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# SURVEY REPORT ON STRUCTURAL DESIGN OF PIPING SYSTEMS AND COMPONENTS

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# FOREWORD

This report was written under subcontract to the Oak Ridge National Laboratory, operated by Union Carbide Corporation for the U. S. Atomic Energy Commission, in support of the ORNL Piping Program — Design Criteria for Piping, Pumps, and Valves. The ORNL Piping Program is funded by the USAEC under the Nuclear Safety Research and Development Program (AEC Activity No. 04 60 80 03 1) as the AEC supported portion of an AEC-Industry cooperative effort for the development of design criteria for piping components, pumps, and valves. It is related to both water-cooled nuclear reactor plants and to liquid-metal fast breeder reactors under Section 3-8.2 of the LMFBR Program Plan. The program is under the direction of W. L. Greenstreet, Head, Applied Mechanics Section, Oak Ridge National Laboratory, and S. E. Moore, Program Coordinator. The USAEC cognizant engineer is J. L. Mershon.

Information developed under the ORNL Piping Program is provided to both government and industrial groups engaged in writing codes and standards for the design and construction of nuclear plant piping systems. These include the AEC Division of Reactor Development and Technology RDT Standards Program, the American National Standards Institute, and the American Society of Mechanical Engineers. Liaison between the ORNL Piping Program and the industrial groups is carried out through the Pressure Vessel Research Committee of the Welding Research Council.

# SURVEY REPORT ON STRUCTURAL DESIGN OF PIPING SYSTEMS AND COMPONENTS

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# 1. INTRODUCTION

## 1.1 Background and Acknowledgments

Both the Nuclear Power Piping Code, USAS B31.7 (1.1), and the ASME Code for Pumps and Valves for Nuclear Power (1.2) require analyses of Class I components and piping systems which establish that the stresses do not exceed specified limits. Because of the complex geometric shapes of some of the components and the nature of the loadings on these components, the stress analyses may be quite complicated and costly. However, the design of many of the piping components, pump casings and valve bodies are to some extent standardized. Once an analysis is available which covers an adequate range of dimensional parameters for these standard components, the future cost of the required analyses will become relatively small. The intent of the nuclear piping and pump and valve codes is to provide design analysis procedures for the most commonly used components in order to reduce the time and cost of the analysis and to insure a high degree of structural safety. The intent is to provide information analogous to that given for pressure vessels in the ASME Boiler and Pressure Vessel Code, Section III (1.3), Appendix I, Articles I-6 and I-9.

The present issue of B31.7 contains two acceptable methods of analysis. The "simplified" method is contained in Par. 1-705. The "detailed" method is contained in Appendix F. The simplified method uses stress indices for maximum stresses in a component due to each load, and combines the stresses for various loads by direct addition. This is a conservative method because in general, maximum stresses due to different loads do not occur at the same point in the component. The detailed method permits the analyst to examine each point in the piping component for stresses due to each load. At present, B31.7, Table D-201 contains an extensive, although not complete, coverage of stress indices for the simplified method. Stress indices for the detailed method are, at present, given only for curved pipe or welding-end elbows and for certain types of branch connections with internal pressure loading.

The ASME Nuclear Pump and Valve Code has adopted a design analysis philosophy which is similar in many respects to that of the USAS B31.7 Nuclear Piping Code. However, because of the general absence of published stress analysis information and the more complicated geometries involved, the design section for pumps in the pump and valve code is (at this writing) not fully developed.

The writers of both codes recognized that the existing stress indices were incomplete, needed refinement in detail, extension in coverage and in some cases complete development. It was also recognized that a substantial amount of work would be required to obtain the necessary information. The problem was therefore taken to the Pressure Vessel Research Committee (PVRC) of the Welding Research Council who established (December, 1966) an Ad Hoc Committee to Develop Stress Indices for Piping, Pumps, and Valves. This Ad Hoc Committee developed a suggested program[1.4] consisting of twelve tasks for developing most of the required information.

The PVRC program was sent to the Atomic Energy Commission (AEC) with a formal request for support on August 3, 1967. A reply from the AEC (Milton Shaw, DRDT to C. F. Larson, PVRC, January 9, 1968) gave concurrence, in principle, to the desirability of undertaking a program of the general type proposed as a cooperative effort between AEC and industry. Subsequently the AEC agreed to sponsor a portion of the work on piping components, specifically Tasks 1 through 6 and Task 8 of the PVRC program with management of the AEC sponsored portion to be done through the Oak Ridge National Laboratory (ORNL). The PVRC then dissolved the Ad Hoc committee and formed a permanent Subcommittee to Develop Stress Indices for Piping, Pumps, and Valves. The subcommittee has the responsibility for coordinating the non-AEC portion of the work and for advising the code writing bodies. In addition, the subcommittee was asked to review, consult with, and advise ORNL in its program.

One of the suggestions given by the AEC in their letter of January 9, 1968, concerned the "desirability of a literature survey". While such a survey was implied to some extent in the PVRC program pp. 19-22 and Tasks 2 and 3, no specific task was assigned by PVRC. At the PVRC Ad Hoc Committee meeting on January 17, 1968, the committee agreed that a literature survey was a necessary first step and ORNL assumed the responsibility for developing the report. The "Survey Report on Structural Design of Piping Systems and Components" constitutes this first step.

A preliminary draft of the Survey Report was issued in November, 1968. Review and comments were solicited from members of the PVRC Subcommittee to Develop Stress Indices for Piping, Pumps, and Valves. Comments were received during the first seven months of 1969; these were incorporated in the survey report, along with additions to some of the chapters. The final version was completed in December, 1969.

The following submitted written comments

J. E. Corr, General Electric Co., Nuclear Energy Div.

H. H. George, Tube Turns

J. H. Griffin, Los Alamos Scientific Laboratory

D. F. Landers, Teledyne Materials Research

B. F. Langer, Westinghouse Electric Corp., Atomic Power Div.

C. F. Larson, Welding Research Council, PVRC

M. M. Lemcoe, Liquid Metal Engineering Center

M. V. Malkmus, Tube Turns

M. Pakstys, General Dynamics, Electric Boat Div.

B. J. Round, Combustion Engineering Co.

John Soehrens, C. F. Braun Co.

The authors would like to express their appreciation for the many valuable comments. Most of these are reflected in the Survey Report. Interpretations or opinions given in the Survey Report, however, should not be considered as other than those of the authors. We would also like to thank W. L. Greenstreet and S. E. Moore, of Oak Ridge National Laboratory, for assistance and advice in the preparation of the report.

## 1.2 Scope

The purpose of the Survey Report is to provide a summary of design practices, service experience, and research work on the structural design of piping components and systems, thereby providing a background and direction for future work. The report is restricted to the structural design aspect of metal piping systems. It does not cover such aspects as inspection, quality control, fabrication, deterioration of metals in service (except by fatigue and/or creep), fluid flow, etc.

## 1.3 Supplementary References

Since completion of the original draft of the Survey Report in November, 1968, a significant number of papers and/or research reports pertinent to the Report have become available. It was possible to incorporate only a few of these recent references during the revisions made in August/November, 1969. There are two recent publications which merit particular attention.

The first of these two publications consists of the "Design Guide for LMFBR Sodium Piping"(1.5), along with the background reports leading to the design guide. The Design Guide itself is in two volumes; Volume 1, "Requirements", is in the nature of a code for piping; Volume 2, "Procedures", gives suggested ways of implementing the rules given in Volume 1 and to amplify and explain those rules. The background reports are entitled:

| TECHNICAL REPORT NO. | | ISSUE DATE |
|---|---|---|
| 100 | The Development and Verification of a Design Guide for LMFBR Sodium Piping | 7-23-69(F)* |
| 110 | LMFBR System Requirements | 10-25-68 |
| 210 | A Study of Failure Theories as Related to LMFBR Piping Systems | 1-31-69 |
| 214 | A Review of Piping Failure Experience | 3-28-69(F) |
| 217 | A Review of Piping and Pressure Vessel Code Design Criteria | 4-18-69(F) |
| 220 | A Review of Fabrication and Installation Requirements for LMFBR Piping | 6-6-69(F) |
| 223 | A Study of Heating and Insulation Methods For LMFBR Sodium Piping | 2-28-69 |

* (F) Date of final issue. Other dates are for preliminary issue.

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| 228 | A Review of LMFBR Piping Materials | 4-3-69 |
| 231 | A Study of LMFBR System Interfaces | 3-28-69 |
| 234 | A Study of Scale Model Testing Methods Applicable to LMFBR Piping Design | 5-6-69 |
| 237 | A Study of Dynamic Analysis Methods As Related to LMFBR Piping Systems | 5-14-69(F) |
| 240 | A Study of Instability Analysis Methods As Related to LMFBR Piping Systems | 2-13-69 |
| 243 | A Review of In-Service Surveillance Methods Applicable to LMFBR Piping | 5-23-69 |

The LMFBR background reports listed above are in some aspects parallel to coverage in the Survey Report. However, the LMFBR reports are aimed at the problems of high-temperature piping and the specific characteristics of piping containing liquid sodium. The Survey Report, in contrast, is generally directed towards information pertinent to the design of present-day, water-cooled-reactor piping systems.

The second of these two publications is Pressure Vessel Technology[1.6]. This is a publication of the proceedings of the First International Conference on Pressure Vessel Technology, Delft, September, 1969. The Table of Contents of this two-volume publication is shown as Table 1.1 herein.

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# 1. REFERENCES

(1.1) USA Standard Code for Pressure Piping, Nuclear Power Piping, USAS B31.7-1969. Published by the American Society of Mechanical Engineers, 345 East 47th Street, New York, N. Y. 10017.

(1.2) ASME Standard Code for Pumps and Valves for Nuclear Power, Draft dated November, 1968, Published by the American Society of Mechanical Engineers, 345 East 47th Street, New York, N. Y. 10017.

(1.3) ASME Boiler and Pressure Vessel Code, Section III, Nuclear Vessels, 1968, Published by the American Society of Mechanical Engineers, 345 East 47th Street, New York, N. Y. 10017.

(1.4) "Program and Request for Proposals for Development of Stress Indices and Methods of Analysis for Piping Components, Valves and Pumps", Unnumbered PVRC report of the Ad Hoc Committee to Develop Stress Indices for Piping Pumps, and Valves, dated July 1, 1967.

(1.5) "Design Guide for LMFBR Piping", Preliminary Draft dated August 22, 1969. Prepared for U.S.A.E.C., by C. F. Braun Co., Alhambra, California.

(1.6) Pressure Vessel Technology, Part 1, Design and Analysis and Part 2, Materials and Fabrication, First International Conference on Pressure Vessel Technology, Delft, Holland, 1969. Published by the American Society of Mechanical Engineers, 345 East 47th Street, New York, N. Y. 10017.

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# 2. FACTORS INVOLVED IN STRUCTURAL DESIGN

A summary of factors involved in the design of piping components and systems is shown in Table 2.1; these are discussed in the following.

## 2.1 Design Requirements

The structural design requirements for a piping system can be simply stated. During the specified lifetime:

(a) The system shall not leak excessively.

(b) The system shall not deform to the extent that it is no longer functional.

(c) The system shall not impose loads on equipment attached to the piping system that would damage that equipment.

There is an equally important engineering requirement; i.e., the design requirements shall be met as economically as possible. Table 2.1 lists a few "failure examples". The term "rupture", as used herein, could include anything from a pinhole leak to a major tear and could be caused by a single load application or many loads; i.e., a fatigue failure.

## 2.2 Loads

In order to meet the design requirements, it is necessary to know the loads that will be applied to the piping system. Typical types of loads are listed in Table 2.1. The magnitude of loads, number of applications (fatigue) and duration of the loads (creep) are all significant aspects of the loads. In many piping systems, only the internal pressure is accurately known in the early design stage and estimates of other loads must be made.

## 2.3 Material Properties

The response of the material to the various loads applied to the system must be established. Table 2.1 lists typical material properties that are significant in piping design. Properties of the weld metal as well as base metal must be considered. While selection of a material, in some piping systems, depends upon its corrosion resistance, erosion resistance, resistance to radiation damage, etc., this report does not cover such considerations.

## 2.4 Analysis

The synthesis of design requirements, loads and material properties into an acceptable and economic piping system is considered herein as the product of analysis. Consideration of the system as a whole, as well as the components in the system, are included in the analysis. Broadly speaking, analysis methods may be classified as theoretical or experimental. Analytical methods are discussed in more detail in the next section of this report.

TABLE 2.1 FACTORS INVOLVED IN THE DESIGN OF PIPING COMPONENTS

---

<u>Design Requirements - Failure Examples</u>

Rupture due to:

- Single, short-time load (including brittle fracture)
- Repeated loads (fatigue)
- Long-time load at elevated temperature (creep-rupture)
- Combinations of the above

Excessive deformation, leading to:

- Valve seat leakage
- Valve mechanism jamming
- Flanged-joint leakage

Excessive loads on attached equipment, leading to

- Rupture of attached equipment
- Binding of bearings on attached equipment such as pumps, compressors, turbines
- Loss of clearance on rotating parts, with possible damage to those parts

<u>Loads</u>

| | |
|---|---|
| Internal pressure (operation and test)<br>Line expansion forces<br>Weight, wind<br>Thermal gradients<br>Vibration, shock<br>Bolt loads (flanged joints)<br>Stem loads (valves)<br>Pressure shock (water hammer) | What magnitude?<br>How often applied? |

<u>Material Properties</u>

| | |
|---|---|
| Modulus of elasticity<br>Poisson's ratio<br>Ultimate strength<br>Yield strength<br>Creep strength<br>Long-time rupture strength<br>Fatigue strength<br>Ductility<br>Toughness | At test temperature<br>At minimum operating temperature<br>At maximum operating temperature |

<u>Analysis</u>

| | |
|---|---|
| Theoretical<br>Experimental | Elastic behavior<br>Plastic behavior |

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CHAPTER 3

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# 3. ANALYTICAL METHODS

As indicated by Table 2.1 of Chapter 2, the structural analysis of piping components taxes to the fullest all of the tools of applied mechanics. The purpose of this chapter is to list certain theoretical developments and experimental techniques which have been applied to piping components in the past, or may be so applied in the near future.

While this chapter is subdivided into theoretical and experimental analyses, it is recognized that almost all theoretical analyses involve some empirically developed "laws"; similarly, most experimental analyses involve some "theory" in the sense that the results are interpreted by means usually classified as theoretical (e.g., conversion of measured strains to stresses).

## 3.1 Theoretical Analysis

Many of the more pertinent theoretical developments are referenced and, in some cases, discussed in Chapters 6 through 17 herein. The purpose of this section is to briefly outline the status of theoretical analysis tools particularly applicable to piping components; these tools consist, for the most part, of computer programs.

A complete set of references to the many computer programs which have been and are being developed is beyond the scope of this report. Additional references are given by the six papers included in a recent ASME publication, Use of the Computer in Pressure Vessel Analysis[3.1]. The listings of computer programs given in the following subsections (3.11 on axisymmetric components, 3.12 on general components, 3.13 on specific components, and 3.14 on piping system analysis) should be considered as examples of existing computer programs. No implication is intended that the programs cited are better than other programs not included.

Three agencies which are sources of computer programs are:

(1) Air Force Flight Dynamics Laboratory, Wright-Patterson Air Force Base, Ohio.

(2) COSMIC Computer Center, University of Georgia, Athens, Georgia.

(3) Argonne Reactor Code Center, Argonne National Laboratory, Argonne, Illinois.

## 3.11 Axisymmetric Components

There are a number of piping components which can be classified as geometrically axisymmetric; e.g., straight pipe, concentric reducers, closures, radial nozzles in closures, and bolted-flanged joints. The theory for axisymmetric shells, both with axisymmetric and asymmetric loads, is relatively well developed. A number of computer programs applicable to axisymmetric components have been developed, some of which are listed and briefly discussed below.

(1)* AXISOL[3.2]: Applicable to bodies of revolution, subjected to symmetric mechanical or thermal loads. Employs finite elements (rings); generates and inverts a stiffness matrix.

(2)* BASIC[3.3]: Applicable to bodies of revolution, subjected to symmetric mechanical or thermal loads. Employs point-matching techniques, including both spherical and toroidal stress functions.

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* These are acronyms used for computer programs at Battelle-Columbus. Similar programs based on the same reference may be available under different acronyms.

(3) DUZ-1[3.4]: Applicable to bodies of revolution, subjected to symmetric mechanical or thermal loads. Employs finite element techniques.

(4)* MOLSA[3.5]: Applicable to multilayer, orthotropic shells of revolution, subjected to either axisymmetric or non-symmetric mechanical and thermal loads. Employs numerical (Runge-Kutta) integration of shell equations over appropriate shell length.

(5)* NONLIN[3.6]: An extension of MOLSA to include elastic non-linear effects.

(6) SAFE-PCRS[3.7]: Applicable to composite bodies of revolution, subjected to symmetric mechanical or thermal loads. Employs finite elements.

(7) SEAL-SHELL-2[3.8]: Applicable to shells of revolution, subjected to symmetrical mechanical or thermal loads. Employs strain-energy to obtain stiffness matrix, includes thick-shell effects.

(8) SHOREF[3.9]: An adaption of MOLSA to determine natural frequencies.

All of the above except (5) apply to the linear, elastic regime. Additional developments related to finite-element approaches are given in References (3.10) through (3.15). Two recent text books on the finite-element methods are by Przemienicki[3.16] and by Zienkiewicz and Cheung[3.17].

The analysis of bodies of revolution (including not-too-thin shells of revolution) in the elastic-plastic range is contained in a

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* See footnote on p. 3-2.

computer program "FEELAP" by Marcal[3.18]. This program is analogous to AXISOL[3.2], extended into the plastic range. It employs the octahedral shear stress yield criteria, and an arbitrary (non-linear) material stress-strain relationship may be used in increments. For thin-wall shells, the computer program "NONLEP" developed at Battelle-Columbus may be more appropriate. This program is an extension of "NONLIN" (Reference (3.6)) into the plastic range and also uses the octahedral shear stress yield criteria and an incremental stress-strain relationship.

Extension of the elastic-plastic range analyses into the creep regime is a relatively easy step; the strain-load dependence used in the elastic-plastic regime is replaced by the strain-time dependence in the creep regime.* Greenbaum, et. al.[3.19] have prepared such a creep program, using finite-element methods. A similar program has also been completed at Battelle-Columbus, using the program FEELAP as a basis.

