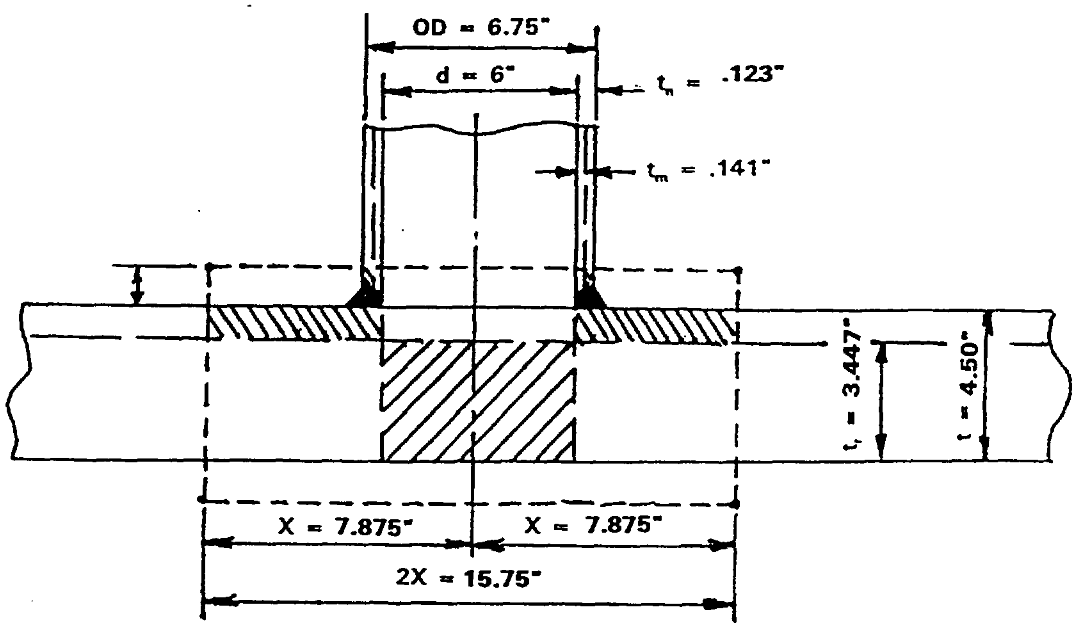
$$t_{rn} = 0.123 \text{ in.}$$

The first solution will compare thickness requirements of the flat head using the reinforced opening method and the C-value method. Since most of the replacement area must come from excess head thickness, an estimate of the required thickness with the opening for trial is $1.3t_r = 1.3(3.447) = 4.48$ in.

Nominal head thickness used is $t = 4.50$ in.

With very little of the replacement area to come from excess thickness in the nozzle, nozzle thickness will be increased only a nominal amount above 0.123 in.

Nominal nozzle thickness used is $t_n = 0.375$ in.



(4) Limit parallel to the flat head surface:

$X = d$ or $(.5d + t + t_n)$, whichever is larger

$X = 6$ or $(3+4.5+.375=7.875)$,use $X = 7.875$ in.

(5) Limit perpendicular to the flat head surface:

$Y = 2.5\ t$ or $2.5\ t_n$, whichever is smaller.

$Y = 2.5(4.5) = 11.25$ in. or $2.5(.375) = 0.938$in.

use $Y = 0.938$ in.

(6) Reinforcement area required following UG-39(b) is:

$A_r = 0.5\ dt_r = .5(6)(3.447) = 10.341$ in.^{2}

(7) Reinforcement area available in flat head is:

$A_1 = (2X - d)(t - t_r) = (2x7.875\text{-}6)(4.5\text{-}3.447)$

$A_1 = 10.267$ in.^{2}

(8) Reinforcement area available in nozzle is:

$A_2 = 2Y(t_n - t_{rn}) = 2(.938)(.375\text{-}.141)$

$A_2 = 0.473$ in.^{2}

(9) Total reinforcement area available from flat head and nozzle is:

$$A_T = A_1 + A_2 = 10.267 + .473 = 10.740 \text{ in.}^2$$

Area available of 10.740 in.^{2} > area required of 10.341 in.^{2}

(10) As an alternative, the minimum required thickness of a flat plate with openings, t_r', may be determined according to UG-39(d)(2). This method requires the value of C used in the flat head equation without openings to be equal to the lesser of 2C or 0.75 as follows:

$$2C = 2(.33) = .66 \text{ or } 0.75\text{, whichever is less.}$$

(11) Using t_r from (2), above, the minimum required thickness of the flat head with openings, t_r', is:

$$t_r' = \sqrt{2}\, t_r = (1.414)(3.447) = 4.87 \text{ in.}$$

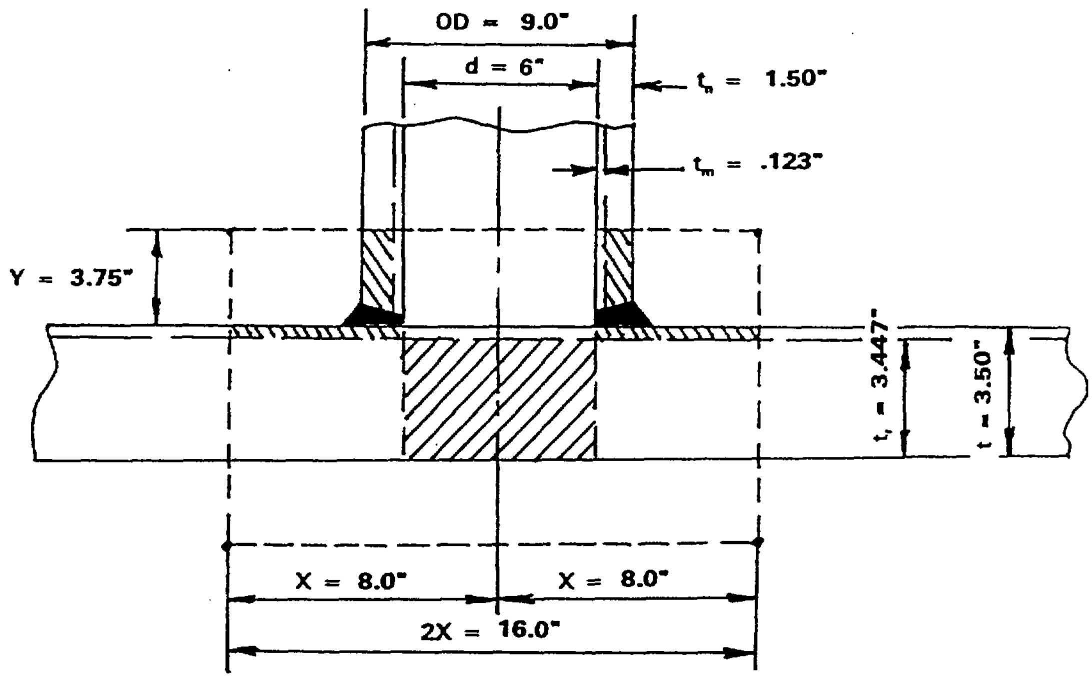
For this solution, the reinforced opening method results in lesser minimum required thickness of $t_r = 4.50$ in. for the flat head with the opening than the 2C method which requires $t_r = 4.87$ in.

The second solution will obtain most of the replacement area from excess nozzle wall thickness with the head thickness remaining close to the required thickness without an opening which is $t_r = 3.447$ in.

Nominal head thickness used is $t = 3.50$ in.

The nominal wall thickness is determined by trial-and-error because the limits of reinforcement change with a change in thickness.

Nominal nozzle thickness used is $t_n = 1.50$ in.



(12) Limit parallel to flat head surface:

$$X = 6 \text{ or}(3+3.5+1.5=8.00), \text{ use } X=8.00 \text{ in.}$$

(13) Limit perpendicular to flat head surface:

$$Y = 2.5(1.50) = 3.75 \text{ in., use } Y = 3.75 \text{ in.}$$

(14) Reinforcement area available in flat head is:

$$A_1 = (2\text{x}8.0\text{-}6)(3.5\text{-}3.447) = 0.530 \text{ in.}^2$$

(15) Reinforcement area available in nozzle wall is:

$$A_2 = 2(3.75)(1.50\text{-}.123) = 10.327 \text{ in.}^2$$

(16) Total reinforcement area available:

$$A_T = A_1 + A_2 = 0.530 + 10.327 = 10.857 \text{ in.}^2$$

Area available of 10.857 in.^{2} > area required of 10.341 in.^{2}

Comparing all results, it appears that the most economical solution is to increase the thickness of the nozzle wall with no (or very little) increase in the head thickness above that required for the head thickness without an opening.

Example No. 13

Problem

Using the rules of Appendix 2, determine the minimum required thickness of a weld neck flange as shown with the following design data:

Design pressure = 2,000 psi,

Design temperature = 650 °F,

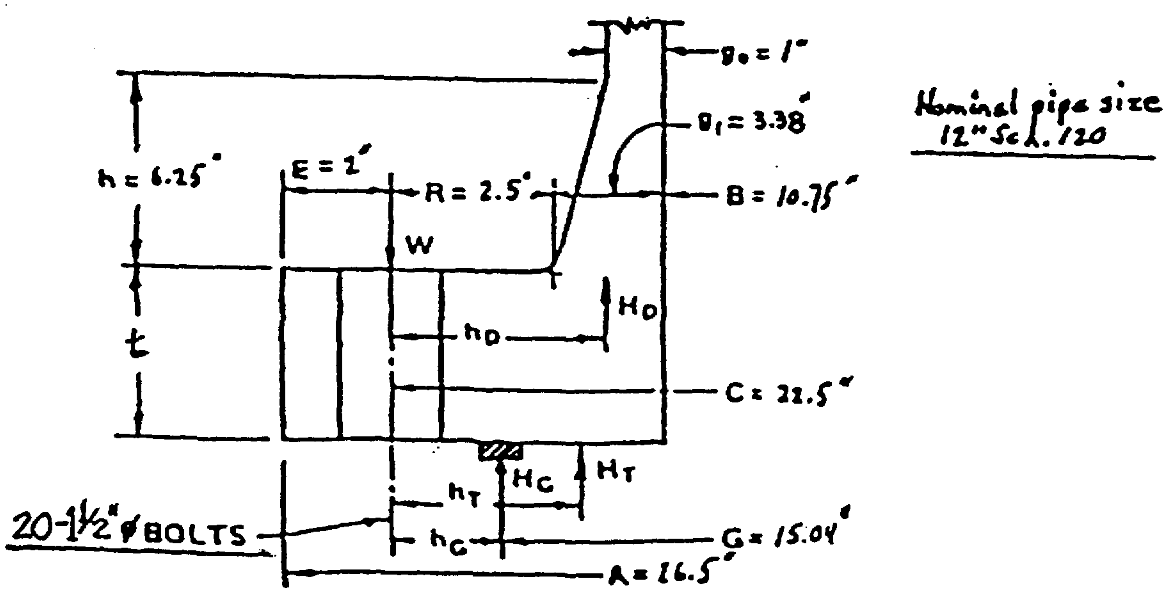
Flange material is SA-105,

Bolting material is SA-325 Gr.1,

Gasket is spiral-wound, fiber-filled, stainless steel,

13.75 in. I.D.x 1.0 in. wide,

No corrosion allowance.