The limit loads of a shell of revolution can be obtained by the computer program "CLPSHL"[3.20]. The analysis is based on the Nakamura[3.21] approximation to the Tresca yield criterion and gives an "exact" (not an upper or lower bound) solution for axisymmetric mechanical loadings.

## 3.12 General Components

There are many piping components which are not axisymmetric in geometry; e.g., curved pipe, eccentric reducers, nozzles in cylinders, tees, and valve bodies. Certain theoretical developments specifically applicable to curved pipe and to cylinder-to-cylinder branch connections are listed in the following subsection. Aside from these, the theoretical analyses of

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* A specific example of the analogy between plastic analysis and creep analysis is given in Chapter 6, Paragraph 6.22.

such non-symmetric components pose as yet unsolved problems. There are, at this time, a number of developments underway which may provide adequate tools in this area; these are briefly discussed below.

The present trend in analysis of complex structures involves the use of finite elements. One of the earliest formulations of this approach is given by Hrennikoff[3.22] in 1939. Because this approach requires the solution of a large number of simultaneous equations, the method was not used much until large-capacity, high-speed computers became generally available. In the past decade, particularly in the aircraft industry, the finite-element methods have undergone intensive development.

Up to the present time, the finite-element approaches have not been used to any significant extent for the analysis of piping components. It might be noted that a certain degree of skill is necessary in selecting suitable size and types of elements in order to obtain accurate results, particularly accurate stresses in areas of rapidly varying stress. Selecting and describing (for input data) an appropriate set of elements may involve a significant amount of labor. Further, even with the best present-day computers, the running time may be measured in hours. However, improvements along these lines may be expected in the near future and these types of computer programs may prove quite useful in the analysis of piping components.

Some examples of existing computer programs applicable to non-axisymmetric components are listed below.

(1)* CSMTRX[3.23]: Straight or curved beam elements.

(2) ELAS[3.24]: Solid elements, plate elements, beam elements (18 element types).

* See footnote on p. 3-2.

(3) FORMAT II[3.25]: Beam and/or panel elements. Provides capability for formation and manipulation of large matrices.

(4) GENSAM[3.26]: Tetrahedral elements.

(5) PAPA[3.27]: Curved, trapezoidal or triangular plate or panel elements.

(6) SAFE-3D[3.28]: (Description unavailable).

(7) SAMIS[3.29]: Beams and/or triangular plate elements. Provides capability for formation and manipulation of large matrices. Can calculate natural frequencies and mode shapes. Recent modifications include buckling subroutine.

In addition to the programs listed above, attention should be drawn to the work of Clough and his co-workers in the field of finite elements. The latest reference, by Clough and Johnson[3.30], deals with the analysis of thin shells using finite elements consisting of flat, triangular plates.

The finite element approach has been applied to vibration analysis; see, for example, Reference (3.31). In principle, the finite element technique could be extended into the elastic-non-linear, plastic and creep regimes as has been done for axisymmetric structures. It is understood that work is under way of the Jet Propulsion Laboratory to extend ELAS into the elastic-plastic regime.

## 3.13 Specific Components

In the preceding section, some indication of the availability of general purpose computer programs is given. In addition, a number of computer programs have been developed for application to relatively specific configurations. These kinds of programs are useful in that input data are simple and computer running times short, compared to the general purpose programs. Accordingly, such special purpose programs for specific components serve a useful function, particularly if a parametric study for development of design curves or graphs is desired.

(1) Radial, cylindrical nozzles in cylindrical vessels[3.32]

Based on shell theory with boundary point matching. Gives stresses due to internal pressure loading. Limited to $d/D \leq 1/3$, $(d/D)\sqrt{D/T} < 1.1$, where d = nozzle diameter, D = vessel diameter, T = vessel wall thickness.

(Work is underway to extend the analysis and develop a computer program for out-of-plane bending moment applied to the nozzle.)

(2) Curved Pipe[3.33]

Based on shell theory using minimized energy to develop a series solution. Loadings consist of either in-plane or out-of-plane bending moments, including the effect of internal pressure on stress and displacements due to those moment loads. Does not include "end-effects".

(3) Curved Pipe[3.34]

Based on shell theory using numerical (Runge-Kutta) integration in two directions. Loadings include in-plane or out-of-plane mements and internal pressure but not the interaction between pressure and moments. Includes "end-effects".

(4) Local Loads on Shells[3.35]

Based on shell theory. For spherical shells, employs a closed form solution based on Bessel-Kelvin functions. For cylindrical shell, employs a double Fourier series solution. Loadings include distributed loads such that the resultant is (a) a radial force, (b) in-plane moment, (c) out-of-plane moment, (d) shear force.

(5) Nozzles in Spheres

(a) Waters[3.36]

Based on shallow-shell theory but includes certain thick-wall aspects. Internal pressure loading only.

(b) CERL [3.37]

Based on shell theory. Loadings include internal pressure, thrust on nozzle, moment on nozzle, and shear force on nozzle.

(c) Bijlaard [3.35]

Based on shallow-shell theory. Loadings same as (b) above except for internal pressure.

(6) Tapered-wall transition joints in cylinders[3.38]

Based on shell theory with solution obtained in terms of Bessel-Kelvin functions. Internal pressure loading only. (Work is underway to extend the solution to tapered wall transitions between cylinders and spherical heads and to include thermal gradient stresses.)

(7) Bolted-Flanged Joints [3.39]

Based on shell and plate theory. Includes the ASME design method but also includes internal pressure and thermal gradient loadings and gives the variation in bolt stress as a function of these loads.

In addition to the computer programs listed above, theoretical analysis methods exist for other aspects of piping componnd dessign, particularly for straight pipe. It is pertinent to list some of the theoretical developments for straight (uniform wall, circular cross section) pipe.

(1) Elastic behavior under combinations of internal or external pressure, moment, torsion, and axial loads and thermal gradients.

(2) Internal pressure loading, yield pressure, and post yield behavior up to and including burst (maximum, instability) pressure.

(3) Internal pressure loading, behavior under creep conditions.

(4) Combinations of internal pressure, moment, torsion and axial thrust. Limit load combinations. Post yield behavior considering strain hardening. Behavior under creep conditions.

(5) External pressure loading, elastic or elastic-plastic buckling.

(6) Buckling as a beam or as a shell; natural frequencies and response to given forcing loads and damping.

As an ideal, but far-distant goal, an equally complete theory for other components would be valuable. For example, out-of-round pipe, curved pipe, branch connections, etc. The status of the theory for these and other components is discussed in Chapters 6 through 17.

## 3.14 Piping System Analysis*

The purpose of a piping system analysis is twofold:

(1) To check whether any of the piping components of the system are overloaded (over-stressed), either statically or cyclically (fatigue conditions).

(2) To check whether the connected equipment is overloaded. (For compressors, pumps, valves, etc., excessive forces may impair functioning.)

The loads normally considered are: (a) weight of piping components, contained fluid and insulation, (b) wind and/or earthquake, (c) movements of connected equipment, and (d) change in pipe length due to temperature change.

The piping system analysis is dependent upon the layout and selection of supporting elements (see Chapter 15) and, in turn, produces information needed in the design of the supporting elements; i.e., loads and displacements. Permissible stress ranges are given in USAS Piping Codes[3.40]. Permissible loads on attached equipment must also be established. Reference (3.41) gives some guidance for piping loads transmitted to steam turbines. USAS piping codes, in particular B31.1.0-1967, simply state that

> "The reactions computed shall not exceed limits which the attached equipment can safely sustain."

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* The analysis of a piping system is traditionally called a "piping flexibility analysis" or a "flexibility analysis". We have chosen the term "piping system analysis" because:

  (a) This nomenclature indicates that the analysis is applicable to a piping system as distinct from the analysis of some component part of the system.

  (b) Structural analysis of the type involved may be based on either a "flexibility approach" or a "stiffness approach". The piping system analyses methods are not restricted to either approach.

  (c) The piping system analysis is generally used to establish not only force/deflection relationships but also end-loads and stresses.

An extensive review of the subject of piping system analysis is given by Brock(3.42), who includes 265 related references. We will, in the following, touch on only a few aspects of piping system analysis; these and other aspects are covered in considerable detail by Brock(3.42).

Up to the present time, piping system analyses have been made on an elastic basis. The piping system is modeled as an assemblage of straight and curved beams, with appropriate restraints or motions at anchors, guides, hangers, connections to hangers, etc. Most piping system analyses include flexibility and stress-intensification factors for curved pipe; the derivation of these is discussed in Chapter 7. Flexibility factors for other components are not used, although in the case of small branch connections, the contribution of local deformations may significantly alter the deformation of the piping system (see Chapter 8).

The analytical solution of the general problem of a three-dimensional piping system with two or more anchor points, while basically simple, involves a large amount of computations and careful "bookkeeping". Prior to about 1950, considerable effort was devoted to the development of "simplified solutions", which necessarily introduce a certain degree of approximations. Examples of these simplified solutions are given in References (3.43), (3.44), and (3.45). The advent of high-speed digital computers and associated computer programs, however, has to a large extent eliminated the need for simplified solutions.

Brock(3.42) gives the historical background to the development of computer programs for piping system analysis. At the present time a multitude of such programs exist. These programs encompass various capacities with respect to number of restraint points, number of branches, and number of loops.

Apparently, one of the most widely used programs is that designated as MEC-21. It was originally developed by John Olson and Robert Cramer of the Mare Island Naval Shipyard in 1959. Revisions and expansions were made in 1963 (MEC-21/704) and in 1964 (MEC-21/7094). The latest version, written for an IBM-7094*, is described by Griffin[3.46]. The maximum problem size is 99 branches, 99 branch-intersection-points, and/or 999 data points. Each data point may describe one to three elements. Machine time (IBM-7094) varies between 0.02 and 0.05 minutes per element, depending upon the complexity of the piping system. The report by Griffin[3.46] covers the application of the program and serves as an instruction manual for the user. This program is available from the Los Alamos Scientific Laboratory, Los Alamos, New Mexico 87544.

The program PIPE[3.47] is available from Argonne National Laboratory. It is described as encompassing structures, the elements of which are straight or curved, and rigid or elastic. The structure may have branches, loops, and rigid or flexible anchors. Limitations are: 100 nodes, 20 loops, 25 external loads, 50 redundants, 10 sets of material properties.

Another widely used program is provided on a commercial basis by the Service Bureau Corporation. Program scope limitations may be obtained by contacting the Service Bureau Corporation.

Many companies dealing with piping systems have their own computer programs; e.g., Bechtel Corporation, C. F. Braun Co., Electric Boat Division of General Dynamics, Esso Research and Engineering, Fluor Corporation, and M. W. Kellogg Co. Some of these will perform analyses for others on a commercial basis.

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* This program was converted at Battelle-Columbus for use with a Control-Data 6400 computer. The conversion required only about one day of programmer time and involved the change of only a few cards.

It is generally recognized that for piping systems operating at temperatures in the creep range, the forces and moments obtained from the elastic-theory piping system analysis due to either movements of attached equipment or change in pipe length from temperature change are not actually present in the piping system except possibly for a short time after initial start-up. The forces and moments decrease as a function of time because of the plastic flow due to creep. At shut-down and return to atmospheric temperature, some part of these elastically-calculated forces and moments (with reversed signs) will be present because of the permanent plastic flow. The elastic analysis does give bounds on forces and moments applied to attached equipment; accordingly, even in the creep range, the elastic analysis does give the desired information in this respect.

While the elastic analysis gives the range of loads applied to the piping system, under creep conditions the range of strains encountered at some locations may be grossly underestimated by the elastic analysis. This possibility arises where a relatively small portion of the piping system has a higher stress, or is at a higher temperature than the remainder of the system. Some aspects of this problem are discussed by Robinson[3.48].

While the problem of strain concentration in a piping system is usually discussed in relationship to creep, an analogous condition arises when stresses are permitted to exceed the yield strength of the material. In the American Standard Code for Pressure Piping, expansion stresses are permitted to exceed yield strength. The problem is recognized in the Piping Code and precautionary guides are given. The following excerpt from USAS B31.1[3.49] is typical:

"119.3 Local Overstrain

All the commonly used methods of piping flexibility analysis assume elastic behavior of the entire piping system. This assumption is sufficiently accurate for systems where plastic straining occurs at many points or over relatively wide regions, but fails to reflect the actual strain distribution in unbalanced systems where only a small portion of the piping undergoes plastic strain, or where, in piping operating in the creep range, the strain distribution is very uneven. In these cases, the weaker or higher stressed portions will be subjected to strain concentrations due to elastic follow-up of the stiffer or lower stressed portions. Unbalance can be produced:

(a) by use of small pipe runs in series with larger or stiffer pipe, with the small lines relatively highly stressed,

(b) by local reduction in size or cross section, or local use of a weaker material, or

(c) in a system of uniform size, by use of a line configuration for which the neutral axis or thrust line is situated close to the major portion of the line itself, with only a very small offset portion of the line absorbing most of the expansion strain.

Conditions of this type should preferably be avoided, particularly where materials of relatively low ductility are used: if unavoidable, they may be mitigated by the judicious application of cold spring.

It is recommended that the design of piping systems of austenitic steel materials be approached with greater over-all care as to general elimination of local stress raisers, inspection, material selection, fabrication quality and erection."

"Cold Spring" is sometimes used in critical piping systems, particularly those operating at high temperatures. During installation the pipe is cut short by some percentage of the calculated thermal length change of the piping system. The pipe is then "sprung" into position. For 100 percent cold spring, the piping system would theoretically have no forces or moments due to thermal expansion when the pipe reaches its operating temperature. The advantage is that the connected equipment (pumps, turbines, etc.) is better able to withstand the forces and moments when cold than when hot. With 50 percent cold spring, the maximum load at operating temperature would be one-half of that with no cold spring (assuming modulus of elasticity change with temperature is negligible). However, the range of forces, moments and stresses is not affected by the amount of cold spring for a piping system operating in the creep range. In this case, relaxation will tend to produce the equivalent of cold-spring. Hence, after a period of time the loading conditions may not depend greatly on whether the piping system was originally cold sprung or not. The piping system analysis is used to establish the dimensional requirements needed for a given percentage of cold springing and the loading conditions arising therefrom.

## 3.15 Material-Strain Relationships

In the preceding sections methods for calculating the strains or stresses in, and displacements of, piping components subjected to various loads have been discussed. For the most part, the designer is not interested in these quantities, per se, but rather uses these quantities as a guide to whether the component will fail in service. Traditionally, an allowable stress has been established based on the yield strength, ultimate tensile strength, creep strength or creep-rupture strength; all as established on the basis of tensile tests. It is generally recognized that the stress-strain conditions in a piping component are not necessarily indicated directly by those in a tensile test. The basic problem is to obtain correlations between properties of materials as given by simple, inexpensive tests and the behavior of these materials when used to construct a complex structure under complex loadings. These kinds of correlations may be generally classed as "Failure Theories". The literature on this subject is very extensive and cannot be covered herein. Many books are available either entirely on the subject or with chapters on the subject; for example, see Nadai[3.50] on plasticity, Finnie and Heller[3.51] on creep, and Grover, Gordon, and Jackson[3.52] on fatigue. Some general observations on the status of failure theories are made in the following three sections.

### 3.151 Combined Stresses

Numerous experimental investigations have been conducted in an effort to answer the question: What is the relationship between material behavior under a three-dimensional stress field and the material behavior under a uniaxial stress? The question has been investigated for (a) onset of plasticity and flow directions, (b) creep rate and flow directions, and (c) fatigue*. For isotropic ductile materials, the answer to this question for all three phenomena usually is that the octahedral shear stress theory agrees best with test data; the maximum shear stress theory is usually slightly conservative with respect to the test data. These theories are expressed by the relationships:

Octahedral Shear Stress

$$\sigma_o = \frac{1}{\sqrt{2}}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right]^{1/2} \qquad (3.1)$$

Maximum Shear Stress

$$\sigma_o = \text{maximum of } |\sigma_1 - \sigma_2|,\ |\sigma_2 - \sigma_3|,\ |\sigma_3 - \sigma_1| \qquad (3.2)$$

where $\sigma_o$ = equivalent tensile stress

$\sigma_1, \sigma_2, \sigma_3$ = principal stresses

3.152 Fatigue

Material properties with respect to fatigue are usually available in the form of cycles to failure for some given stress or strain cycle. The stress or strain cycle is constant during the test and usually the cycle is either completely reversed ($\sigma_{min}/\sigma_{max} = -1$) as in a rotating beam test or varies from $\sigma_{min} = 0$ to $\sigma_{max}$ as in a tensile fatigue test. The designer is interested in correlations for some particular structure and loadings in which the ratio of $\sigma_{max}/\sigma_{min}$ may be different than directly available from material tests and further, there may be a number of different stress cycles imposed on the material in the structure during

* There may be a difference in failure criteria for fatigue crack initiation as opposed to crack propagation.

its lifetime. While there are many gaps and contradictory evidence and opinions on these aspects, designers have found the Goodman diagram and its modifications useful in assessing the effect of mean stress on fatigue life and that Minor's hypothesis as useful in assessing cumulative damage due to a variety of stress cycles. These correlations are discussed in Reference (3.52).

3.153 Creep and Fatigue

In higher temperature piping systems, the design may involve both creep and cyclic loads. At present, little in the way of generally applicable correlation methods are available. Some of the basic aspects of the problem are discussed by Coffin[3.53] and Benham[3.54].

## 3.16 Fracture Mechanics*

The significance of fracture mechanics approaches derives from the hypothesis that there may be cracks (flaws or defects) in components. The approach is then directed toward supplying quantitative information on questions such as:

(1) What are the critical crack sizes (i.e., sizes required to cause failure) in the various portions of a component at the expected test and/or operational stress levels?

(2) Will cracks, initially present but below critical size, grow to critical size and cause failure during the expected service life of the component?

(3) If a critical crack size does exist, will the resulting failure be relatively small in extent, causing a leak,

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* The alternate design approach involving material selection on the basis of "toughness" tests is briefly described in Par. 3.23.

or will the fracture propagate over a large area, causing a major break in the component?

It might be noted that these three questions involve three aspects of fracture mechanics:

(1) Crack-growth initiation

(2) Crack propagation

(3) Crack propagation arrest.

At this point, it is pertinent to quote an extract from Strawley and Brown[3.56]:

"Because of its rapid development over the last decade or so, fracture mechanics has seemed confusing to many interested parties (and we do not exclude ourselves)."

Nevertheless, at least a few phases of fracture mechanics as related to the structural design of piping components will be discussed herein. ASTM publications STP 380, STP 381, and STP 410 (References (3.57), (3.58), and (3.59)) give much of the background involved and the papers therein provide several hundred references. One of the latest compilation of papers, along with many recent references, is contained in the 1969 International Conference on Fracture[3.60]. The seven-volume treatise on fracture, edited by Liebowitz[3.61] is a major source of reference in this area.

Some pertinent Welding Research Council Bulletins are listed as References (3.62) through (3.73). References (3.62) and (3.63) give some historical background on "brittle fracture" types of service failures which gave considerable impetus to the study of fracture mechanics. A discussion of fracture mechanics as applied to the specific problem of the fracture behavior of defects in pipe is given in Chapter 6, Par. 6.56.

3.161 Linear Elastic (Plane Strain) Fracture Mechanics

The linear elastic fracture mechanics approach entails a stress intensity factor*, K, and a critical stress intensity factor, $K_c$. The magnitude of K is dependent upon the geometry of the body containing the crack (flaw, defect), the size and location of the crack, and the distribution and magnitude of the external loads on the body. The value of K can be calculated, at least in the elastic regime and for some relatively regular crack shapes and distribution of nominal stresses. It is related to material properties only through the modulus of elasticity and Poisson's ratio. The general form is:

$$K = \alpha \sigma \sqrt{a} \tag{3.3}$$

where

$\alpha$ = a dimensionless constant determined by the body geometry and crack shape

$\sigma$ = nominal stress at the crack fronts

a = a significant dimension of the crack

The form of Equation (3.3) shows that K has the dimensions of psi - $\sqrt{\text{in}}$.

The criterion for brittle fracture in the presence of a crack-like flaw or defect is that crack growth (instability) will initiate when the crack-tip stresses exceed some critical condition. For the opening mode (I) of loading (tensile stresses perpendicular to the major plane of the flaw) under brittle plane-strain conditions (limited crack-tip plasticity) the critical stress intensity factor for fracture instability is designated as

* The term "stress intensity factor" or "stress intensity", as used in the field of fracture mechanics, must not be confused with the term "stress intensity" as used in recent pressure vessel and piping codes. In the latter case, stress intensity is defined as twice the maximum shear stress.

The subscript I is used to designate the opening mode of crack displacement. Subscripts II and III are used to designate shear modes of displacement. Use of K without a subscript denotes generality as to displacement mode.

$K_{Ic}$. The value of $K_{Ic}$ can be considered as a material property which represents the material's resistance to failure in the presence of a crack or crack-like defect.

The fracture mechanics' answer to the question of critical crack sizes is then: if the appropriate stress intensity factor expression is known for the specific component, crack location, crack shape and loadings, and if the $K_c$ for the material is available, it is possible to establish the maximum allowable flaw size by use of the criterion:

$$K < K_c \tag{3.4}$$

The analogy of the fracture mechanics approach to traditional design methods should be noted. Traditionally a value of $\sigma_m$ is calculated which, like K, depends upon the geometry of the component, the loadings and the modulus of elasticity and Poisson's ratio of the material, where $\sigma_m$ is the maximum stress at the point on the component under consideration. The value of $\sigma_m$ is then compared with an allowable stress* $S_A$ which, like $K_c$, is determined from a test (usually a tensile test) on the material and is a material property. The design is considered acceptable if $\sigma_m < S_A$; analogous to the fracture mechanics $K < K_c$.

It should be noted that use of linear elastic fracture mechanics does not eliminate the need for the kinds of stress analysis tools discussed in Paragraphs 3.11, 3.12, and 3.13; K is dependent upon the value of $\sigma$. If, for example, a crack exists (or is postulated to exist) at the inside corner of a nozzle in a pressure vessel, then an appropriate value of $\sigma$ would be that calculated as existing in that general area.

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* The value of $S_A$ may, of course, be different depending upon the type of stress, $\sigma_m$, involved.

The linear elastic fracture mechanics' answers to the question of crack propagation appear to be less clearly established. There are two aspects of crack-growth propagation:

(1) Growth of a crack under a constant nominal stress. This kind of phenomena is evidenced in hydrostatic tests in which failure occurs several hours after the initial application (and subsequent holding) of the test pressure.

(2) Growth of a crack under a variable nominal stress. This is, of course, the problem of the rate of growth of a fatigue crack. Recent investigations [e.g., References (3.74), (3.75), (3.76), and (3.77)] have made correlations of fatigue-crack growth rate with the range or amplitude of K. These results indicate that such correlations may be possible over a significant range of different materials. A rather consistent relationship emerging from the tests of crack-growth rate in cylindrical shells with internal pressure loading is:

$$\frac{da}{dN} = \alpha (\Delta K)^n \qquad (3.4)$$

where da = crack growth increment

dN = number of cycles increment

$\alpha$ = proportionality constant

$(\Delta K)$ = range of the applied stress intensity

n = $\sim 4$.

References (3.74) and (3.75) give similar correlations for crack-growth rate in plates. The exponent of K still appears to be about 4, but both references note a shift in the data; that is, the growth rate for shells seems to be higher than for plates. Reference (3.75) notes that a shift by a factor of ten in growth rate brings their plate data more-or-less into agreement with their shell data. Reference (3.76) suggests further correlations by the relationship:

$$\frac{da}{dN} = \beta \frac{(\Delta K)^4}{K_c} \tag{3.5}$$

and

$$\frac{da}{dN} = \gamma \frac{(\Delta K)^4}{K_c^{\,2} S_y^{\,2}} \tag{3.6}$$

where $\beta$ and $\gamma$ = proportionality constants

$S_y$ = material yield strength.

The relationship between Equations (3.5) and (3.6) imply that $S_y$ is proportional to $1/\sqrt{K_c}$ . Discussions of this paper, and the author's reply, suggest that the specific value of the exponent of $\Delta K$, as well as the functional relationships of Equations (3.5) and (3.6), may not be valid over a wide range of materials and magnitudes of $\Delta K$. For example, Reference (3.77) indicates n-values of 2.2 for A533-B steel; 3.0 for A216 cast steel.

It should be noted that environment may play a significant role in the rate of crack growth.

The linear elastic fracture mechanics' answers to the question concerning crack propagation arrest seems to be in a tentative stage at the present time. A general discussion of the problem is given by Bluhm[3.78]. Conditions for crack propagation arrest are complex because of the dynamic effects involved and because the energy stored in the component appears to have a marked effect on the extent of crack propagation.

Tests[3.73, 3.79, 3.80] on cylindrical vessels with high energy content (air, mixtures of air and water, natural gas, superheated water used as pressurizing media) have indicated the possibility of long fractures under conditions of stresses, crack lengths and material properties at which only a short crack extension might be expected for the same test except using water as the pressurizing media. Some of these tests were beyond the realm of linear elastic fracture mechanics because, at least in some parts of the crack propagation stage, significant plastic effects occurred.

Robertson[3.81] developed a test in which an initiated crack arrested as it progressed through a test piece with a controlled thermal gradient. The temperature in the test piece at which the crack stopped has been designated as the crack arrest temperature. This type of information seems applicable to structures in which the stored energy is relatively small but may be inapplicable to piping or pressure vessel components with a large amount of stored energy in the pressurizing fluid.

3.162 Limitations to, and Extensions of, Linear Elastic Fracture Mechanics

Direct application of linear elastic fracture mechanics to typical pressure vessels and piping is limited because of the occurrence of significant plastic zones at the crack tip for most materials at temperatures of interest. Some indication of the thickness restraint necessary to insure negligible small plasticity effects is given by the recommended thicknesses[3.59] for valid $K_{Ic}$ tests:

$$T \geqq 2.5 \left(\frac{K_{Ic}}{S_y}\right)^2 \qquad (3.7)$$

where

$T$ = test specimen thickness

$K_{Ic}$ = expected value critical stress intensity factor

$S_y$ = yield strength.

If Equation (3.7) is applied to a material/temperature such that the expected value of $K_{Ic}/S_y$ = 2.0 (e.g., ASTM A533, Grade B, Class I steel at 50 F, $S_y \cong$ 70,000 psi), then the required test specimen thickness is 10". Determination of a valid $K_{Ic}$ for a typical piping material such as A106 Gr B, at temperatures of 70 F or higher, would seem to be a hopeless task under the restraints of Equation (3.7). Accordingly, the <u>direct</u> application of linear elastic fracture mechanics seems to be limited to material/temperature combinations where $(K_{Ic}/S_y)$ is less than about unity. For ferrous alloys at room temperature, this appears to limit applicability to those alloys which have yield strength of about 160,000 psi or higher. Such alloys are being used in the construction of pressure vessels for aerospace and other weight-critical applications.

Actually, linear-elastic fracture mechanics may have a much broader application than implied by the above discussion, centered about Equation (3.7). The necessary conditions are that the crack initiate with an insignificant amount of plasticity at the crack tip. Numerous "brittle failures" have occurred in pressure containing components in which, at least on a macroscopic scale, the crack initiated at a notch or defect with no apparent plastic deformation. These have occurred in structures with much smaller thicknesses than implied by Equation (3.7).

The concepts of linear-elastic fracture mechanics have been and are being extended well beyond the range of purely plane strain (insignificant plasticity). Some of these efforts (e.g., References (3.82),(3.83), and (3.84)) are directed towards inclusion of the effect of the plastic zone size as a correction to the linear-elastic theory for calculation of K. Other efforts are directed at a direct evaluation of the effect of a known size crack in a component where the requirements for plane strain are not necessarily met. Data on cracks in cylindrical vessels or pipe is discussed in Chapter 6, Par. 6.56.

Another significant approach is that suggested by Pellini and his co-workers (References 3.64, 3.70, 3.71, 3.72, 3.73). First Pellini points out that "valid" $K_{Ic}$ tests are essentially limited to tests at or below the nil-ductility temperature. He suggests the use of the dynamic tear test for extending the temperature range above the nil-ductility temperature and indicates correlations between the dynamic tear tests and the $K_{Ic}$ tests for temperatures below the nil-ductility temperature. Critical crack sizes in the temperature range up to the nil-ductility temperature were originally based on service data. Later, in Reference (3.70), Pellini shows that the crack lengths are in reasonable agreement with those derived from linear-elastic fracture mechanics for the particular conditions of thicknesses from 1 to 3", $(K_{Ic}/S_y)$ of 1.2 for the static case, and for steels of less than 150,000 psi yield strength $(S_y)$. Later, in Reference (3.73), the fracture analysis diagram is modified to indicate the effects of large thicknesses and to illustrate the behavior of cylindrical vessels with flaws tested with internal pressure.

## 3.2 Experimental Analysis

In principle, the suitability of any particular piping component can be established experimentally by simply applying those sets of loads which the component will endure in service to that component and observe if the component does satisfactorily withstand those loads. Where the only significant loading is static, and it can be reasonably assumed that time in service does not alter the material properties, such a test is often feasible and convincing. The hydrostatic tests and hydrostatic proof tests prescribed in the ASME Boiler Code[3.85] may be considered as an example of such a test philosophy. Similar tests form the basis for pressure ratings of butt-welding pipe components made to USAS B16.9 [3.86].

However, piping components are usually subjected to a variety of loadings; internal pressure, bending moments, torsion, thermal gradients, etc. The loads may be cyclic to the extent that fatigue failure must be considered. Further, operating temperatures may be sufficiently high so that a short-time test does not necessarily indicate the long-time load capacity of the component. It is often necessary, therefore, to separate the various loadings and conditions in order to obtain generally applicable test data.

In the following, a few of the most commonly used experimental analysis methods will be briefly discussed. These are divided into: (a) direct methods in which failure is apparent by direct observation such as a crack through the wall of the component or gross plastic deformation, and (b) indirect methods in which strains in the component are determined as a function of load. For indirect methods, the basic hypothesis is that the strains so determined can be correlated with material properties to indicate necessary load limitations to avoid failures.

## 3.21 Direct Test Methods

### 3.211 Burst Tests

The burst test, for internal pressure loading, is an acceptable method of establishing design adequacy in the ASME Boiler and Pressure Vessel Code(3.85)* except under Section III and Section VIII, Division 2. Roughly, the maximum allowable working pressure is one-fifth of the actual burst pressure of the prototype vessel, adjusted for the prototype material actual tensile strength as compared to the minimum specified tensile strength for the material; and adjusted for higher (than test) operating temperature by the ratio of allowable stress at operating temperature to allowable stress at test temperature.

The burst test is also used as a basis for design in USAS B16.9, "Wrought Steel Butt Welding Fittings"(3.86). The prototype fitting must be welded to straight pipe "legs". The pipe legs must be at least two-diameters in length and wall thickness equal to the designated wall of the fitting. The burst pressure must be at least equal to:

$$P = \frac{2St}{D} \qquad (3.8)$$

where

$P$ = required minimum bursting pressure

$S$ = minimum specified tensile strength of designated fitting material

* Par. A-22 of Section I, Par. UG-101 of Section VIII. The proof test are permitted only for parts for which design rules are not given in the Code.

t = minimum (87-1/2% of nominal) wall thickness of pipe with which the fitting is recommended for use

D = outside diameter of pipe.

USAS B16.9 fittings are not necessarily geometrically similar for various sizes and nominal wall thicknesses. The standard does not give any guidance as to what range of sizes and wall thicknesses should be tested in order to adequately check the entire range of fittings. The test data are considered proprietary by the manufacturers, hence the extent of testing (if any) by the various manufacturers is not known.

### 3.212 Yield Tests

The yield test, for internal pressure loading, is also an acceptable method of establishing design adequacy in the ASME Boiler and Vessel Code.* This method is applicable only to materials with $S_y \leq 0.625 S_u$; $S_y$ = minimum specified yield strength, $S_u$ = minimum specified ultimate strength. Roughly, the maximum allowable working pressure is one half of the test yield pressure, with adjustments for the prototype vessel material actual yield strength as compared to the minimum specified yield strength for the material; and adjusted for higher (than test) temperature by the ratio of allowable stress at operating temperature to allowable stress at test temperature.

Unlike the burst test, which has a well defined end-pressure, the definition of yield pressure is more difficult. In Section VIII, Division 1, three types of yield tests are permitted:

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* See footnote on page 3-28.

(1) Brittle-Coating Test Procedure. The part is coated with a lime-wash or other brittle coating. Pressure is increased until yielding occurs as evidenced by flaking of the brittle coating, or by the appearance of strain lines. (See Section 3.221 for further discussion of brittle coatings.)

(2) Strain Measurement. Strain gages are placed at the most highly stressed points*. Pressure is then increased until the most highly strained gage reaches a value of

0.2% permanent strain for aluminum-base and nickel-base alloys and for carbon, low-alloy and high-alloy steels.

0.5% strain under pressure for copper-base alloys. (The use of strain gages is discussed in Section 3.222.)

(3) Displacement Measurement. Displacements are measured at the most highly stressed parts by means of devices capable of measuring to 0.001". Pressures are applied incrementally and released. Plots of pressure vs displacement and pressure vs permanent displacement after release of pressure, are constructed. The yield pressure is taken as that pressure at which the curve representing displacements under pressure deviates from a straight line and/or that pressure at which the permanent displacements begin to increase regularly with further increase in pressure.

* As a check that the measurements are being taken on the most critical areas, the Inspector may require a lime wash or other brittle coating to be applied on all areas of probably high stress concentrations.

It is perhaps apparent that these three kinds of yield tests will not necessarily give the same or even approximately the same yield pressure. The brittle coating procedure is limited to visible portions of the outside surface. Yielding could occur on the inside surface at a lower pressure. Also, flaking of a lime-wash coating is stated[3.87] to occur at about 1% strain as compared to specified strains of 0.2 to 0.5% in the strain measurement method. The strain measurement method might be construed as requiring gages on the inside surface, if such surface includes the highest stressed point.* The displacement method does not necessarily give any information on the magnitude of plastic strains.

There is one other statement in both ASME Section I and VIII, Div. 1, which presumably was added to help clarify the intent of the yield tests. The following is quoted from Section I, Par. A-22 (d)

> "Note: Strains should be measured as they apply to membrane stresses and to bending stresses within the following range. It is recognized that high localized and secondary bending stresses may exist in pressure parts designed and fabricated in accordance with these rules. In so far as practical, design rules for details have been written to hold such stresses at a safe level consistent with experience."

The writer would interpret this statement as indicating that a degree of judgement should be used in interpreting highly localized measured strains. If the local strains (e.g., at the toe of a fillet weld)

* Many components do have the highest stress on the inside surface. Failure to recognize this could lead to acceptance of a component on the basis of a proof test during which test the interior surface not only yielded but may have also cracked.

are comparable to those expected for other constructions specifically permitted by the Code rules, those local strains should not, per se, limit the allowable working pressure of the component. The entire Code section on yield proof tests implies good engineering judgement on the part of both the manufacturer and the inspector.

The internal pressure yield test, extended somewhat to onset of gross plastic deformations, has been used extensively in connection with nozzles in pressure vessels[3.88]. Here, the intent is to compare test data with "collapse pressure" or "limit pressure" theories for such construction.

Yielding of piping components under other loads is also of some significance. The extension of yielding to "limit loads" is perhaps of more significance in piping systems. Some test data of this type are discussed in Chapters 6, 7, and 8.

### 3.213 Fatigue Tests

Fatigue tests on piping components have principally been run with either cyclic internal pressure or cyclic bending moments, the later sometimes combined with static internal pressure. Available fatigue test data are discussed in Chapters 6, 7, 8, and 14. In almost all fatigue tests on piping conponents, fatigue failure is defined as occurring when the fatigue crack penetrates the wall of the component. Comparatively few data exist on crack initiation or propagation. Essentially all tests have been run at room temperature. For cyclic moment tests, an important distinction is whether the applied moment or the displacement are actually controlled during the test. The stress intensification factors used in the American Standard Code for Pressure Piping are based on Markl's[3.89] test data in which displacements are controlled.

The ASME Code Section III[3.85] (Nuclear Vessels), Par I-1080 and USAS B31.7[3.90] (Nuclear Power Piping), Appendix E, Par. E-180, give rules for guidance in experimentally determining the fatigue strength of pressure retaining parts. Failure is defined as a crack through the wall. The implication is that tests are to be load controlled. (These sections were presumably written with internal pressure loading primarily in mind.)