Solution

(1) The allowable tensile stress for SA-105 is S_{fa} = 20.0 ksi at room temperature and S_{fo} = 17.8 ksi at 650°F and for SA-325 Gr.1 is $S_a = S_b$ = 20.2 ksi at R.T. thru 650°F.

(2) The diameter of the gasket line-of-action, G, is determined as follows:

$$b_o = N/2 = 0.5 \text{ in. and } b = 0.5(b_o)^{1/2} = 0.3535 \text{ in.}$$

$$G = 13.75 + (2 \times 1) - (2 \times .3535) = 15.043 \text{ in.}$$

(3) Bolt loadings and number & diameter of bolts with $N = 1$, $b = 0.3535$, $m = 3.0$, and $y = 10{,}000$ is:

$H = (\pi/4)G^2p = (\pi/4)(15.043)^2(2000) = 355{,}500$ lbs.

$H_p = 2b\pi Gmp = 2(.3535)\pi(15.043)(3.0)(2000)$

$H_p = 200{,}500$ lbs.

$W_{m1} = H + H_p = 355500 + 200500 = 556{,}000$ lbs.

$W_{m2} = \pi bGy = \pi(.3535)(15.043)(10000)$

$W_{m2} = 167{,}100$ lbs.

A_m = larger of W_{m1}/S_b or $W_{m2}/S_a = 556{,}000/20.2 = 27.5$ in^2

Using 20 bolts at 1½-in. dia.:

A_b = actual bolt area = 20(1.41) = 28.2 in.^{2}

$W_a = 0.5(A_m + A_b)S_b = 0.5(27.5 + 28.2)(20{,}200)$

$W_a = 562{,}600$ lbs.

$W_o = W_{m1} = 556{,}000$ lbs.

(4) The total flange moment for gasket seating condition is:

Flange Load

$H_G = W_a = 562{,}600$ lbs.

Lever Arm

$h_G = 0.5(C - G) = 3.729$ in.

Flange Moment

$M_{GS} = H_G \times h_G = (562600)(3.729) = 2{,}098{,}000$"#

(5) The total flange moment for operating condition is:

Flange Loads

$H_D = (\pi/4)B^2p = (\pi/4)(10.75)^2(2000) = 181{,}500$ lbs.

$H_G = H_p = 200{,}500$ lbs.

$H_T = H - H_D = 355500 - 181500 = 174{,}000$ lbs.

Lever Arms

$h_D = R + 0.5g_1 = 2.5 + 0.5(3.375) = 4.188$ in.

$h_G = 0.5(C - G) = 0.5(22.5 - 15.043) = 3.729$ in.

$h_T = 0.5(R + g1 + h_G) = 0.5(2.5 + 3.375 + 3.729)$

$h_T = 4.802$ in.

Flange Moments

$M_D = H_D \times h_D = (181500)(4.188) = 760{,}100$ in.-lbs.

$M_G = H_G \times h_G = (200500)(3.729) = 747{,}700$ in.-lbs.

$M_T = H_T \times h_T = (174000)(4.802) = 835{,}500$ in.-lbs.

$M_o = M_D + M_G + M_T = 2{,}343{,}000$ in.-lbs.

(6) Shape factor from Appendix 2, $K = A/B = 26.5/10.75$,

$K = 2.465$

From Fig. 2-7.1 of VIII-1:

$T = 1.35$ $Z = 1.39$ $Y = 2.29$ $U = 2.51$

$g_1/g_o = 3.375/1.0 = 3.375$

$h_o = (Bg_o)^{½} = [(10.75)(1.0)]^{½} = 3.279$

$h/h_o = 6.25/3.279 = 1.906$

From VIII-1, Appx. 2: $F = 0.57$; $V = 0.04$; $f = 1.0$

$e = F/h_o = 0.57/3.279 = 0.174$

$d = (U/V)h_o g_o^2 = (2.51/0.04)(3.279)(1)^2 = 205.76$

(7) Calculate stresses for gasket seating condition based on M_{GS} and S_{fa} and for operating condition based on M_o and S_{fo}. If M_o is larger than M_{GS}, and S_{fo} is equal to or less than S_{fa}, only the operating condition stresses are calculated as in this design. Calculate stresses based on $M_o = 2,343,000$ in.-lbs.

Nominal flange thickness used is $t = 4.0$ in.