#### 3.214 Creep Tests

A number of internal pressure creep tests have been run on straight pipe; these are listed in Chapter 6. Insofar as the writer is aware, no creep test data exist for other piping components; i.e., curved pipe, elbows, tees, branch connections, etc. For piping components, the creep or creep-rate is usually of secondary interest; the time-to-rupture is of primary interest. To the extent that local creep strains and strain rates can be measured, such information would be indirectly useful in that it could be correlated with material creep tests under known loading conditions.

### 3.22 Indirect Test Methods

#### 3.221 Brittle Coating

Use of a brittle coating to detect yielding in pressure vessels dates back many years. Cracking of mill scale is one such indication.

Whitewash was used to more clearly show cracking of oxide scale. The use of specially formulated resinous coatings started in 1925 and in the past 40 years considerable improvement in coatings and techniques have been developed. "Stresscoat", a product of the Magnaflux Corporation, is widely used in this country. Application and interpretation techniques are discussed in several books; e.g., Reference (3.91).

Under closely controlled conditions of temperature and humidity, brittle coatings can give fairly accurate indications of the magnitude and direction of the maximum principal strain. Brittle coatings are also used to establish locations and directions of high stresses in a complex structure; strains at these locations can then be determined quantitatively by use of electrical resistance strain gages. For pressure containing structures, brittle coatings are useful only to obtain information on visible portions of the outside surface.

### 3.222 Strain Gages

The most common method of determining strains in piping components consists of the use of electrical resistance strain gages*. The principles and techniques involved in using such gages are given in several books; e.g., Reference (3.92).

As applied to piping components, the principle problem is to locate gages at the points of maximum strain. In areas of high strain gradient, this may require the use of very small, carefully placed gages.

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* Mechanical or optical methods of strain measurement are also available; however, such methods are of relatively limited application as compared to electrical resistance strain gages.

Many existing tests on piping components are of limited validity because the strain gages were too large and/or were not located at points of maximum strain.

3.223 Photoelastic-Optical Methods

A realistic test of piping components almost necessarily involves three-dimensional models. The stresses in such models can be analyzed by the "frozen stress" technique in which the models are heated and then loaded. The model is then cooled at a slow rate (e.g., $2^{\circ}$ C per hour) so that no thermal stresses develop. The model is then sliced and the fringe patterns are determined. A general discussion of the technique is given in Reference (3.93). References (3.94) and (3.95) are examples of application of the technique to nozzles in pressure vessels.

A limitation of the technique is that a given model can be used only to determine stresses for a single load or load combination. Scattered light techniques, pioneered by Weller and Bussey[(3.104)] and now in the process of development [see References (3.96) through (3.100)], may permit use of photoelastic models for several loads.

As applied to piping components, two questions may arise:

(1) What is the effect of Poisson's ratio for the photoelastic material as compared to the actual metal component?

(2) Are displacements sufficiently small in the photoelastic model so that non-linear effects do not occur?

Stresses at the surfaces of structures may also be determined by coating the surface with a suitable birefringement material. This technique is discussed in Reference (3.93). A new method, undergoing considerable recent development work, is the Moire method of strain analysis, in which a fine grid pattern of lines is placed on the surface of the material in which strains are to be determined. Optical interference lines between the original grid and strained grid develop. The technique is discussed in Reference (3.101).

## 3.23 Material Toughness Tests

It is well known that ferritic alloys may exhibit "brittle" fracture at low temperatures. The resistance to such brittle behavior might be characterized by an elusive material property designated as "toughness" herein. The problem was recognized at least 60 years ago, at which time Izod and Charpy impact tests were first used in an attempt to characterize the brittle vs. tough characteristics of materials. Such tests have been used to the present day and are described in ASTM A370, "Methods and Definitions for Mechanical Testing of Steel Products"[(3.102)].

Pressure vessel and piping codes (see Chapter 5) have in the past (and generally at the present time) established a lower temperature limit of -20 F for non-impact-tested ferritic alloys; despite the occurrence of a significant number of failures at higher temperatures (see References (3.62) and (3.63)). For lower temperatures, the codes and material specifications* have required Charpy Keyhole or Charpy U-notch impact tests. The normally applied criteria

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* For example, ASTM A350-65, "Specification for Forged or Rolled Carbon and Alloy Steel Flanges, Flanged Fittings, and Valves and Parts for Low-Temperature Service".

of acceptability was that, at the minimum service temperature, the energy absorption in the Charpy tests should not be less than:

| Size of Charpy Specimen, mm | Energy Absorption, ft-lb. Average of Three | Energy Absorption, ft-lb. Minimum of Set of Three |
|---|---|---|
| 10 x 10 | 15 | 10 |
| 10 x 7.5 | 12.5 | 8.5 |
| 10 x 5 | 10 | 7.0 |
| 10 x 2.5 | 5 | 3.5 |

It has always been recognized, of course, that the Charpy impact values do not give any direct quantitative design guidance. Rather, the tests are used as a dividing line for material purchasing acceptance tests, based on past service experience with carbon and low-alloy steels.

In the past few years, the potential shortcomings of Charpy Keyhole or U-notched specimens has been widely recognized. Concern over the problem is justified because of the trend towards the use of higher-strength steels, with allowable design stresses tending towards a greater fraction of the ultimate tensile strength; and, in nuclear and other large high-pressure vessels, the use of much greater thicknesses than used in the past. This concern has led, on the one hand, to more discriminating impact tests, including the Charpy V-notch test; to the concepts of transition temperatures; and on the other hand to the use of linear elastic fracture mechanics and extensions thereof, as discussed in Par. 3.16.

The present trends in piping codes* can be illustrated by reference to USAS B31.3-1966, Petroleum Refinery Piping[3.40]. This code specified in Tables of allowable stresses, minimum temperatures with a note indicating

* A complete discussion of relevant piping codes and standards can be found in Chapter 5.

that the minimum temperatures shown are those for which the material is normally suitable, but requirements for design below -20 F are established elsewhere in the code. Those rules state that a material may be used below the minimum temperature shown in the tables (Table 302.3.1, A and B; Appendix A, or 304.5.1, A and B, Appendix C) provided it meets the applicable impact test requirements of Par. 323.2.2. Those requirements are:

> 323.2.2 Impact Tests.
>
> (a) Impact tests shall be made for the following combinations of materials and temperatures:
>
> (1) all materials for temperatures below the minimum temperature shown in the stress tables (Table 302.3.1A and B, Appendix A; or Table 304.5.1A and B, Appendix C).
>
> (2) bolting material conforming to ASTM A193 Grade B7, and to ASTM A194 Grade 2H for temperatures below minus 50 F, and to ASTM A194 Grade 4 for temperatures below minus 150 F
>
> (3) The following material below -20 F, except that no impact testing of these materials is required for metal temperatures below -20 F but not below -50 F if the design pressure does not exceed 15% of the maximum allowable pressure at temperature:
>
> carbon and low-alloy steels other than Grade B7 of ASTM A193 and Grades 2H and 4 of ASTM A194
>
> ferritic chromium stainless steels
>
> austenitic chromium-nickel stainless steels with a carbon content greater than 0.10%
>
> austenitic chromium-nickel stainless steel materials in the form of deposited weld metal regardless of carbon content

austenitic chromium nickel stainless materials not in the solution heat treated condition.

(b) The impact tests shall be made in accordance with and shall meet the requirements of UG-84 of Section VIII of the ASME Boiler and Pressure Vessel Code (see Chapter 5), with the following substitution for UG-84(b), (1), (2), and (3):

(1) a welded test section shall be prepared from a piece of plate, pipe, or tubing for each material specification certified by the manufacturer in accordance with UG-84(e). If the material to be used is not certified, test sections shall be prepared from each piece of pipe, plate, or tubing used. One set of impact-test specimens shall be taken across the weld with notch in the weld (the metal tested in the weld metal) and one set shall be taken similarly with the notch at the fusion line (the metal tested is the base metal).

(2) One set of impact test specimens with the notch in the weld metal and one set with the notch at the fusion line shall be made for each range of pipe thickness that does not vary by more than 1/4 inch from the tested thickness for each material specification called for by the engineering design.

(3) Unless otherwise specified in the engineering design, the testing required in (1) and (2) need not be performed on material from individual lots at any time, nor from material from each job, provided other material in the

thickness range of (2) and to the same specification (except for heat or lot) has been tested as required and the records of those tests are available; and testing need not be repeated on any individual piece for which testing was required in (1) if that piece can be associated with a satisfactory test record.

The following points are pertinent with respect to these rules:

(1) The minimum temperatures shown in the Tables are never higher than -20 F.

(2) Austenitic stainless steel with carbon content greater than 0.10% or in the form of deposited weld metal are not exempt from impact test requirements.

(3) The impact tests of UG-84, Section VIII, (Division 1) of the ASME Boiler and Pressure Vessel Code, are Charpy Keyhole or Charpy U-notch types. Impact requirements are those (foot-pound) values tabulated previously.

The ASME Boiler Code, Section VIII, Division 1 has similar requirements.

The requirements of Section III (Nuclear Vessels) of the ASME Boiler Code illustrate perhaps the most rigorous approach to guarding against brittle fracture, insofar as such can be done in the present state of the art using impact tests as a criterion. These requirements are:

N-331 Ductile-Brittle Transition Tests

Carbon steel, alloy steel, and chromium stainless steel (Series 4XX) shall be tested for ductile to brittle transition (NDT) temperature by either the dropweight test (ASTM E208) or the Charpy V-notch impact test (ASTM A-370 Type A). Such tests are not required where section thickness is less than 1/2 in. or for austenitic stainless steels or non-ferrous materials. These tests shall be conducted at temperatures based on the limits imposed by hydrostatic testing temperatures and service temperatures of the vessel. Both test techniques are permitted without stated preference as to one over the other. Either technique is presently considered to be an adequate test for new construction. However, information now being developed may provide additional clarification with respect to the significance of these data.

From N-331.1 Dropweight Tests

An acceptance tests shall consist of at least two dropweight specimens tested by the "break or no-break method" described in E-208. Each specimen shall exhibit "no-break" performance at a test temperature 60° F below the lower of the vessel hydrotest temperature or the lowest service metal temperature.

From N-331.2 Charpy V-notch Tests

An acceptance test shall consist of a set of three specimens tested at temperatures 60° F below the lower of the vessel hydrotest temperature or the lowest service metal temperature. The specimens shall break at energies no less than those indicated in Tables N-421 and N-422 by steel grade and the absorbed energy values shall be reported. The test temperature, the lateral expansion in inches and the percent ductile fracture area for each specimen shall be reported for information.

Full-size, Charpy V-notch 10 x 10 mm specimens shall be used unless the material section requires that smaller specimens be made. In this case, reduced-width specimens may be used where the dimension along the notch shall be the largest possible of 7.5, 5, or 2.5 mm. The acceptance values for the Charpy V-notch impact energy as listed in Table N-421 shall be multiplied by 5/6, 2/3, and 1/3 respectively, to establish the acceptance value for the subsize specimens.

The ASME Code, Section III, in contrast to the USAS B31.3 and Section VIII, Division 1, requires impact tests for ferritic alloys regardless of temperature (in contrast to -20 F or lower) and has gone to Charpy V-notch or dropweight tests (in contrast to Charpy keyhold or U-notch).

The dropweight test is standardized in ASTM E-208[(3.103)] and its development and significance is discussed by Pellini[(3.73)]. Briefly, it consists of a 1" x 3-1/2" x 14" specimen* which has a short weld bead of brittle material running centered on one of the 3-1/2" x 14" sides. A 1/16" wide cut is machined in the 3-1/2" direction across the weld bead. The specimen is tested by an impact on the side opposite the weld with the specimen supported by two knife edges. This places the weld side in tension with the bending stress direction normal to the machined cut. The nil-ductility temperature (NDT) is defined as the maximum temperature at which the specimen breaks.

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* These dimensions are specifically for the P-1 specimen. Smaller sizes are also used.

Charpy energy absorption values (ft-lb.) are given in Section III tables of allowable stress intensities for ferritic alloys. For most alloys, the requirements are the same as thos tabulated previously in this section (e.g., 15 ft-lb average in full-size Charpy specimens). Additional requirements for measurement of lateral expansion and percent ductile shear area represent other approaches to the evaluation of Charpy impact test results; however no quantitative evaluation thereof is, at present, included in Section III.

One anomalous point in the comparisons is that the B31.3 piping code and Section VIII, Division 1 do not exempt welds in austenitic stainless steel from impact test requirements whereas Section III apparently does so.

The requirements of Section VIII, Division 2, represent another example of trends in code application of material toughness requirements. Section VIII-2, like Section III, has gone to the Charpy V-notch test specimen, but has not yet accepted the dropweight test. The impact test exemptions for carbon steels are of particular interest and are shown herein as Figure 3.1.

Section VIII-2 provides another type of exemption which is included, in a somewhat analogous manner, in USAS B31.3 and Section VIII-1 but not in Section III. For "High-Alloy Steels", impact tension is not required provided the stress* does not exceed 6000 psi. For "Carbon Steels", impact tension is not required provided the stress* does not exceed 6000 psi and the temperature is not lower than -50 F. A similar exemption for "Low-Alloy Steels" is "in course of preparation". Presumably this type of exemption stems from test data indicating that, even in the presence of a crack-like defect, a nominal stress of 6000 psi is unlikely to cause crack growth.

* This is not an exact quote from the Code.

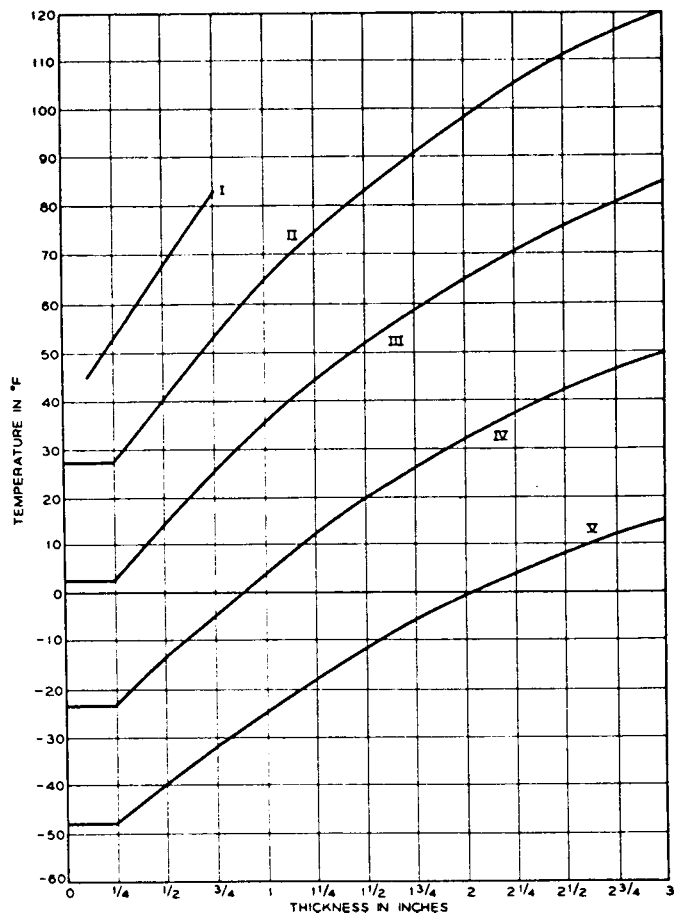


FIG. AM-218.1 IMPACT TEST EXEMPTION CURVES FOR CARBON STEELS

NOTES FOR FIG. AM-218.1

Note 1: Impact tests are not required for a given minimum design temperature if a steel is selected so that the curve representing that steel at the required thickness is below the minimum design temperature.

Note 2: The carbon steel groupings as listed below apply to the standard specification conditions of heat treatment, grain size, and chemistry limits, unless otherwise noted:

GROUP I: Includes only SA-36 plate up to 3/4 in. in thickness when welded to primary pressure components;

GROUP II: (a) Plate steels: SA-36 over 3/4 in. in thickness when welded to primary pressure components; SA-285 and SA-515; (b) all other product forms of carbon steel conforming to specifications listed in Table ACS-1;

GROUP III: (a) Plate steels: SA-442 up to 1 in. in thickness inclusive; (b) all other product forms of carbon steel up to 1 in. in thickness (see Note 3) having special carbon and manganese limits the same as for SA-442 for comparable strength grades;

GROUP IV: (a) Plate steels: SA-442 over 1 in. in thickness when not normalized; SA-516 up to 1½ in. in thickness inclusive; (b) other product forms: Up to 3 in. in thickness inclusive, when made to fine-grain practice, and with carbon and manganese limits the same as for SA-516 plate for comparable thicknesses and strength grades.

GROUP V: For all product forms: Steels as listed for Group IV when normalized (note: SA-516 requires normalizing as a standard specification requirement over 1½ in. thick).

Note 3: Thickness Definition for Standard Flanges: For application of Fig. AM-218.1 the thickness of standard flanges conforming to USAS-B16.5 shall be defined as the maximum nominal thickness of the pressure part under consideration at any strength weld, including those attaching non-pressure parts.

Note 4: Impact Testing at Temperatures Below Curves: When impact tested according to Par. AM-204.1 steels may be used at temperatures lower than those established by the curves, provided they meet the impact test requirements, but no lower than the impact test temperature.

Note 5: Impact Testing of Thick Sections: For thicknesses greater than 3 in. when the minimum design temperature is lower than +120 F, impact tests shall be made according to Par. AM-204.1.

Note 6: Impact Testing of Materials Subjected to Accelerated Cooling: For thicknesses greater than 2 in. and when the design temperature is lower than +120 F, material subjected to accelerated cooling (by liquid sprays or immersion) shall be impact tested according to Par. AM-204.1.

FIGURE 3.1. IMPACT TEST EXEMPTION CURVES FOR CARBON STEEL, FROM ASME BOILER AND PRESSURE VESSEL CODE, SECTION VIII, DIVISION 2.

Section VIII-2, like Section III, requires that lateral expansion and percent shear of Charpy tests be recorded. Section VIII-2 (Summer 1969 Addenda) had added a specific lateral expansion requirement for certain materials (e.g., carbon and low-alloy steels having a specified minimum ultimate strength of 100,000 psi or more); the minimum lateral expansion is 0.015 in. for an average of 3 tests; 0.010 for any one of the three tests.

REFERENCES

(3.1) Use of the Computer in Pressure Vessel Analysis, Papers presented at ASME Computer Seminar, Dallas, Texas, September 20, 1968, Published by the ASME, 345 E. 47th St., New York, N. Y. 10017.

(3.2) AXISOL, See "Structural Analysis of Axisymmetric Solids", by E. L. Wilson, AIAA Journal, Vol. 3, pp 2269-2274 (1965).

(3.3) BASIC, See "A Study of Rotary-Shaft-Sealing Concepts for Pressurized Water-Reactor Applications", by Grieser, et. al., Battelle Memorial Institute, Report No. BMI-1676, June 30, 1964.

(3.4) DUZ-1, See "A Numerical Solution for Axially Symmetrical and Plane Elasticity Problems", by D. S. Griffin and R. B. Kellogg, Int. J. of Solids and Structures, Vol. 3, 1967, pp 781-794.

(3.5) MOLSA, See "Analysis of Shells of Revolution Subjected to Symmetrical and Nonsymmetrical Loads", by A. Kalnins, ASME J. of App. Mech., September, 1964.

(3.6) NONLIN, See "On Nonlinear Analysis of Elastic Shells of Revolution", by A. Kalnins and J. F. Lestingi, J. App. Mech., Trans. ASME, Series E, Vol. 34, No. 1, pp 59-64 (March, 1967).

(3.7) SAFE-PCRS, See "Analysis of Axisymmetric Composite Structures by the Finite Element Method", by Y. R. Rachid, Nuclear Engineering and Design, Vol. 3, 1966, pp 163-182.

(3.8) SEAL-SHELL-2, See "A Computer Program for the Stress Analysis of a Thick Shell of Revolution with Axisymmetric Pressure, Temperatures and Distributed Loads", by C. M. Friedrich, Bettis Atomic Power Lab., Report WAPD-TM-398.

(3.9) SHOREF, See "On Free and Forced Vibration of Rotationally Symmetric Layered Shells", by A. Kalnins, J. App. Mech., Vol. 32, pp 941-943 (1965).

(3.10) Meyer, R. R. and Harmon, M. B., "Conical Segment Method for Analyzing Open Crown Shells of Revolution for Edge Loading", AIAA Journal, Vol. 1, No. 4, p 886 (1963).

(3.11) Grafton, P. E. and Strome, D. R., "Analysis of Axisymmetric Shells by the Direct Stiffness Method", AIAA Journal, Vol. 1, No. 10, p 2342 (1963).

(3.12) Popov, J. H., Penzien, J., and Lu, Z. A., "Finite Element Solution for Axisymmetrical Shells", ASCE Eng. Mech. Div. Journal, p 119, October, 1964.

(3.13) Percy, J. H., Pian, T.H.H., Klein, S., and Navaratna, D., "Application of Matrix Displacement Methods to Linear Elastic Analysis of Shells of Revolution", AIAA Paper No. 65-142, January, 1965.

REFERENCES (contd.)


(3.14) Clough, R. W. and Rashid, Y., "Finite Element Analysis of Axisymmetric Solids", ASCE Eng. Mech. Div. Journal, Vol. 91, No. EMI, p 71 (1965).

(3.15) Jones, R. E. and Strome, D. R., "Direct Stiffness Method Analysis Using Influence Coefficient Techniques".

(3.16) Theory of Matrix Structural Analysis, by J. S. Przemieniecki, McGraw-Hill Book Co., New York, N. Y., 1968.

(3.17) The Finite Element Method in Structural and Continuum Mechanics, by O. E. Zienkiewicz and Y. K. Cheung, McGraw-Hill Book Co., London, 1967.

(3.18) FEELAP, See "Elastic-Plastic Analysis of Two-Dimensional Stress Systems by the Finite Element Method", by P. V. Marcal and I. P. King, Int. J. Mech. Sci., Vol. 9, pp 143-155 (1967).

(3.19) Greenbaum, G. A., and Rubinstein, M. F., "Creep Analysis of Axisymmetric Bodies Using Finite Elements", Nuclear Engineering and Design, Vol. 7, 1968, pp 379-397.

(3.20) CLPSHL, See "Theoretical Analysis of the Plastic Collapse of Thin Shell Structures", Battelle Memorial Institute, Columbus, Ohio, December, 1966.

(3.21) Nakamura, T., "Plastic Analysis of Shells of Revolution Under Axisymmetric Loads", Ph.D. Dissertation, Stanford University, Palo Alto, California (1961).

(3.22) Hrennikoff, A., "Solutions of Problems of Elasticity by the Framework Method", Trans. ASME J. of Applied Mechanics, Vol. 63, p A-169 (1941).

(3.23) CSMTRX, See "Space-Vehicle Stabilized-Platform Gimbel-System Weight-Reduction Study, Phase 1, Design of Ring Gimbels, Battelle Report to NASA, December 28, 1962.

(3.24) ELAS, See "ELAS - A General-Purpose Computer Program for the Equilibrium Problems of Linear Structures", Jet Propulsion Laboratory Technical Report 32-1240, February, 1968.

(3.25) FORMAT II, See "FORMAT-II-Second Version of Fortrain Matrix Abstraction Technique", by J. Pickard, et. al., Douglas Aircraft Co., AFFDL-TR-66-207, Volumes I, II, III, and IV.

(3.26) GENSAM, See "GENSAM - A General Structural Analysis and Matrix System", Electric Boat Division Report No. P411-60-021, March, 1969.

(3.27) PAPA, See "PAPA--The Digital Computer Program for the Static Analysis of Structures Made Up of Plate and Panel Elements", by L. Beitch and E. North, General Electric Co., Report GEAP-5553, October, 1967.

(3.28) SAFE-3D, See "SAFE-3D, A Computer Program for the Three-Dimensional Stress Analysis of Composite Structures", Gulf General Atomic Report GA-7855, 1967.

REFERENCES (contd.)

(3.29) SAMIS, See "Summary of the Functions and Capabilities of the Structural Analysis and Matrix Interpretive System Computer Program", by T. E. Lang, NASA Technical Report 32-1075, April, 1967.

(3.30) Clough, R. W. and Johnson, C. P., "A Finite-Element Approximation for the Analysis of Thin Shells", Int. J. Solids Structures, Vol. 4, p 43, (1968).

(3.31) Baker, W. E. and Daly, J. M., "Dynamic Analysis of Continuum Bodies by Direct Stiffness Method", the Shock and Vibration Bulletin, No. 36, Part 5, January, 1967.

(3.32) Eringen, Naghdi, Mahmood, Thiel, and Ariman, "Stress Concentrations in Two Normally Intersecting Cylindrical Shells Subject to Internal Pressure", General Technology Corp., Welding Research Council Bulletin No. 139, April 1969.

(3.33) Rodabaugh, E. C. and George, H. H., "Effect of Internal Pressure on Flexibility and Stress-Intensification Factors of Curved Pipe or Welding Elbows", Trans. ASME, Vol. 79, p 939 (1957).

(3.34) Kalnins, A., "Analysis of Curved Thin-Walled Shells of Revolution", Proceedings of the AIAA/ASME 8th Structures, Structural Dynamics and Materials Conference, 1967, p 675.

(3.35) Bijlaard, P. P.

"Stresses from Radial Loads in Cylindrical Pressure Vessels", The Welding Journal Research Supplement, December, 1954.'

"Stresses from Local Loadings in Cylindrical Pressure Vessels", Trans. ASME, Vol. 77, p 805 (1955).

"Stresses from Radial Loads and External Moments in Cylindrical Pressure Vessels", The Welding Journal Research Supplement, p 608s, December, 1955.

"Computation of the Stresses from Local Loads in Spherical Pressure Vessels or Pressure Vessel Heads", Welding Research Council Bulletin No. 34 (March, 1957).

"Local Stresses in Spherical Shells from Radial or Moment Loadings", The Welding Journal Research Supplement, p 240s, May, 1957.

"Stresses in a Spherical Vessel from Radial Loads Acting on a Pipe"; "Stresses in a Spherical Vessel from External Moments Acting on a Pipe"; "Influence of a Reinforcing Pad on the Stresses in a Spherical Vessel Under Local Loading", Welding Research Council Bulletin No. 49 (April, 1959).

"Stresses in Spherical Vessels from Local Loads Transferred by a Pipe"; "Additional Data on Stresses in Cylindrical Shells Under Local Loading".

REFERENCES (contd.)

(3.36) Waters, E. O., "Stresses Near a Cylindrical Outlet in a Spherical Vessel", Welding Research Council Bulletin No. 96 (May, 1964). (Computer program contains additional thick-wall effects, based on a private communication from E. O. Waters to E. C. Rodabaugh, May, 1963; these factors are discussed in Battelle Memorial Institute Report to the USAEC, "Evaluation of Experimental and Theoretical Data on Radial Nozzles in Pressure Vessels".)

(3.37) Moore, S. E. and Witt, F. J., "CERL-II, A Computer Program for Analyzing Hemisphere - Nozzle Shells of Revolution with Axisymmetric and Unsymmetric Loadings", Oak Ridge National Laboratory, ORNL-3817, October, 1965.

(3.38) Rodabaugh, E. C. and Atterbury, T. J., "Stresses in Tapered Transition Joints in Pipelines and Pressure Vessels", Trans. ASME Vol. 84B, p 321 (1962).

(3.39) Rodabaugh, E. C., "Computer Programs for the Analysis of Flanged Joints", Preliminary Draft of Phase Report 115-7, Battelle-Columbus to Oak Ridge National Laboratory (1969).

(3.40) USA Standard Code for Pressure Piping, Published by the American Society of Mechanical Engineers, 345 East 47th St., New York, N. Y. 10017.

(3.41) Mechanical Drive Steam Turbines, NEMA Pub. No. SM-20-1958, Published by National Electrical Manufacturers Association, 155 E. 45th St., New York, N. Y.

(3.42) Brock, J. E., "Expansion and Flexibility", Chapter 4 of Piping Handbook, 5th edition (1967), McGraw-Hill Book Co., New York.

(3.43) Piping Handbook, 4th edition, 1945 (edited by S. Crocker), McGraw-Hill Book Co., New York.

(3.44) Piping Design and Engineering, Grinnell Co., Providence, Rhode Island, 1951.

(3.45) Z-, L-, and V- Expansion Bends, Tube Turns Piping Engineering Paper No. 4.02, Tube Turns Div. of Chemetron Corp, Louisville, Kentucky, 1950.

(3.46) Griffin, J. S., "MEC-21/7094, A Piping Flexibility Analysis Program for the IBM-7090 and 7094", Los Alamos Scientific Laboratory Report LA-2929, July, 1963.

(3.47) PIPE, See "PIPE - A Computer Program for the Elastic Stress Analysis of a Piping Structure with Thermal Strains and External Loads", by C. M. Friedrich, Westinghouse WAPD-TM-752, March, 1968.

(3.48) Robinson, E. L., "Steam-Piping Design to Minimize Creep Concentrations", Trans. ASME, Vol. 77, p 1147 (1955).

REFERENCES (contd.)

(3.49) Power Piping, USAS B31.1.0 (1967), Section 1 of the American Standard Code for Pressure Piping, Published by ASME, 345 E. 47th St., New York, New York 10017.

(3.50) Nadai, A., Theory of Flow and Fracture of Solids, McGraw-Hill Book Co., New York, New York (1950).

(3.51) Finnie, I. and Heller, W. R., Creep of Engineering Materials, McGraw-Hill Book Co., New York, New York (1959).

(3.52) Grover, H. J., Gordon, S. A., and Jackson, L. R., Fatigue of Metals and Structures, Supt. of Documents, U.S. Govt. Printing Office, NAVWEPS 00-25-534 (1960).

(3.53) Coffin, L. F., "Introduction to High-Temperature Low-Cycle Fatigue", Experimental Mechanics, May, 1968.

(3.54) Benham, P. P., "High-Temperature Low-Cycle Fatigue: Survey of British Work", Experimental Mechanics, July 1968.

(3.55) ASME Boiler and Pressure Vessel Code, Published by the American Society of Mechanical Engineers, 345 E. 47th St., New York, New York 10017.

(3.56) Strawley, J. E. and Brown, W. F., "Fracture Toughness Testing Methods", Included in Reference (3.58).

(3.57) Flow and Fracture of Metals and Alloys in Nuclear Environments, STP 380 (1965), Published by the American Society for Testing and Materials, 1916 Race Street, Philadelphia, Pa. 19103.

(3.58) Fracture Toughness Testing and its Applications, STP 381 (1965), Published by the American Society for Testing and Materials, 1916 Race Street, Philadalphia, Pa. 19103.

(3.59) Plane Strain Crack Toughness Testing of High Strength Metallic Materials, STP 410 (1967), Published by the American Society for Testing and Materials, 1916 Race Street, Philadelphia, Pa. 19103.

(3.60) Proceedings of the Second International Conference on Fracture, Brighton, April, 1969. Published by Chapman and Hall, Ltd., 11 New Fetter Lane, London EC4.

(3.61) Fracture, an Advanced Treatise, Edited by H. Liebowitz, Seven volumes, Academic Press, New York and London, 1969.

(3.62) Shank, M. E., "A Critical Survey of Brittle Fracture in Carbon Plate Steel Structures Other than Ships", Welding Research Council Bulletin No. 17, January, 1954.

REFERENCES (contd.)

(3.63) Acker, H. G., "Review of Welded Ship Failures", Ibid., No. 19, November, 1954.

(3.64) Pellini, W. S. and Puzak, P. P., "Fracture Analysis Diagram Procedures for the Fracture-Safe Engineering Design of Steel Structures", Ibid, No. 88, May, 1963.

(3.65) Masubuchi, K., Monroe, R. E., and Martin, D. C., "Interpretive Report on Weld-Metal Toughness", Ibid., No. 111, January, 1966.

(3.66) Landerman, E. and Yanichko, S. E., "Determination of Fracture Toughness of Heavy-Section Pressure Vessel Steels Using a Fracture Mechanics Approach", Ibid., No. 120, February, 1967.

(3.67) Yoder, G., Weiss, V., and Liu, L. W., "Notch Properties of Some Low and Medium Strength Constructional Steels", Ibid.,

(3.68) Cooley, L. A. and Lange, E. A., "Fracture Development and Material Properties in PVRC-Penn State Pressure Vessel", Ibid., No. 126, November, 1967.

(3.69) Crooker, T. W., Cooley, L. A., and Lange, E. A., "Fatigue-Crack Propagation and Fracture Studies of a Pressure-Vessel Steel Temper Embittled to Simulate Irradiation Damage", Ibid.

(3.70) Pellini, W. S., "Advances in Fracture Toughness Characterization Procedures and in Qualitative Interpretations to Fracture-Safe Design for Structural Steels", Ibid., No. 130, May, 1968.

(3.71) Goode, R. J., Judy, R. W., and Huber, R. W., "Procedures for Fracture Toughness Characterization and Interpretations to Failure-Safe Design for Structural Titanium Alloys", Ibid., No. 134, October, 1968.

(3.72) Judy, R. W., Goode, R. J., and Freed, C. N., "Fracture Toughness Characterization Procedures and Interpretations to Fracture-Safe Design for Structural Aluminum Alloys", Ibid., No. 140, May, 1969.

(3.73) Pellini, W. S. and Loss, F. J., "Integration of Metallurgical and Fracture Mechanics Concepts of Transition Temperature Factors Relating to Fracture-Safe Design for Structural Steels", Ibid., No. 141, June, 1969.

(3.74) Catanach, W. M. and Erdogan, F., "Fatigue Crack Propagation in Cylindrical Shells", Included in Reference (3.60).

(3.75) General Electric Co., APED, San Jose, California; Reactor Primary Coolant Loop Rupture Study, Quarterly Progress Reports 1-16, April, 1965 through April, 1969.

(3.76) Edmondson, B., Formby, C. L., Jurevics, R. and Stagg, M. S., "Aspects of the Failure of Large Steel Pressure Vessels", Included in Reference (3.60).

REFERENCES (contd.)


(3.77) Wessel, E. T., Clark, W. G., and Pryle, W. H., "Fracture Mechanics Technology Applied to Heavy Section Steel Structures", Included in Reference (3.60).

(3.78) Bluhm, J. I., "Fracture Arrest", Included in Volume 5 of Reference (3.61).

(3.79) Symposium on Line Pipe Research, March, 1966. Available from the American Gas Association, 605 Third Ave., New York, N. Y. 10016, A.G.A. Catalogue No. L30000, pp 83-118.

(3.80) Eiber, R. J., Maxey, W. A., Duffy, A. R., and Atterbury, T. J., "Investigation of the Initiation and Extent of Ductile Pipe Rupture", Battelle-Columbus Report BMI-1866, July, 1969.

(3.81) Robertson, T. S., "Propagation of Brittle Fracture in Mild Steel", J. of the Iron and Steel Institute, pp 361-374, December, 1963.

(3.82) Dugdale, D. S., "Yielding of Thin Sheets Containing Slits", J. Mech. Phys. Solids, Vol. 8, p 100, 1960.

(3.83) Hahn, G. T., Kanninen, M. F., and Rosenfield, A. R., "Ductile Crack Extension in Steel Foil", Included in Reference (3.60).

(3.84) McClintock, F. A. and Irwin, G. R., "Plasticity Aspects of Fracture Mechanics", Included in Reference (3.57).

(3.85) ASME Boiler and Pressure Vessel Code, Published by the American Society of Mechanical Engineers, 345 E. 47th St., New York, N. Y. 10017.

(3.86) American Standard USAS B16.9, Wrought Steel Buttwelding Fittings, Published by the American Society of Mechanical Engineers, 345 E. 47th St., New York, New York 10017.

(3.87) Handbook of Experimental Stress Analysis, Chapter 14 on Brittle Coatings by M. Hetényi, John Wiley & Sons (1954).

(3.88) Rodabaugh, E. C. and Cloud, R. L., "Assessment of the Plastic Strength of Pressure Vessel Nozzles", ASME Transactions, J. of Engr. for Power, April, 1968.

(3.89) Markl, A.R.C., "Fatigue Tests of Piping Components", Trans. ASME, p 287 (1952).

(3.90) American Standard Code for Pressure Piping, USAS B31.7 (Nuclear Power Piping), 2/1968, for Trial Use and Comment, Published by the American Society of Mechanical Engineers, 345 E. 47th St., New York, New York 10017.

(3.91) Durelli, A. J., Phillips, E. A., and Tsao, C. H., Introduction to the Theoretical and Experimental Analysis of Stress and Strain, McGraw-Hill Book Co. (1955).