$L = [(te+1)/T + (t)^3/d] = 1.256+0.311 = 1.567$

Longitudinal hub stress:

$S_H = fM_o/Lg_1^2B$

$S_H = (1)(2343200)/(1.567)(3.375)^2(10.75)$

$S_H = 12,210$ psi

Radial flange stress:

$S_R = [(4/3)te + 1]M_o/Lt^2B$

$S_R = (1.928)(2343200)/(1.567)(4)^2(10.75)$

$S_R = 16,760$ psi

Tangential flange stress:

$S_T = [(YM_o/t^2B) - ZS_R]$

$S_T = \{[(2.29)(2343200)/(4)^2(10.75)]$

$- (1.39)(16760)\}$

$S_T = 7{,}900$ psi

Combined stresses:

$$0.5(S_H + S_R) = .5(12210 + 16760) = 14{,}490 \text{ psi}$$

$$0.5(S_H + S_T) = .5(12210 + 7900) = 10{,}060 \text{ psi}$$

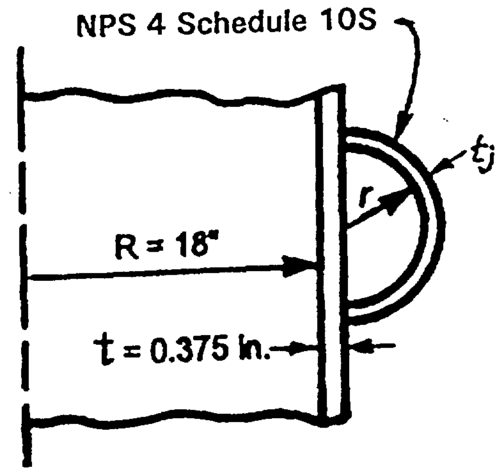
(8) Calculated vs. allowable stresses:

$S_H \leq 1.5S_f$:	12,210 psi < 26,700 psi
$S_R \leq S_f$	16,760 psi < 17,800 psi
$S_T \leq S_f$:	7,900 psi < 17,800 psi
$0.5(S_H + S_R) \leq S_f$:	14,490 psi < 17,800 psi
$0.5(S_H + S_T) \leq S_f$:	10,060 psi < 17,800 psi

Since all calculated stresses are less than the allowable stresses, the selection of $t = 4.0$ in. is adequate! If an optimum minimum thickness of the flange is desired, calculations must be repeated with a smaller value of t until one of the calculated stresses or stress combinations is approximately equal to the allowable stress even though other calculated stresses are less than the allowable stress for that calculated stress.

Example No. 16

Problem



Using Appendix EE, determine the required thicknesses of a cylindrical shell with an internal design pressure of 400 psi and an externally attached NPS 4 Sch. 10S half-pipe jacket with an internal design pressure of 500 psi in noncyclic service. Both have a design temperature of 100°F. The inside diameter of the shell is 36 in. and E = 1.0. The shell material is SA-516 Gr. 70 and the jacket material is SA-53 Gr.S/A. There is no radiography of jacket/shell weld joints and no corrosion allowance.

Solution

(1) From Table 1A of II-D, at 100°F, the allowable stress for SA-516 Gr.70 is 20.0 ksi and for SA-53 Gr. S/A seamless pipe is 13.7 ksi.

(2) Using UG-27(c)(1), minimum required thickness of the cylindrical shell is:

$$t_r = (PR) / (SE - 0.6P)$$

$$t_r = [(400)(18)] / [20000 \times 1.0 - 0.6 \times 400]$$

$$t_r = 0.364 \text{ in.}$$

Nominal thickness used is t = 3/8 in.

(3) From Fig. EE-3 with D = 36 in. and t = 3/8 in., K = 40.

$S' = PR/2t = (400x18) / (2x0.375) = 9600$ psi

$P' = F/K = (1.5x20000 - 9600)/40$

$P' = 510$ psi > 500 psi, o.k.

(4) Assuming NPS 4 Schedule 10S,

$t_j = 0.120 \times 0.875 = 0.105$ in.

$r_j = [(4.5/2) - 0.105] = 2.145$ in.

(5) Minimum thickness of jacket using EE-1, Eq.(2) is:

$t_{rj} = [(500)(2.145)] / [0.85x13700 - 0.6x450]$

$t_{rj} = 0.095$ in. < 0.105 in. actual

(6) Minimum fillet weld size is 0.120 x 1.414 = 0.170 in. Use 3/16 in. weld

(7) **Summary:** Use shell thickness of 3/8 in.; half-pipe jacket of NPS 4 Schedule 10S; and 3/16 in. fillet weld size to attach half-pipe jacket to shell.

Suggested references

- Farr, James R., and Maan H. Jawad: "Guidebook for the Design of ASME Section VIII Pressure Vessels," ASME Press, 1998.

- Jawad, Maan H., and James R. Farr: "Structural Analysis and Design of Process Equipment," Second Edition, John Wiley & Sons, Inc., 1988.

- Brownell, L. E., and E. H. Young: "Process Equipment Design," John Wiley & Sons, Inc., 1959.

- Harvey, John F.: "Theory and Design of Pressure Vessels," Second Edition, Van Nostrand Reinhold Co., 1991.

- Megyesy, Eugene F.: "Pressure Vessel Handbook," Eleventh Edition, Pressure Vessel Handbook Publishing Co., 1998.

- Young, Warren C.: "Roark's Formulas for Stress & Strain," Sixth Edition, McGraw-Hill Book Co., 1989.

The Design of Vertical Pressure Vessels Subjected to Applied Forces

By E. O. BERGMAN,[1] ALHAMBRA, CALIF.

Pressure-vessel codes do not give design methods except for the relatively simple case of cylindrical shells with standard-type heads and openings under uniform pressure. The designer must apply engineering principles when he deals with more complicated structures and loading systems. This paper discusses some design principles that are not covered in the codes. It deals with vessels that are subjected to various applied forces acting in combination with internal or external pressure. The type of vessels considered is limited to cylindrical shells with the longitudinal axis vertical.