REFERENCES (contd.)

(3.92) Perry, C. C., and Lissner, H. R., The Strain Gage Primer, McGraw-Hill Book Co. (1955).

(3.93) Holister, G. S., Experimental Stress Analysis, Cambridge University Press (1967).

(3.94) Taylor, C. E., and Lind, N. C., "Photoelastic Study of Stresses Near Openings in Pressure Vessels", Welding Research Council Bulletin No. 113, April, 1966.

(3.95) Leven, M. M., "Photoelastic Determination of the Stresses in Reinforced Openings in Pressure Vessels", Ibed.

(3.96) Taylor, Bowman, North & Swinson, "Applications of Lasers to Photo-elasticity", Experimental Mechanics, June, 1966.

(3.97) Swinson, W. F., and Bowman, C. E., "Application of Scattered-light Photoelasticity to Double Connected Tapered Torsion Bars", Experimental Mechanics, June, 1966.

(3.98) North, W.P.T., and Taylor, C. E., "Dynamic-stress Concentration Using Photoelasticity and a Laser Light Source", Experimental Mechanics, p 337, July, 1966.

(3.99) Cheng, Y. F., "Some Techniques for Employing a Continuous-wave Gas Laser as a Light Source in Scattered-light Static Photoelasticity", Experimental Mechanics, p 431, August, 1966.

(3.100) Cheng, Y. F., "A Dual-Observation Method for Determining Photoelastic Parameters in Scattered Light", Experimental Mechanics, p 140, March, 1967.

(3.101) Riley, W. F., "Moire Method of Strain Analysis", Experimental Mechanics, p 19A, February, 1967.

(3.102) "Methods and Definitions for Mechanical Testing of Steel Products", ASTM Specification A370-67, Published by American Society for Testing and Materials, 1916 Race St., Philadelphia, Pa. 19103.

(3.103) Conducting Drop-Weight Test to Determine Nil-Ductility Transition Temperature of Ferritic Steels, ASTM Designation E-208-66T, Published by American Society for Testing and Materials, 1916 Race St., Philadelphia, Pa. 19103.

(3.104) Weller, R. and Bussey, J. K., "Photoelastic Analysis of Three-Dimensional Stress Systems Using Scattered Light", NACA Tech. Note No. 737, Nov. 1939.

CHAPTER 4

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# 4. FIELD FAILURES

Failures in piping components in light water cooled and moderated nuclear reactor power plants have been few and of minor consequence. This can be attributed to limited service experience with this class of hardware as well as the comprehensive engineering design effort and extraordinary quality control and inspection requirements used in their construction. The failures experienced conform in origin, nature, development, and characteristics to those experienced in fossil fueled power plants. At the present time, therefore, the best available guide for prediction of failure behavior of piping in the nuclear plants is previous experience with fossil fueled plants correlated with observed behavior in nuclear plants and the comparison of the differences in construction and operation of the types of plants that affect structural response.

## 4.1 Significance of Failure

It is essential to point out a difference in significance of one type of component failure in the two different power systems. That is, inspection, repair, and maintenance of piping which contains and circulates a radioactive fluid or is in a high radiation level area is so costly and time consuming (where feasible) that events considered as normal operational procedures in fossil fueled plants are unacceptable in nuclear plants. Consequently, there are insufficient records available from fossil fueled plant experience to provide a historical background suitable to provide statistics for the most frequent type of failure expected for the nuclear plant, which would be leaks found and repaired in normal plant maintenance.

## 4.2 Failure Incidence Surveys

The incidence of failure is of particular interest. Surveys have been performed here and abroad for the AEC and the UKAEA which, while based on incomplete data, are in general agreement.[4.1, 4.2] The conclusion of the UKAEA investigation is that operational inspection of fossil fueled power plants has resulted in the detection and repair of component failures which could have initiated catastrophic failure at an average rate of $10^3$ - $10^4$ plant years per failure. The conclusion of the AEC investigation is that failure-free service lives of fossil fueled power plant piping requiring major repair ranges from 1 year to 24 years with average failure-free service life of 9.8 years. The incidence of failures of subsidiary components, such as instrumentation connections, which are routinely repaired as a maintenance procedure and for which records are not kept, is much higher, according to informal discussions with utility operating personnel. Despite the increase in sophistication of design and quality control of nuclear plants, compared with fossil fueled plants, which may be balanced with the lack of opportunity for preventive maintenance inspection, the experience with light water cooled and moderated nuclear plants fits the fossil fueled plant record. There have been, of course, no catastrophic failures. Service failure experience of the same type found to occur at 9.8 year average period in piping components have occurred at periods ranging from 1 to 6 years after startup with but few of these incidents having occurred. Several subsidiary component failures are known to have occurred that have been treated as ordinary maintenance activities. Those failures in nuclear plants which have been recorded, are furthermore, no different in origin and development than samples recorded from fossil fueled plant experience. Consequently, it appears justified to use fossil fueled plant experience as a basis for predicting nuclear plant behavior, if due attention is paid to the differences in structural geometry, materials of construction, and environment on hardware behavior. In particular, it is worthwhile to note that no atypical failures in nuclear piping have been observed; that is, every failure would have been expected to occur had the full set of factors contributing to failure (stress, materials properties, and environment) been known and taken into account beforehand.

## 4.3 Causative Factors

Typical failures are usually the result of a set of causative factors of which one factor is usually of primary significance. These factors may be characterised as:

1. Design oversight. That is, failure to accurately predict service loadings or stress magnitudes and history in response to service loadings. In the nuclear plant, this is likely to occur because of the differences in piping layout geometry derived from the process, containment, and shielding requirements as compared with conventional piping layout geometries. This includes short, stiff runs of pipe without opportunity to accommodate expansion requirements by plan and elevation layout of equipment and piping components; piping restraints required for unusual events such as seismic loading; and accidental restraints developed by malfunction of devices installed to permit component movement. The lack of analytical methods to predict some service loadings, such as fluid flow induced vibrations, and to predict flexibility and stress magnitudes for some piping components should also be considered in

this category because such information can be derived empirically when the problem is recognized to exist.

2. Unsatisfactory materials properties. That is, the choice of materials whose as-fabricated properties are inadequate for imposed service loads and environments. In the nuclear plant, this is likely to occur from modification of properties by fabrication procedures or lack of knowledge of environmental effects on materials of construction.

3. Inadequate quality control and inspection. That is, the undiscovered use of unsatisfactory materials or fabrication processes, including cleaning, handling, and storage methods and the acceptance of hardware with unsatisfactory flaws and defects.[4.3]

It is obvious that these factors are related and can all, in fact, be considered as design considerations in the broad sense. At present, there is not adequate information available to the designer to select materials and write specifications to avoid failures such as have occurred in nuclear service. This does not absolve the designer of the responsibility for the incidents because it is the designer's function to analyze the serviceability problem, define the criteria, and establish methods for solving them. The development of a design without consideration of the effects of fabrication on the materials of construction, for example, is not sound engineering practice.

## 4.4 Some Typical Failure Examples

A few typical examples of the service failures experienced in nuclear plant piping with comparison of failures in fossil fueled plants will be presented. These examples will be augmented by laboratory or pressure vessel experience that is directly related to piping serviceability. There are several other examples of failure which have occurred during hydrostatic test or startup which will not be included herein because they resulted from such poor engineering practice that they cannot be considered typical.

The bimonthly publication Nuclear Safety* contains brief descriptions and references to detailed reports on safety-related occurrences in the field of reactors and radioactivity handling operations. For example, the May-June 1968 issue lists 76 "occurrences" in production and utilization facilities in 1966, of which 7 are related to failures of piping components.

The Vallecitos Boiling Water Reactor recirculating pipe failure is a typical example of design oversight.[4.4] VBWR is a forced circulation boiling water reactor operated in connection with various research and development programs hence is subjected to more load cycles than a utility power reactor. The recirculation outlet pipes are 10" diameter SA 240 Type 304 stainless steel 0.65 inch wall thickness sections fabricated from plate. A straight, short run of this pipe was restrained by the vessel nozzle to which it is joined and the three foot thick biological shield brick wall which it penetrates. A 1" feedwater line connects to the recirculating pipe between the vessel nozzle and brick wall introducing water at 70° to 100°F into the 500° to 550° line without a thermal shield. The recirculation pipe to nozzle fitup was poor so that extra welding and grinding at the weldment was required

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* Prepared by Division of Technical Information, U. S. Atomic Energy Commission by the Nuclear Safety Information Center, Oak Ridge National Laboratory, Oak Ridge, Tennessee.

during fabrication. The pipe was not radiographed for acceptance or ultrasonically tested following fabrication. A leak was discovered in the recirculation pipe after about five years of service and failure analysis performed.

The crack that leaked was between the recirculation pipe and the vessel nozzle and well outside of the pipe-to-vessel nozzle weldment. It was circumferential and about 2-1/2" in length on the outer surface and 6" in length on the inner surface. There were, in addition, typical "craze" cracks adjacent to the feedwater branch line with depths of 1/8 to 1/4 inch. Microscopically, these cracks were both intergranular and transgranular. It is estimated, from the operating record, that 500 cycles of mechanical loading to about 0.2% strain (from temperature loading and restraint) and 1,000 cycles of thermal loading to about 0.5% strain (from temperature difference between feedwater and recirculation water) occurred and were the primary causes of low cycle fatigue failure which resulted in the leak.

A similar, though less complex failure example, is that of a 3" tee at Consumer's Big Rock Point Plant.[4.5] This component made of ASTM A182-F304 steel joined the 80°F control rod drive hydraulic system bypass and the 450°F pressure vessel cleanup system return line with a thermal sleeve. A leak type failure developed after 4 years operation and the failure investigation disclosed transgranular "craze" cracking both in the tee and an adjacent valve body. The opinion was that poor thermal sleeve design resulted in turbulent flow, alternately cooling and heating pipe wall sections, and consequent thermal fatigue failure. Similar failures are often observed in similar components in which fluids of different temperatures are mixed, such as catapult system desuperheaters in aircraft carriers. Traditionally, the analytical problem is bipassed by thermal shield installation. The efficiency or reliability of a thermal shield design is seldom questioned although, as in this case, it may not be efficacious and also provides an additional structural discontinuity in the system.

The LaCrosse Boiling Water Reactor[4.14] experienced a leak type failure during initial operation from intergranular cracking in furnace sensitized austenitic stainless steel type 304 initiated in the crevice between thermal shield and safe-end of feedwater inlet nozzles in the recirculation header. The largest crack extended 180° around the ID of the safe-end and penetrated the wall at one location. Two of four feedwater nozzles exhibited such cracking and post-failure examination disclosed very shallow cracks in the sensitized material outside of the thermal shield in one nozzle. The second nozzle had a crack which extended 180° around the ID of the safe-end with a maximum depth of 0.16 inch. The thermal shield was installed after stress-relief. The cracked nozzles were attached to piping which was permanently displaced during startup because of interference between a hanger lug and biological shield that restricted pipe expansion. Thus, previous plastic strain may also be a factor in this incident, along with sensitization, the environment, and the crevice.

The Dresden Nuclear Power Station No. 1 6" horizontal bypass line failure is a typical example of unsatisfactory materials properties.[4.6] This failure occurred after 6 years of operation at a 4" secondary steam generator riser branch connection of A312 Type 304 material. An intergranular, circumferential crack developed on a machined surface adjacent to the branch pipe weld in a zone of low service stresses. The cracks which occurred in the 6-inch bypass pipe and the 4-inch decon riser stubs were intergranular in nature. Two were found adjacent to welds made in the shop with a "block" welding procedure. This procedure inherently creates higher strains as well as wider heat-affected zones. It is interesting to note that the crack in the 6-inch elbow-to-pipe initiated in a counterbored area away from the weld fusion line but partially in the heat affected zone. The crack in the 4-inch riser weld occurred, or at least propagated, a considerable distance from the weld in an area with little evidence of sensitization.[4.6, 4.15] This area was said to have been machined.

The 6-inch pipe which cracked showed no evidence of sensitization, but the cracks were intergranular and were presumably caused by environmental corrosion in a severely cold-worked zone.

Thus, two of the three areas where cracking was observed were unsensitized, and one was partially sensitized by the heat of welding. From this, one can postulate that carbide precipitation is not necessary to cause intergranular cracking in Type 304. This has been confirmed in the laboratory.

The common items in these failures are:

1. High levels of plastic strain from restrained welding, from pipe manufacturing, or from machining.

2. The failures occurred in areas of semi-stagnant flow conditions. The significance of this is not immediately apparent but may be related to local environmental effects.

The Garigliano reactor drain line safe end failure is a more typical example of this type of failure because, in this case, it is clear that the austenitic stainless steel was sensitized by pressure vessel stress relief during specified fabrication operations. The area was plastically strained as evidenced by the martensite microstructure. The dissimilar metal construction at such a location provides the requisite stress for slow crack growth.

The BONUS[4.16] superheater piping failures are examples of inadequate quality control and inspection. The initial failures were the result of the use of stainless steel pipe whose chemistry did not meet specifications, was sensitized by fabrication operations, and failed by stress corrosion cracking very shortly after startup. The cracking was largely intergranular, although some small transgranular cracks were noted. Intergranular cracks were observed in both weld sensitized and unsensitized areas. The first failures were observed in material which was not 304 but had higher carbon content. Failures were later observed in 304 material. It was of interest that the failures in unsensitized material occurred in a counterbored area much like the Dresden 6-inch pipe. Because of design, it was also in a stagnant flow area. BONUS had also been subjected to a confirmed high chloride incident.

The Elk River reactor[4.17, 4.18] developed cracks in stainless steel cladding, evaporator tubes, and nozzle to piping attachments attributable to several of the aforementioned factors. The cladding cracks developed on the vessel flange bore. This is clearly intergranular; it occurred in an area which contained no ferrite and even a small amount of martensite. In this case, it was deposited by submerged arc welding and stress relieved with the vessel. Although it is intergranular, it is judged that it is not specifically related to the type of cracking observed in Dresden 1 but rather the result of unique chemistry and metallurgy of this particular weld deposit.

The failure which occurred in the evaporator tubes is of interest since, after this failure, specimens of Type 304 stainless steel, both solution treated and sensitized, were exposed in the primary head of the evaporator. In this case, the sensitized specimens experienced general intergranular attack; whereas, the unsensitized specimens showed evidence of transcrystalline attack. After this experience, a tube was removed from the evaporator which showed no intergranular attack, and a number of welds in the primary circuit were ultrasonically examined with no evidence of failure. This included the welds between the pipe and the furnace-sensitized "safe-ends" on the evaporator.

Subsequently, a primary system leak was discovered and, after extensive search, located.[4.19, 4.20, 4.21]

After the primary system leak was isolated, a field inspection and a laboratory investigation revealed that three intergranular, through-wall cracks had formed in the sensitized stainless steel upper liquid level nozzle extension piece. Since there were other similarly heat treated stainless steel components which were accessible for inspection, a visual, ultrasonic and radiographic inspection program was conducted to determine if there were other potential leaks in the system. A number of other defect indications were found in stainless steel components, some of which proved to be intergranular cracks in sensitized material. The survey indicated that:

1. The cracking appears to be concentrated in the steam phase. Unfortunately, this cannot be statistically documented.
2. Cracks occur in random orientations; e.g., longitudinal, circumferential and at odd angles.
3. Cracks are more frequent in furnace sensitized stainless steel.
4. Cracks are most frequently associated with areas of cold work or high stress (e.g., welding residual stresses).
5. Cracks initiate from either the inside or the outside of the pipe.

There have been also many failures in secondary system components due to poor fabrication practice or unforeseen loadings that are typical of fossil fueled plant failures in field run piping which is not engineered but just built. A considerable number of failures in nuclear plants may have been prevented because of rejection of poorly fabricated or mistreated piping due to the use of more and better inspection than is usual in utility practice. Good workmanship does not require extra inspection effort and is usually obtained in utility construction by their practices and insisted upon for operational economical reasons. The construction of nuclear power plants is not yet a standard utility operation and, for a number of reasons, substandard construction has been the end result in a few instances. Fortunately, the substandard construction has, to date, been located and disposed of in functional or extra requirement testing before startup.

## 4.5 Discussion of Differences Between Nuclear and Fossil Fueled Plants

The significant finding of service experience in nuclear power plant piping is that failures in nuclear plants have been, in type and statistics as far as it is discernible from the limited information available, similar to those in fossil fueled power plants. This finding poses two questions:

1. Is piping serviceability experienced in fossil fueled plants satisfactory for nuclear service?
2. What differences are required in nuclear piping construction by the nature of consequence of failure?

It is first necessary to consider the differences in the piping systems themselves to develop preliminary answers to those questions.[4.7 thru 4.11]

The first obvious difference in the power plant piping systems is the structural analysis problem derived from geometric differences in the piping. These differences derive from the boiler, biological shielding, and containment geometries that restrict equipment location and piping layout design to avoid the problem plus some additional requirements to prevent release of radioactive material or other nuclear incidents as the result of unusual occurrences such as earthquakes, tsunamis, tornados, and so on required for public protection. The current piping program is designed to obtain the requisite load response and stress analysis information required by those geometric differences. This problem is largely limited to the research and first-of-a-kind plant and, as has been the case with other systems, will dwindle as knowledge is gained by experiment and experience. Prediction of structural behavior in response to service loadings should be simpler for nuclear plants, when the presently lacking information on flexibilities and stress distribution is acquired, than for other plants because of the imposed operational requirements. The development of loading input for unusual incidents is far more difficult but not considered herein.

The second difference is in materials of construction. Austenitic stainless steel, for example, is currently specified in nuclear plants for pressure and temperature conditions for which carbon or low alloy steels are used in fossil fueled plants. In some instances, stainless cladding is used with the conventional materials. The reason for this specification, of course, is process derived to reduce corrosion and corrosion products and, as an incidental attribute, to reduce the likelihood of fast fracture. The failures previously reported demonstrate that fabrication processing of stainless steel materials can result in stress corrosion or other stress and environment enhanced cracking in the nuclear coolant environment. Other information demonstrates that serious damage to the stainless steel materials may occur during handling, cleaning, and storage. Stainless steel cladding is particularly sensitive to fabrication processing, unforeseen environments, and dissimilar metal stresses which result in cladding cracks. It has been demonstrated[4.22] that cracks in stainless steel cladding degrade serviceability of base material as severely as any other crack and that the electrochemical behavior at the dissimilar metal interface in nuclear boiler water further accelerates crack propagation.[4.23, 4.24, 4.25, 4.26]

It should be remembered that nuclear piping is designed for finite service life and that a design goal is not to have unacceptable failures which reduce availability during this period. The requirement for an early failure is a sharp notch that can propagate by fatigue, stress corrosion or other mechanism from service loadings and environments.

The examples presented all concern stainless steel in plants whose water chemistry is deionized water with 0.2 to 0.3 ppm $O_2$. Evidence exists that candidate substitute materials, such as Inconel[4.27, 4.28, 4.29, 4.30, 4.31] can behave similarly. An aggravating feature of observed behavior is the difficulty of obtaining laboratory results duplicating service behavior to enable prediction of failure. We cannot feel secure that failures may not occur in other materials and environments until adequate service experience is gained. Experience indicates that failures, such as those previously discussed, may occur with increasing frequency along with increased nuclear plant operational hours. Unfortunately, this problem has not been recognized in time to develop the information from research investigations essential to establishing cause and cure relationships. A particular item of concern is valve and pump bodies whose design, soundness evaluation, and metallurgical properties are far more complex and less well understood than are piping sections. The failures of these components, to date, have been limited to mechanical malfunction but the potential of structural failure cannot be dismissed. The alterations in material chemistry required for casting fluidity, for example, are those which are avoided in piping fabrication because they have been found to result in sensitivity to environment in piping applications and the repair and soundness of these items is not monitored as well as in piping. However, piping sections pass through and are operated on by many more hands and with less documentation than are individual system components such as pressure vessels, valves, or pumps.

At the very least, there are involved three operations, pipe fabrication, shop assembly, and field installation which affect the properties that determine serviceability and are not monitored in detail as are similar operations in pressure vessel fabrication. Each piping material has specific properties that are affected by fabrication operations. The sensitive properties of stainless steels are their environmental resistance and for the carbon and low alloy steels, their fracture toughness. In the latter case, the possibility of low energy fast fracture at high temperatures relative to nil ductility transition temperatures must be recognized and guarded against by appropriate specifications including materials test requirements. The best example appropriate to this concern is the frangible failure of the piping section of a nozzle of PVRC vessel number 7. This component, a 2-1/4 Cr-1 Mo forging which is a material used in the piping of a research reactor and a candidate material for utility reactor systems, fragmented completely (360° break) at a nominal stress level in operating stress range for nuclear systems from a minute flaw which would not be detected in service or by usual inspection procedures and away from geometric discontinuities.[4.12] The fast fractures experienced in the PVRC full size and half scale test series at Southwest Research Institute demonstrate that current carbon and low alloy steel piping component specification requirements are not adequate safeguards against installation of material of inadequate fracture toughness just as they are not adequate to eliminate stainless steel materials with inadequate environmental resistance.

The third difference is in the properties of the process fluid. Fossil fueled power plant water properties differ significantly from nuclear reactor water properties. The radioactivity level does not affect materials properties but it does affect failure criteria and preventive maintenance inspection procedures. A design criteria for nuclear plants is leak before break behavior and leak detection is the primary method of hardware integrity surveillance in the nuclear plant. Leaks can be tolerated in fossil fueled power plants, observed, and repair properly scheduled. A leak in the nuclear plant means immediate and costly shutdown, often a long and costly search to locate it, and, usually a considerable engineering effort to effect a repair. This is only one aspect of the chemistry of the process fluid that is important, however. Fossil fueled power plant water chemistry is carefully tailored to protect the hardware by pH control, oxygen scavenging, etc. while nuclear power plants are forced to operate with "high-purity, deionized" water that has certain properties which have been found to be harmful to candidate materials of construction. Radiolytic decomposition products and contaminants provide chloride ions and oxygen concentrations in many reactors that are excessive for as-fabricated stainless steel materials properties, for example, while the pH level is too low for corrosion inhibition of carbon and low alloy steels. False ideas of the combined effects of stress and nuclear reactor coolant water on material serviceability were developed by experiments that did not evaluate all of the variables which affect the behavior of real hardware which resulted in the selection of materials in current use. Failure experience, such as the examples previously cited, and surveys of actual materials properties and water chemistries demonstrate the need for more sophisticated problem evaluation, design, and specifications to increase piping system serviceability.

The fourth, and last difference, in nuclear and fossil fueled power plant piping system is in the definition of failure and surveillance accessibility.[4.13] These system features are lumped together, because they derive from radiation hazard, and while they have been referred to previously, are of major importance, with respect to the economics of nuclear power plants, and so deserve separate treatment. The availability of each power plant in a utility network is an important factor in the operational costs of the utility. Lengthy shutdowns of a plant for inspection or repair are very costly. The definition of failure, that requires a shutdown, in a nuclear plant is a leak in the piping system and the discovery and repair of this leak is likely to require a long period of time. A comparable leak in a fossil fueled power plant could probably be tolerated in operation and repaired without interfering with plant availability or at least repaired during a scheduled shutdown. Surveillance of fossil fueled power plant piping can be accomplished by visual observation during operation. For the nuclear plant, limited areas can be inspected only by remote observation techniques and only during shutdown. This is also a costly operation because it requires reduction in plant availability as well as costly equipment and manpower. If repair is found to be necessary, another expensive operation is required, both in loss of availability and in performance. It is not facetious to say that the entire cost of a simple repair in a fossil fueled piping system is probably less than the editorial costs essential for reporting a similar failure and repair in a nuclear plant piping system. One of the failure examples given previously would have been a normal maintenance procedure not costed in a fossil fueled plant because its repair would not interfere with operation while the inspection and repair, based on lost revenue alone, would cost a utility almost half a million dollars in a large nuclear plant because of the extraordinary amount of time required to locate and effect the repair and to assure that a similar failure was not imminent. Surveillance is a major problem in nuclear piping systems today. The term surveillance is used, in this case, to mean nondestructive inspection that does not interfere with operation or plant availability to detect and interdict failure. Essentially, adequate surveillance is detection of a cracklike flaw before it initiates a failure by any mode and the determination, or prediction, of flaw growth behavior in terms of system operational parameters. The objective of surveillance is to preclude failure and to schedule repair so as not to interfere with plant availability. Service experience demonstrates that terminal growth to failure is by the low cycle fatigue mechanism. An essential element of the surveillance procedure is therefore a method of predicting fatigue life. The input to such a procedure must include the characterization of cracklike flaws in the hardware, the load induced stresses at the flaw locations, the relevant materials properties in the areas where the flaws reside, and the environmental effects on flaw growth. The major area of ignorance is on the effects of environment. The major primary difficulty, in real hardware, is inspectability of suspect areas. The criteria for selection of locations to inspect must be based on likelihood of existence of fabrication flaws and defects, highest operational stresses or strains, and poorest materials properties. The criteria for locations that can be inspected economically derive from structural geometry of system components. These criteria are usually not compatible and it is now necessary to devise engineering compromises whereby certain critical areas are sampled as indices of system performance.

The answers to the two questions that initiated this discussion of the differences between nuclear and fossil fueled power plants have been delineated:

1. Piping serviceability must be considerably improved in the nuclear plant compared with the fossil fueled plant to increase plant availability. This has not yet been accomplished because the service record of nuclear and fossil fueled plant piping systems are near identical.

2. The differences required in nuclear piping construction by the consequences of failure are in the area of assurance against leak type failure by the usual requirements for superior design and inspection and the novel requirement for design for inspection and surveillance which includes consideration of accessory design geometries such as biological shielding and component location. The basis for satisfactory operation must include:

a. Definition of potential modes of failure.

b. Synthesis of analytical models of structural response to loads and environments of system elements for the service life of these elements including the terminal failure event.

c. Development of means of detection of the occurrence of the precursory events.

d. Delineation of the characteristics of instrumentation hardware suitable for surveillance and monitoring systems.

This basis permits interdiction of unscheduled failure by repair during scheduled plant outage, such as for refueling. The potential mode of failure observed and considered likely is leak type failure developed by low cycle fatigue. The failure precursor event is a propagating crack of a size defined by loading, materials properties, and operating schedule. Presently, ultrasonic inspection is used to detect crack growth and define crack size. It is essential to have a method of predicting crack growth from information available to the designer. Methods for predicting crack growth are discussed in Chapter 3.

# 4. REFERENCES

4.1 Gibbons, W. S., and Hackney, B. D., "Survey of Piping Failures for the Reactor Primary Coolant Pipe Rupture Study," GEAP-4574, Reactor Technology (TID-4500, 41st Ed.), May, 1964.

4.2 Irvine, W. H., "The Application of Fracture Mechanics to the Design, Manufacture and Operation of Critical Pressure Components," Authority Health and Safety Branch, Risley, Warrington, Lancashire, 1968.

4.3 Thielsch, H., "Failures in Critical Piping Systems Correlated to Nuclear Power Plant Service," ASME Winter Meeting, Philadelphia, Pa., 1963.

4.4 Smith, W. R., Sr. and Violette, J. B., "Summary Report of the Cause of Failure of the VBWR Recirculating Pipe," October 22, 1962.

4.5 Higgins, J. P., "Evaluation of Failed 3-Inch Tee at Consumers Big Rock Point Plant," CECR-5401, January, 1967, Cl I-67APE1.

4.6 Higgins, J. P., and Walker, W. L., "Investigation of Stainless Steel Pipe Failures at DNPS No. 1," APED-5170, Cl II, July, 1966.

4.7 Tagart, S. W., Jr., "Mechanical Design Considerations in Primary Nuclear Piping," GEAP-4578, Reactor Technology (TID-4500, 41st Ed.), March, 1964.

4.8 Kilsby, E. R., Jr., "A Survey of Water Reactor Primary System Conditions Pertinent to the Study of Pipe Rupture," GEAP-4445, Reactor Technology (TID-4500, 41st Ed.), January, 1964.

4.9 Brothers, A. J., Coffin, L. F., Jr., Klepfer, H. H., Low, J. R., Jr., and Reynolds, M. B., "A Review of Fracture Modes As Related to Reactor Primary Coolant Pipe Rupture," GEAP-4446, May, 1964.

4.10 Reynolds, M. B., "Fracture Mechanics and the Stability of Engineering Structures," GEAP-4678, AEC Research and Development Report, September, 1965.

4.11 Klepfer, H. H., "Experimental and Analytical Program Recommendations," GEAP-4474, Reactor Technology (TID-4500, 41st Ed.), January, 1965.

4.12 Pickett, A. G., and Grigory, S. C., "Studies of the Fatigue Strength of Pressure Vessels," Technical Summary Report to USAEC, September, 1966.

4.13 Irvine, W. H., "Developments in the Periodic Inspection of Primary Circuit Pressure Components of Nuclear Reactors," Authority Health & Safety Branch, United Kingdom Atomic Energy Authority, 1966.

4.14 Burghard, H. C., Jr., and Lyle, F. F., Jr., "Metallurgical Investigation of Cracking in LACBWR Nozzle Safe-Ends, SwRI Interim Report, Dec. 26, 1969.

4.15 Johnson, C. R., and Shield, W. A., "Metallurgical Examination of 6-Inch Stainless Steel Pipe from Dresden 1, "BAW-TM-215, Babcock and Wilcox Co., Nuclear Development Center, Lynchberg, Virginia, Nov. 1967.

4.16 Jameson, A. S., "Metallographic Investigations and Reports of the BONUS Steam Dryer Preheater Piping," Combustion Engineering Co. Report CEND-284, AEC Contract AT(40-1)-2674, Nov. 1966.

4.17 Norris, E. V., et al, "The Second Inspection of the Elk River Reactor Pressure Vessel After Operation," SwRI Report 1228P9-13, AEC Contract AT(11-1)-1228, Project Agreement No. 9, July 1967.

4.18 Bergen, C. R., "Corrosion Samples and Tests Evaporator Water Boxes," Allis-Chalmers Manufacturing Company, Final Report, Task 615, ACNP 66542, June 1966.

4.19 Burghard, H. C., Jr., Norris, E. G., and Wylie, R. D., "Examination of Upper Liquid Level Line from Elk River Reactor," USAEC Report SwRI 1228 pg. 16, Feb. 27, 1969.

4.20 Lautzenheiser, C. E., and Viaclovsky, S. A., "Ultrasonic Inspection of Selected Welds in Stainless Steel Primary Steam Piping in Elk River Power Plant" USAEC Report SwRI 1228-9-10, June 29, 1966.

4.21 Lautzenheiser, C. E., and Viaclovsky, S. A., "Second Ultrasonic Inspection of Selected Welds in Stainless Steel Primary Steam Piping in Elk River Power Plant," USAEC Report SwRI 1228 pg. 12, June 7, 1967.

4.22 "Design and Fabrication of the BONUS Reactor Pressure Vessel" GNEC 210, Supplement 2, Dec. 21, 1962.

4.23 Chapter 5. Effect of Environment on Materials. "Technology of Steel Pressure Vessels for Water-Cooled Nuclear Reactors," ORNL-NSIC-21, Dec. 1967.

4.24 "A Summary Report of the Hairline Cracks on the Clad of the JPDR Pressure Vessel," Japan Atomic Energy Research Institute, April 1968.

4.25 Kondo, T., "Metallographic Analysis of the Cladding Failure in JPDR Pressure Vessel," Japan Atomic Energy Research Institute, Report No. JP-M-2, May 9, 1968.

4.26 Kondo, T., "Metallographic Analyses of the Cladding Failure in JPDR Pressure Vessel-I," Japan Atomic Energy Research Institute, Report No. JP-M-1, April 10, 1968.

4.27 Coriou, H., Grall, L., et al, "Corrosion Fissurante sous contrainte de l'Inconel dans l'eau a haute temperature" Colloque de metallurgie." Corrosion 3. Saclay 29-30 Juin, 1 Juillet 1959, Amsterdam, Holland Publ. 1960 p. 161-168.

4.28 Coriou, H., Grall, L., et al, "Fissuration sous contrainte en milieu et dans 1 eau pure a haute temperature d alliages inoxydables au nickel." Colloque de la Societe Europeenne d Energie Atomique, Studsvik 22-23 mai 1962.

4.29 Coriou, H., and Grall, L., "Problemes de corrosion aqueuse de materiaux de structure dans les constructions nucleaires." 1964. (CEA-R-2600).

4.30 Gronwall, B., Ljungberg, L., Hubner, W., and Stuart, W., "Intercrystalline Stress Corrosion Cracking of Inconel 600 Inspection Tubes in the Agesta Reactor, " AE-245 Aktiebolaget Atomenergi Stockholm, Sweden, 1966.

4.31 Coriou, H., Grall, L., and Pelras, M., "Study of Chrome-Nickel-Iron Austenites for their Corrosion Behavior from the Standpoint of their Application in the Nuclear Field," 3rd International Congress on Metal Corrosion, Moscow, May 16-25, 1966.

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# 5. CODES AND STANDARDS

There are a large number of documents concerning piping design requirements which have some degree of national* recognition. These documents may be referred to as codes, standards, regulations, rules, specifications, standard practices, etc. In this Chapter,† a listing by sponsoring organization is given, followed by a brief description of the type of documents published by the various organizations. In the second part of this Chapter, a discussion of the inter-relationship and contents of some of the most pertinent documents is given.

## 5.1 Sponsoring Organizations

| | |
|---|---|
| USASI | USA Standards Institute<br>10 E. 40th Street, New York, N.Y. 10016 |
| ASME | American Society of Mechanical Engineers<br>345 E. 47th Street, New York, N.Y. 10017 |
| ASTM | American Society for Testing and Materials<br>1916 Race Street, Philadelphia, Pa. 19103 |
| MSS | Manufacturers Standardization Society of the Valve and Fittings Industry, 420 Lexington Avenue, New York, N.Y. 10017 |
| API | American Petroleum Institute<br>1271 Avenue of the Americas, New York, N.Y. 10020 |
| AWWA | American Water Works Association<br>2 Park Avenue, New York, N.Y. 10016 |
| FSSC | Federal Specification: Superintendent of Documents<br>U.S. Government Printing Office, Washington, D.C. 20402 |

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* This Chapter covers only U.S.A. codes and standards. Other countries have analogous documents and the International Standards Organization (ISO) also sponsors pertinent codes and standards.

† A much broader list of U.S. Nuclear Standards has been completed by USAS Subcommittee N6.9; See Nuclear Safety, Vol. 6, No. 4, pp 354-367 (1965); Vol. 7 No. 4, pp 415-417 (1966)

PFI Pipe Fabricators Institute
992 Perry Highway, Pittsburgh, Pa. 15237

AEC-RDT U.S. Atomic Energy Commission, Div. Reactor Development & Technology, RDT Standards Program, Oak Ridge National Laboratory, Oak Ridge Tennessee 37830

In addition, a number of government agencies publish pertinent documents, primarily for internal use. Some of these are: U.S. Navy, U.S. Navy Bureau of Ships, U.S. Coast Guard, American Bureau of Shipping, and the Department of Commerce.

There is a considerable amount of duplication of documents between the various organizations. For example, ASME Section II duplicates many ASTM standards; USASI A21,-standards are, in part, duplicates of a number of AWWA standards; a number of USASI B36.-standards are duplicates of ASTM standards, etc.

## 5.11 USASI

USASI standards cover a wide variety of subjects, only a few of which are of interest herein. USASI standards are identified by prefix letters, followed by several numbers. The prefix letter B indicates standards on Mechanical Engineering. There are several groups of standards within this classification which are of particular interest herein. These are:

B2.—:Pipe and hose coupling threads (3 standards)

B16.—:Pipe Flanges and Flanged Fittings (24 standards)

B31.—:Pressure Piping (8 standards)

B36.—:Iron and steel pipe (38 standards, all except 2 of which are duplicates of ASTM standards)

There are also 12 standards with the prefix letter A (Civil Engineering) which are pertinent herein. These are on cast-iron pipe and fittings and on ductile iron pipe.

These standards are also frequently referred to as the USAS, ASA or ANSI codes. The latter designation is the result of a recent change in the name of the sponsoring organization to the American National Standards Institute (ANSI).