Nomenclature

The following nomenclature is used in the paper:

- A = cross-sectional area of shell, sq in.
- C = acceleration ratio specified by structural codes for use with increased stress values
- D = outside diameter of shell, in.
- e = eccentricity of resultant load, in.
- K = equivalent acceleration ratio for use with basic stress values permitted by vessel codes
- L = length of shell between stiffeners, in.
- M = bending moment due to horizontal loads, in-lb
- m = numerical ratio depending on R and L
- n = number of lobes into which shell may buckle
- P = end load in addition to external pressure, pounds per lineal inch
- p = internal pressure, psi
- R = outside radius of shell, in.
- R = Reynolds number
- r = numerical ratio defined by Equation [1]
- r' = numerical ratio defined by Equation [2]
- r'' = numerical ratio defined by Equation [3]
- S_a = basic allowable stress value permitted by codes
- S_b = bending stress on outermost fiber, psi
- S_c = longitudinal compressive stress in shell, psi
- S_E = stress produced by seismic loads for an acceleration ratio of unity, psi
- S_t = longitudinal tensile stress in shell, psi
- S_V = stress produced by vertical loads, psi
- t = thickness of shell, in.
- W = weight above section under consideration, lb
- W_c = collapsing pressure for external pressure acting on sides of vessel only, psi
- W_c' = collapsing pressure for external pressure acting on sides and ends of vessel, psi
- W_c'' = collapsing pressure for external pressure on sides and ends when acting in conjunction with an axial compression of P pounds per lineal inch of shell, psi
- W_4' = W_c' divided by a factor of 4 against collapse
- W_4' = W_c' divided by a factor of 4 against collapse
- Z = section modulus of shell, cu in.
- $\alpha = \dfrac{2P}{W_c R}$ = axial compression per lineal inch due to externally applied loads divided by axial compression set up by that value of external pressure in pounds per square inch which acting by itself would produce collapse

[1] Staff Consultant, C. F. Braun & Company. Mem. ASME.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, August 18, 1954. Paper No. 54—A-104.

Introduction

The pressure-vessel codes (1, 2)[2] give a list of the principal loading conditions that the designer should consider in designing a vessel. These conditions may be divided into pressure loadings and applied forces. Pressures are applied either internally or externally over the surface of the vessel. Applied forces act either at local points or throughout the mass of the vessel.

The codes furnish the designer with a list of approved materials and the maximum stress values in tension permitted over their usable range of temperatures. The design rules in the codes are limited to vessels of cylindrical or spherical shape under internal or external pressure, and to heads and nozzle attachments for such vessels. Rules for more complicated types of construction and for loadings other than that due to pressure are beyond the scope of the code. To include such rules would turn the code into a design handbook. And it would restrict the designer in working out his design in accordance with acceptable engineering principles. The code requires that he "shall provide details of construction that will be as safe as those provided by the rules of the code."

This paper discusses some problems of design of cylindrical pressure vessels that have their axes vertical and are subjected to applied forces in addition to internal or external pressure. The vertical forces considered are the weight of the vessel and its contents and the weight of any attachments to the vessel. The horizontal forces include wind pressures, seismic forces, and piping thrusts.

Loads

The vertical loads consist primarily of forces due to gravity, that is, to weight. The vertical component of piping thrusts also must be considered. Liquid contents normally are carried by the bottom head and the vessel supports. But in fractionating columns, the weight of the liquid on internal trays is transferred into the shell. Part of the weight of stored solids is transferred into the shell by friction. The weights of attachments that are eccentric to the axis of the vessel produce bending moments which must be considered in the design.

Wind Load. The force per unit area exerted by the wind depends on a number of factors, including wind velocity, height above ground, and drag coefficient. This last includes height-to-diameter ratio and shape factor. ASA Standard A58.1-1945, Minimum Design Loads in Buildings and Other Structures (3), gives a map of the United States showing isograms of equal velocity pressures. It includes also a discussion of methods of arriving at wind pressures from Weather Bureau wind velocities.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

The discussion does not consider the effect of velocity on the drag coefficient. The drag coefficient, or friction factor, for a given shape varies with the Reynolds number. For a circular cylinder, the coefficient is practically constant for values of Reynolds number between R = 20,000 and R = 200,000. Above R = 500,000, the coefficient drops to less than half its value in the lower range (4).

The values of wind pressure used by designers are usually taken from structural codes or from a purchaser's specification. All such values appear to be based on the use of a drag coefficient for Reynolds number in the region between R = 20,000 and R = 200,000. The value of Reynolds number for a circular cylinder is equal to 9100 DV, where D is the diameter in feet and V is the wind velocity in miles per hour. With a value of DV greater than 55, the drag on the vessel would be less than half that specified in the codes. Thus the codes might well consider the advisability of reducing the wind pressures to be used with circular vessels.

Earthquake. The behavior of a structure in an earthquake is one of vibration under variable conditions of acceleration. For a discussion of the problem from a dynamics approach, see the paper, "Lateral Forces of Earthquake and Wind," by a Joint Committee of the San Francisco, California Section, ASCE, and the Structural Engineers Association of Northern California (5).

The usual simplified approach to the problem is based on the assumption that the structure is a rigid body which undergoes the accelerations of the supporting ground. The horizontal force which acts on the structure is equal to its mass times the ground acceleration, and has the same ratio to the weight as the ground acceleration has to that of gravity. Structural codes give values of this ratio that are based on engineering experience and judgment.

The most widely used code with rules for earthquake design is the Uniform Building Code of the Pacific Coast Building Officials Conference (6). It gives acceleration ratios, or *C*-factors, for three zones of earthquake intensity. The ratios specified for tanks, smokestacks, standpipes, and similar structures are 0.025, 0.05, 0.10, respectively, for the three zones.

Stress Increase. Structural codes provide for the use of increased allowable stress values when loads due to wind pressure or seismic effect are included. The increase most commonly specified is 33$^1/_3$ per cent. The load values for wind pressures and acceleration ratios in these codes were set with this increase in mind. Pressure-vessel codes do not provide for such stress increases. To maintain consistency between the two types of codes, the acceleration ratios given in structural codes should not be used directly in designing pressure vessels. A correction should be made to offset the effect of the increase in allowable stress values permitted in the structural codes. The correction may be made by modifying the value of the acceleration ratio in such a way that the stresses computed in a shell or structure are the same with either type of code.

The modified acceleration ratio for use with vessel codes must satisfy the following relation

$$\frac{S_v + KS_H}{S_a} = \frac{S_v + CS_H}{1.33S_a}$$

Clearing of fractions

$$S_v + KS_H = 0.75S_v + 0.75CS_H$$

or

$$K = 0.75C - 0.25\frac{S_v}{S_H}$$

The vessel designer who must meet the earthquake requirements of a structural code is in somewhat of a dilemma. The structural code specifies certain acceleration ratios but permits him to use a stress of 4/3 × 20,000 = 26,670 psi for a vessel constructed of plate to Specification SA-285, Grade D. The pressure-vessel codes specify an allowable stress value of 13,750 psi but leave the selection of an acceleration ratio to the designer. Until the vessel codes permit the use of increased stress values for earthquake loads, the equivalent acceleration ratios K derived in the foregoing offers a reasonable solution to the dilemma. Even then, the designer comes up with a thicker shell than the structural rules require because of the lower basic stress values specified in the vessel codes.

It would be possible to calculate a reduced wind load in the same way as was done for earthquake load. There is this difference, however. The specified wind loads are based on measurements of the forces exerted by the wind on structures, and have less of the element of judgment which is involved in setting the earthquake-acceleration ratios. While there is justification for changing the judgment factors for earthquake when the stress values that helped to determine the factors are changed, a parallel change in wind loads which are based on wind intensity has little justification. A direct approach would be for the vessel codes to permit the use of the one-third increase in stress values for wind and earthquake, when designers are to use the wind-load and earthquake-load values that are given in the structural codes.

Stress Determination

The vertical loads on the vessel set up compressive stresses in the shell, and also bending stresses when the resultant force does not coincide with the axis of the vessel. The stresses set up at any section of the shell by the vertical loads are given by equations

$$S_a = \frac{W}{A} = \frac{W}{\pi Dt} \quad \text{and} \quad S_b = \frac{4We}{\pi D^2t}$$

The horizontal loads on the vessel produce bending stresses in the shell. The bending moment at any section is equal to the resultant of the horizontal forces above the section multiplied by the distance between the line of action of the resultant and the section. The stress, set up in the outermost fiber of the shell by the action of horizontal loads, is equal to

$$S_b = \frac{4M}{\pi D^2t}$$

The stresses due to external loads must be considered in combination with those due to pressure in determining the required shell thickness. For internal pressure, the stresses may be combined by simple addition. For external pressure, a more complicated procedure is required.

Allowable Stress Values

The pressure-vessel codes give tables of allowable stress values in tension for all materials approved for code use. The ASME Subcommittee on Unfired Pressure Vessels has approved for submission to the Main Committee a method for obtaining allowable stress values in compression for ferrous materials. These values are obtained from the charts given in the code for determining the thickness of shells and heads under external pressure. The wording of the proposed method is as follows, except for the addition of paragraph references.

The maximum allowable compressive stress to be used in the design of cylindrical shells, subjected to loadings that produce longitudinal compressive stresses in the shell, shall be the smaller of the following values:

1 The maximum allowable tensile-stress value permitted in Par UG-23(a).

2 The value of the factor B determined from the applicable chart in Subsection C for determining the required thickness of

shells and heads under external pressure, using the following definitions for the symbols on the chart (References are to Section VIII of the ASME Boiler and Pressure Vessel Code.):

t_s = minimum required thickness of shell plates, exclusive of corrosion allowance, in.

L_s = inside radius of cylindrical shell, in.

The value of factor B shall be determined from the applicable chart of Subsection C in the following manner:

Step 1. Assume a value of t_s. Determine the ratio $L_s/100t_s$.

Step 2. Enter the left-hand side of the chart in Subsection C for the material under consideration at the value $L_s/100t_s$ determined in Step 1.

Step 3. Move horizontally to the line marked "sphere line."

Step 4. From this intersection move vertically to the material line for the design temperature. (For intermediate temperatures, interpolations may be made between the material lines on the chart.)

Step 5. From this intersection move horizontally to the right-hand side of the chart and read the value of B. This is the maximum allowable compressive stress value for the value of t_s used in Step 1.

Step 6. Compare this value of B with the computed longitudinal compressive stress in the vessel, using the assumed value of t_s. If the value of B is smaller than the computed stress, a greater value of t_s must be selected and the procedure repeated until a value of B is obtained which is greater than the computed compressive stress for the loading on the vessel.

The joint efficiency for butt-welded joints may be taken as unity for compressive loading.

Design With Internal Pressure

The axial stresses set up in the shell may be classified under three types: (*a*) The longitudinal stress produced by the internal pressure; (*b*) the uniform compressive stress produced by the sum of the weights assumed to act along the axis of the vessel; (*c*) the bending stress produced by the horizontal loads and by the resultant weight when eccentric to the axis of the vessel.

Tests carried on at the University of Illinois (7, 8) indicate that a somewhat higher computed stress is required to produce failure under combined bending and compression than under compression alone. Thus we may safely combine compressive stresses due to bending with those due to uniform compression, and design the vessel shell as though these stresses were all due to uniform compression.

The tension side of the shell has its highest stress when the vessel is under pressure. On the compression side, the highest stress occurs when the internal pressure is not acting. The stresses set up in the shell for these two conditions are

Tension $$S_t = \frac{pD}{4t} - \frac{W}{\pi Dt} + \frac{4We}{\pi D^2t} + \frac{4M}{\pi D^2t}$$

Compression $$S_c = -\frac{W}{\pi Dt} - \frac{4We}{\pi D^2t} - \frac{4M}{\pi D^2t}$$

The factor W includes all the vertical loads and the factor M includes all the moments due to horizontal loads for the loading condition under consideration. A value of shell thickness must be selected so that these stresses are not greater than the allowable stress values, taking into account the applicable joint efficiencies.

Design With External Pressure

The code charts for determining the required thickness of shells under external pressure have been developed for the condition of a uniform pressure on the cylindrical surface and the heads of the vessel. The longitudinal compressive stresses set up in the shell by weight and lateral forces have an effect similar to that produced by subjecting the heads of the vessel to a higher external pressure than that which acts on the shell. The charts can be used to give an approximate solution for such a load condition by a suitable change in the vertical scale on which the factor B is read. The nature of the approximations will be discussed later. In practice, it is simpler to leave the scale unchanged and make the adjustment in the value of pressure to be used in reading the chart.

In reference (9) Sturm gives a method of dealing with end loads on the heads. This method can be used as the basis for the design of a vessel under external pressure and subjected to applied loads.

Using Sturm's Equation [45], the ratio of the collapsing pressures W_c and W_c' is equal to

$$r = \frac{W_c'}{W_c} = \frac{F}{F + \frac{\pi^2R^2}{2L^2}}$$

or

$$r = \frac{n^2-1}{n^2-1+m} \qquad [1]$$

where

$$m = \frac{\pi^2R^2}{2L^2} = \frac{\pi^2D^2}{8L^2} = \frac{1.23D^2}{L^2}$$

and F is given[3] as approximately equal to n^2-1, where n is the number of lobes into which the shell may buckle. By Sturm's Equation [46], the ratio of the collapsing pressures W_c and W_c'' is equal to

$$r' = \frac{W_c''}{W_c} = \frac{F}{F + \frac{\pi^2R^2}{2L^2} + \frac{2P}{W_c'R}\frac{\pi^2R^2}{2L^2}}$$

or

$$r' = \frac{n^2-1}{n^2-1+m+m\alpha} \qquad [2]$$

In this equation $\alpha = 2P/(W_c'R)$, where P is the axial compression per lineal inch due to the externally applied loads, and $W_c'R/2$ may be looked on as the axial compression per lineal inch in the shell, if the collapsing value of the external pressure were acting on the ends of the vessel.

Since the ASME Code charts are made up on the basis of the collapsing pressure W_c', the ratio of W_c' to W_c'' is needed to make use of the charts. This is found by dividing Equation [1] by Equation [2], whence

$$r'' = \frac{W_c'}{W_c''} = \frac{n^2-1+m+m\alpha}{n^2-1+m} \qquad [3]$$

Windenburg and Trilling (10) have developed a chart which gives n as a function of t/D and L/D for pressure on the sides and ends of the vessel. This chart is reproduced in Fig. 1. A comparison of Sturm's Figs. 4 and 8 indicates that the values of n for different values of t/D and L/D change very little between the condition of pressure on the sides only and that of pressure on both sides and ends. Thus Fig. 1 should give satisfactory values of n for external loading conditions for which α is not much greater than one. For large values of α, the shell will buckle in fewer lobes than the number given in Fig. 1, or it may even fail by plastic flow without the formation of any lobes. For values of α greater than one, the vessel should also be checked as a cantilever beam, including the axial stress due to external pressure in the computations.

The relation between W_c' and W_c'' given by r'' is a ratio. Hence

[3] Reference (9), p. 25.

the value of r' can be used equally well for the relation between allowable external working pressures. Thus the code charts can be used to determine the required thickness with external loads and moments by using an equivalent-design external pressure W_e' equal to

$$W_e' = \frac{n^2 - 1 + m + m\alpha}{n^2 - 1 + m} \times W_e'' \ldots\ldots [4]$$

where W_e' and W_e'' are equal to the values of W_s' and W_s'' divided by the factor of 4 against collapse.

The use of Equation [4] in connection with the code charts implies that the pressure on the sides of the vessel is increased in the same ratio that the applied vertical forces increase the axial compression in the shell. Since the applied loads do not increase the circumferential compression in the shell, the use of Equation [4] gives answers that are somewhat on the side of safety. The design procedure will be illustrated by an example.

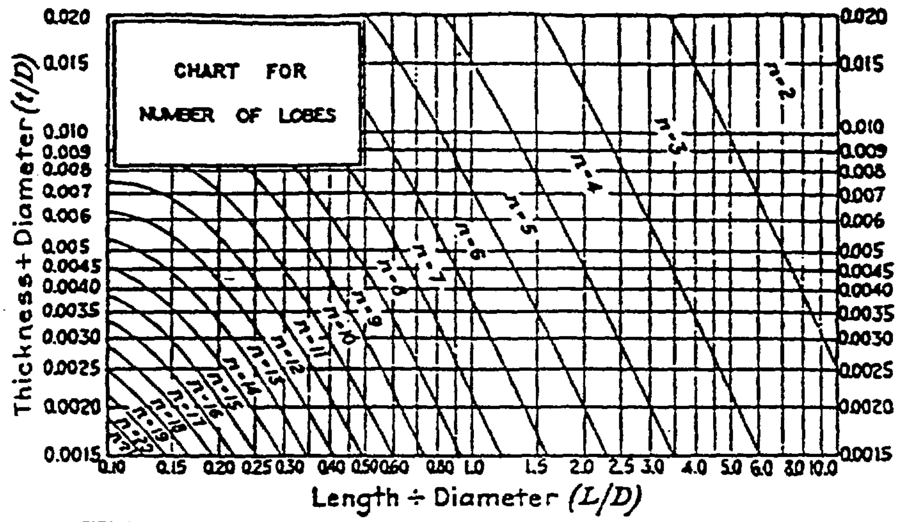

Fig. 1 Number of Lobes n Into Which a Shell Will Collapse When Subjected to Uniform Collapsing Pressure on Sides and Ends

Example. Given a cylindrical vessel fabricated from SA-285, Grade B material, to operate under an external pressure of 15 psi (vacuum) at 200 F. The vessel is 10 ft diam and 100 ft high with stiffening rings spaced 6 ft apart. The total vertical load is 200,000 lb and the moment of the external forces at the bottom head seam is 2,000,000 ft-lb. What is the required shell thickness?

Solution. The maximum compressive load in pounds per lineal inch due to weight and moment is

$$P = \frac{W}{\pi D} + \frac{4M}{\pi D^2}$$

$$= \frac{200{,}000}{120\pi} + \frac{8{,}000{,}000 \times 12}{14{,}400\pi}$$

$$= 530 + 2120 = 2650 \text{ lb per lineal in. of circumference}$$

$$\alpha = \frac{2P}{W_e'R} = \frac{4P}{W_e'D} = \frac{P}{W_e'D} = \frac{2650}{15 \times 120} = 1.47$$

Assume $t = 3/8$ in.; then $D/t = 320$, $t/D = 0.00312$, $L/D = 0.6$, $m = 1.23/0.36 = 3.42$, and from Fig. 1, $n = 9$

$$W_e' = \frac{81 - 1 + 3.42 + 3.42 \times 1.47}{81 - 1 + 3.42} \times 15 = 15.9 \text{ psi}$$

Enter Fig. UCS-28 in the ASME Code for Unfired Pressure Vessels, Section VIII, 1952 edition, with $L/D = 0.6$ and $D/t = 320$. Then $B = 6300$ and $W_e' = 6300/320 = 19.7$ psi.

Since α is greater than 1, check the vessel as a cantilever beam. The axial stress due to vacuum is equal to

$$\frac{15 \times 120}{4} = 450 \text{ lb per lineal in.}$$

This gives a total axial stress of

$$450 + 2650 = 3100 \text{ lb per in. or } 3100/0.375 = 8260 \text{ psi}$$

$$L_1/100t_1 = \frac{60}{100 \times 0.0375} = 1.6$$

Enter Fig. UCS-28 with $L_1/100t_1 = 1.6$ and read $B = 10{,}300$. Thus the assumed thickness of $3/8$ in. is satisfactory.

Summary

External loads applied to vertical pressure vessels produce axial loading and bending moments on the vessel. These result in axial tensions and compressions in the shell, which must be combined with the effects of the pressure loading to give the total longitudinal stress acting in the shell.

The design method to be used depends on whether the longitudinal stress in the shell is tension or compression, and on whether the vessel is subjected to internal or external pressure.

Bibliography

1 "Rules for Construction of Unfired Pressure Vessels," Section VIII, ASME Boiler and Pressure Vessel Code, 1952 edition.

2 "API-ASME Code for the Design, Construction, Inspection, and Repair of Unfired Pressure Vessels for Petroleum Liquids and Gases," fifth edition, 1951.

3 "Minimum Design Loads in Buildings and Other Structures," A58.1-1945, American Standards Association, New York, N. Y., 1945.

4 "Applied Hydro- and Aeromechanics," by Prandtl and Tietjens, McGraw-Hill Book Company, New York, N. Y., 1934, pp. 96–97.

5 "Lateral Forces of Earthquake and Wind," Paper 2514, Trans. ASCE, vol. 117, 1952, pp. 716–780.

6 "Uniform Building Code," Pacific Coast Building Officials Conference, Los Angeles, Calif., 1952 edition.

7 "The Strength of Thin Cylindrical Shells as Columns," by W. M. Wilson and N. M. Newmark, University of Illinois, Engineering Experiment Station, Bulletin No. 255, 1933.

8 "Tests of Cylindrical Shells," by W. M. Wilson and E. D. Olson, University of Illinois, Engineering Experiment Station, Bulletin No. 331, 1941.

9 "A Study of the Collapsing Pressure of Thin-Walled Cylinders," by R. G. Sturm, University of Illinois, Engineering Experiment Station, Bulletin No. 329, 1941.

10 "Collapse by Instability of Thin Cylindrical Shells Under External Pressure," by D. F. Windenburg and C. Trilling, Trans. ASME, vol. 56, 1934, pp. 819–825.