5.12 ASME

The ASME document of particular interest herein is the ASME Boiler Code. This code is divided into seven sections:

I Power Boiler

II Material Specifications (Duplicates of some ASTM specifications)

III Nuclear Vessels

IV Low-Pressure Heating Boilers

VII Suggested Rules for Care of Power Boilers

VIII Unfired Pressure Vessels, Divisions 1 and 2

IX Welding Qualifications

Another pertinent ASME Code is that entitled, "ASME Standard Code for Pumps and Valves for Nuclear Power" (November, 1969). While this code is labeled as "Issued for Trial Use and Comment" and "Tentative, Subject to Revision", it apparently has official status in that it is invoked by ASME Section III (see paragraph N-153, Summer 1969 Addenda).

At this time (September, 1969), work is underway to revise ASME Section III to cover a much broader scope. A tentative outline of the proposed new Section III is shown in Table 5.1. The new version is intended to cover piping, pumps and valves either by reference to the USAS B31.7 "Code for Nuclear Power Piping" and to the ASME Code for Pumps

TABLE 5.1 : TENTATIVE OUTLINE OF REVISED ASME SECTION III

SECTION III - Nuclear Power Plant Components

<u>Division 1</u> - Metal Components

Subsection A - General Requirements

B - Class I Components

C - Class II Components

D - Class III Components

E - Containment Components

F - Component Supports

G - Core Support Structures

H - Shipping Storage

I - Installation of Components

<u>Division 2</u> - Concrete Vessels

Subsection A - Prestressed Concrete Reactor Vessels

B - Concrete Containment Vessels (ACI Code)

<u>Division 3</u> - Recommended Rules for Service

Subsection A - In-service Inspection for Pressure Integrity

B - In-service Performance Relative to Reliability and Valves, or by writing analogous rules in the appropriate subsections of ASME Section III.

Tentative plans have also been made to prepare ASME Section V on Nondestructive Methods of Examination.

## 5.13 ASTM

ASTM specifications are concerned with requirements for material properties and appropriate test methods for establishing those properties. The 1968 "Book of ASTM Standards" consists of 32 parts. With regard to piping components, Part I, "Steel Piping, Tubing and Fittings", includes most of the pertinent standards under one cover.

## 5.14 MSS

MSS publishes (as of July 1, 1965) 26 "Standard Practices", all of which are related to piping components. Some of the standard practices developed by MSS are reviewed and adopted by USASI, in which case the standard practice is discontinued as an MSS document.

## 5.15 API

API publishes a number of standards, a few of which are directly or indirectly related to piping components. The most significant of these, in the present context, are standards 600, 602, 603, and 6D (steel valves); 604 (nodular iron valves); 605 (large diameter flanges); 5L and 5LX (line pipe).

## 5.16 AWWA

AWWA publishes a number of standards related to piping for water service. C100 series is on cast-iron pipe and fittings (these 8 standards are also published by USASI); C200 series is on steel pipe, flanges and fittings; C300 series is on concrete pipe; C400 is on asbestos-cement pipe; C500 is on valves and hydrants, while C600 is on pipe-laying practices.

## 5.17 FSSC

Federal specifications are available for a variety of piping, tubing, and pipe or tubing components (WW——specifications) as well as for certain materials (QQ——specifications).

## 5.18 PFI

PFI publishes (as of Jan., 1968) twenty standards relating to fabrication, inspection and testing of piping systems. These standards do not cover piping components or material properties.

## 5.19 AEC - RDT

The U. S. Atomic Energy Commission, Division of Reactor Development and Technology (RDT) has issued 65 RDT standards as of July 31, 1969. These standards cover various aspects of the planning, building and operation of water-cooled nuclear reactors. Standards pertaining to both water and liquid-metal cooled reactors are being developed jointly with the Liquid Metals Engineering Center. A list of the RDT standards is given in Table 5.2.

TABLE 5.2: RDT STANDARDS ISSUED THROUGH JULY 31, 1969



| NUMBER | TITLE | ISSUE DATE |
|---|---|---|
| E-SERIES | EQUIPMENT, MECHANICAL, FLUID | |
| E 4- 1T | STEAM GENERATOR FOR PRESSURIZED WATER REACTORS | 2/69 |
| F-SERIES | PROGRAMS, PROCEDURES, METHODS | |
| F 2- 2T | QUALITY ASSURANCE-PROGRAM REQUIREMENTS | 6/69 |
| F 3- 1T | INSPECTION SYSTEM REQUIREMENTS | 2/69 |
| F 3- 2T | CALIBRATION SYSTEM REQUIREMENTS | 2/69 |
| F 3- 3T | ULTRASONIC EXAMINATION OF HEAVY STEAL FORGINGS (MODIFIED ASTM A388) | 2/69 |
| F 3- 4T | ULTRASONIC SHEAR-WAVE EXAMINATION OF PLATES (MODIFIED ASTM A577) | 2/69 |
| F 3- 5T | LONGITUDINAL-WAVE ULTRASONIC EXAMINATION OF PLAIN AND CLAD STEEL PLATES (MODIFIED ASTM A578) | 2/69 |
| F 3- 6T | NONDESTRUCTIVE EXAMINATION | 3/69 |
| F 3- 7T | INSPECTION REQUIREMENTS FOR MATERIALS IN WEAR APPLICATIONS | 2/69 |
| F 3- 8T | ULTRASONIC EXAMINATION OF METAL PIPE AND TUBING FOR LONGITUDINAL DISCONTINUITIES (MODIFIED ASTM E213) | 2/69 |
| F 5- 1T | CLEANING AND CLEANLINESS REQUIREMENTS FOR NUCLEAR REACTOR COMPONENTS | 4/69 |
| F 6- 1T | WELDING | 2/69 |
| F 7- 1T | CONTINUOUS IDENTIFICATION MARKING OF WROUGHT PRODUCTS | 2/69 |
| F 7- 2T | PREPARATIONS FOR SEALING, PACKAGING, PACKING, AND MARKING OF COMPONENTS FOR SHIPMENT AND STORAGE | 2/69 |
| F 7- 3T | REQUIREMENTS FOR IDENTIFICATION MARKING OF REACTOR PLANT COMPONENTS AND PIPING | 2/69 |
| F 8- 1T | PRELOADING THREADED FASTENERS AND CLOSURES | 2/69 |
| M-SERIES | MATERIALS | |
| M 1- 1T | CORROSION RESISTING CHROMIUM AND CHROMIUM-NICKEL STEEL COVERED WELDING ELECTRODES (MODIFIED ASTM A298) | 2/69 |
| M 1- 2T | CORROSION RESISTING CHROMIUM AND CHROMIUM-NICKEL STEEL WELDING RODS AND BARE ELECTRODES (MODIFIED ASTM A371) | 2/69 |
| M 1- 3T | MILD STEEL COVERED ARC-WELDING ELECTRODES (MODIFIED ASTM A233) | 2/69 |
| M 1- 4T | LOW ALLOY STEEL COVERED ARC-WELDING ELECTRODES (MODIFIED ASTM A316) | 2/69 |

| NUMBER | TITLE | ISSUE DATE |
|---|---|---|
| M 1- 5T | SURFACING WELDING RODS AND ELECTRODES (MODIFIED ASTM A399) | 2/69 |
| M 1- 6T | MILD STEEL ELECTRODES AND WELDING RODS FOR GAS-METAL ARC-WELDING (MODIFIED ASTM A559) | 2/69 |
| M 1- 7T | COPPER AND COPPER ALLOY ARC-WELDING ELECTRODES (MODIFIED ASTM B225) | 2/69 |
| M 1- 8T | COPPER AND COPPER ALLOY WELDING RODS (MODIFIED ASTM B259) | 2/69 |
| M 1- 9T | BRAZING FILLER METAL (MODIFIED ASTM B260) | 2/69 |
| M 1-10T | NICKEL AND NICKEL ALLOY COVERED WELDING ELECTRODES (MODIFIED ASTM B295) | 2/69 |
| M 1-11T | NICKEL AND NICKEL ALLOY BARE WELDING RODS AND ELECTRODES (MODIFIED ASTM B304) | 2/69 |
| M 2- 1T | CARBON STEEL FORGINGS (MODIFIED ASTM A105) | 2/69 |
| M 2- 2T | STAINLESS AND HEAT-RESISTING STEEL FORGINGS (MODIFIED ASTM A182) | 2/69 |
| M 2- 3T | WROUGHT SEAMLESS CARBON STEEL WELDING FITTINGS (MODIFIED ASTM A234) | 2/69 |
| M 2- 4T | ALLOY STEEL FORGINGS (MODIFIED ASTM A336) | 2/69 |
| M 2- 5T | FACTORY MADE WROUGHT AUSTENITIC STEEL WELDING FITTINGS (MODIFIED ASTM A403) | 2/69 |
| M 2- 6T | LOW-CARBON CHROMIUM STEEL FORGINGS (MODIFIED ASTM A473) | 2/69 |
| M 2- 7T | QUENCHED AND TEMPERED VACUUM-TREATED CARBON AND ALLOY STEEL FORGINGS (MODIFIED ASTM A508) | 2/69 |
| M 2- 8T | QUENCHED AND TEMPERED ALLOY STEEL FORGINGS FOR PRESSURE VESSEL COMPONENTS (MODIFIED ASTM A541) | 2/69 |
| M 3- 1T | SEAMLESS CARBON STEEL PIPE FOR HIGH TEMPERATURE SERVICE (MODIFIED ASTM A106) | 2/69 |
| M 3- 2T | SEAMLESS FERRITIC AND AUSTENIC ALLOY STEEL TUBES (MODIFIED ASTM A213) | 2/69 |
| M 3- 3T | SEAMLESS AUSTENITIC STAINLESS STEEL PIPE (MODIFIED ASTM A376) | 2/69 |
| M 3- 4T | SEAMLESS ANNEALED NICKEL-CHROMIUM-IRON AND NICKEL-IRON-CHROMIUM ALLOY CONDENSER AND HEAT EXCHANGER TUBES (MODIFIED ASTM B163) | 2/69 |
| M 3-12T | SEAMLESS FERRITIC ALLOY STEEL PIPE (MODIFIED ASTM A335) | 2/69 |
| M 4- 1T | CARBON STEEL CASTINGS FOR FUSION WELDING AND HIGH TEMPERATURE SERVICE (MODIFIED ASTM A216) | 2/69 |
| M 4- 2T | AUSTENITIC STAINLESS STEEL CASTINGS FOR HIGH TEMPERATURE SERVICE (MODIFIED ASTM A351) | 2/69 |
| M 4- 3T | COBALT-CHROMIUM ALLOY CASTINGS, WEAR AND CORROSION RESISTANT | 2/69 |
| M 5- 1T | CHROMIUM AND CHROMIUM-NICKEL STAINLESS STEEL PLATE, SHEET, AND STRIP (MODIFIED ASTM A240) | 2/69 |
| M 5- 2T | CARBON STEEL PLATES FOR PRESSURE VESSELS FOR MODERATE AND LOW TEMPERATURE SERVICE (MODIFIED ASTM A516) | 2/69 |

| NUMBER | TITLE | ISSUE DATE |
|---|---|---|
| M 5- 3T | MANGANESE-MOLYBDENUM AND MANGANESE-MOLYBDENUM-NICKEL ALLOY STEEL PLATES (MODIFIED ASTM A533) | 2/69 |
| M 5- 4T | NICKEL-CHROMIUM-IRON ALLOY PLATE, SHEET, AND STRIP (MODIFIED ASTM B168) | 2/69 |
| M 5- 5T | CHROMIUM-MOLYBDENUM ALLOY STEEL PLATE (MODIFIED ASTM A387) | 2/69 |
| M 6- 1T | ALLOY STEEL BOLTING MATERIALS FOR LOW TEMPERATURE SERVICE (MODIFIED ASTM A320) | 2/69 |
| M 6- 2T | MECHANICAL LOCKING DEVICES | 3/69 |
| M 6- 3T | ALLOY STEEL BOLTING MATERIALS FOR HIGH TEMPERATURE SERVICE (MODIFIED ASTM A193) | 2/69 |
| M 7- 1T | LOW CARBON CHROMIUM STEEL BARS (MODIFIED ASTM A276) | 2/69 |
| M 7- 2T | NICKEL-CHROMIUM-IRON AGE-HARDENABLE ALLOY BARS, RODS, AND FORGINGS (MODIFIED ASTM A461) | 2/69 |
| M 7- 3T | STAINLESS AND HEAT-RESISTING STEEL BARS AND SHAPES (MODIFIED ASTM A479) | 2/69 |
| M 7- 4T | NICKEL-CHROMIUM-IRON ALLOY RODS, BAR, AND FORGINGS (MODIFIED ASTM B166) | 2/69 |
| M 7- 5T | WROUGHT COBALT-CHROMIUM-TUNGSTEN-NICKEL ALLOY ROUNDS | 2/69 |
| M 7- 6T | CHROMIUM-NICKEL STEEL BARS AND FORGINGS, CORROSION RESISTANT PRECIPITATION HARDENING | 2/69 |
| M 7- 7T | COBALT-CHROMIUM ALLOY BARS AND SHAPES | 2/69 |
| M 7- 8T | ALLOY WIRE, CORROSION AND HEAT RESISTANT, NICKEL BASE ANNEALED | 2/69 |
| M 8- 1T | HELICAL AGE-HARDENABLE NICKEL-CHROMIUM-IRON ALLOY SPRINGS | 2/69 |
| M 9- 1T | MATERIALS AND FABRICATION REQUIREMENTS FOR NICKEL-CHROMIUM-IRON ALLOY SEAL APPLICATIONS | 2/69 |
| M 9- 2T | EXAMINATION REQUIREMENTS AND ACCEPTABLE STANDARDS FOR SEAL MEMBRANES | 2/69 |
| M11- 1T | NONMETALLIC SEAL MATERIALS | 2/69 |
| M11- 2T | IMPREGNATED ASBESTOS PACKING MATERIAL | 2/69 |
| M12- 1T | TEST REQUIREMENTS FOR THERMAL INSULATING MATERIALS FOR USE ON AUSTENITIC STAINLESS STEELS | 2/69 |

## 5.2 Inter-relationships

The design of piping systems, as well as piping components, is covered by the USASI Code for Pressure Piping. This code is divided into eight sections:

| Section | Title |
|---|---|
| 1 | Power Piping, B31.1.0-1967+ |
| 2 | Industrial Fuel Gas Piping, B31.2-1968 |
| 3 | Petroleum Refinery Piping, B31.3-1966 |
| 4 | Oil Transportation Piping, B31.4-1959 |
| 5 | Refrigeration Piping, B31.5-1962 |
| 6 | Chemical Industry Process Piping* |
| 7 | Nuclear Power Piping |
| 8 | Gas Transmission and Distribution Piping Systems, B31.8-1968 |

Certain sections of the B31. Codes are analogous to sections of the ASME Boiler Code in allowable stress levels and general design philosophy; i.e.,

| ANSI B31. | ASME Boiler Code |
|---|---|
| Section 1 | Section I |
| Section 3 | Section VIII |
| Section 7 | Section III |

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+ In 1955 (and several editions prior thereto), several sections were published under a single cover as the Code for Pressure Piping, ASA B31.1-1955. B31.1.0-1967 is <u>not</u> equivalent to the former B31.1-1955.

* Not published, consult ASA B31 Case No. 49, April, 1961.

It might be noted that the ASME Boiler Code may not include certain piping. Nevertheless, when certain design details are not covered in USAS B31 Codes, it is common practice to use the Boiler Code rules for guidance in such areas.

Piping Components can be placed in two categories:

(1) Components not included in an accepted standard; e.g., fabricated branch connections, mitered pipe bends, special bolted-flanged joints.

(2) Components manufactured and sold as meeting the requirements of accepted standards; e.g., butt-welding end fittings to USAS B16.9, flanged fittings to USAS B16.5, socket-welded fittings to USAS B16.11.

Components in the first category are designed individually to meet the stress limitations and detailed requirements contained in the pertinent B31. Code Section or ASME Boiler Code Section. Components in the second category are included in B31. Sections or ASME Boiler Code Sections by reference to component standards. Both categories of components are subject to material requirements through reference to ASTM standards or specifications. The inter-relationship is as sketched below:

{ASME Sections / B31. Sections}

Component Standards → ASTM Standards (Materials)

ANSI
MSS
API
AWWA

Two points concerning the inter-relationship between B31. Sections and component standards should be noted.

(1) Component standards are not automatically included in B31. Sections; they must be specifically accepted. Restrictions to the use of such standard components may be placed in the B31. Sections.

(2) Component standards generally serve a dual purpose:

(a) They establish certain "fit-up" dimensions (center-to-end, bolting, etc.) so that components of various manufacturers are interchangeable.

(b) They establish certain pressure ratings. Insofar as the writer is aware, none of the component standards explicitly include any loading other than static internal pressure. Certain limits to bending moments on common types of pipe fittings are included in the B31. Sections as a result of the "Expansion and Flexibility" rules. No analogous limit exists for valves. One must assume that the B31. Sections accept such valve ratings on the basis of generally acceptable experience, rather than any explicit requirement that a valve should be capable of withstanding some stated combination of moment loads and thermal loads along with the rated pressure load.

## 5.3 Dimensional Controls

From a structural analysis standpoint, it should be recognized that "standard" piping components are, with few exceptions, not completely standardized. The dimensional controls of USAS B16.9, "Wrought Steel Buttwelding Fittings", are typical. This standard gives:

(a) Center-to-ends or overall dimensions with tolerances

(b) Outside and inside diameters at ends, with tolerances

(c) Angularity tolerance of end planes

(d) Minimum wall thickness, anywhere in fitting.

For structural analysis, what is not specified is:

(A) Shape of fitting

(B) Wall thickness of fitting, except at ends.

With regard to shape of the fitting, Figure 5.1 shows some extremes of shapes permitted under B16.9. The shape obtained will depend upon the particular manufacturer and his manufacturing process and controls at the time he is making the fitting.

With regard to wall thickness of the fitting, only a minimum thickness is specified. The minimum specified is 0.875 times the nominal thickness of the pipe with which the fitting is recommended for use. For example, a 12" std. wt. tee would have a minimum required thickness of 0.875 x 0.375" = 0.328". It is well known by most manufacturers that a straight tee, at least in certain areas, must be made significantly thicker than the matching pipe, just in order to pass the required burst pressure test. Actual thicknesses of B16.9 tees will depend upon the particular manufacturer's decision in this respect. The manufacturer may or may not be influenced in this decision by consideration of loadings other than internal pressure. A second aspect of wall-thickness control is that because only a minimum is specified, the fitting may be significantly thicker than anticipated by the users. For example, a user may order a 12" std. wt.

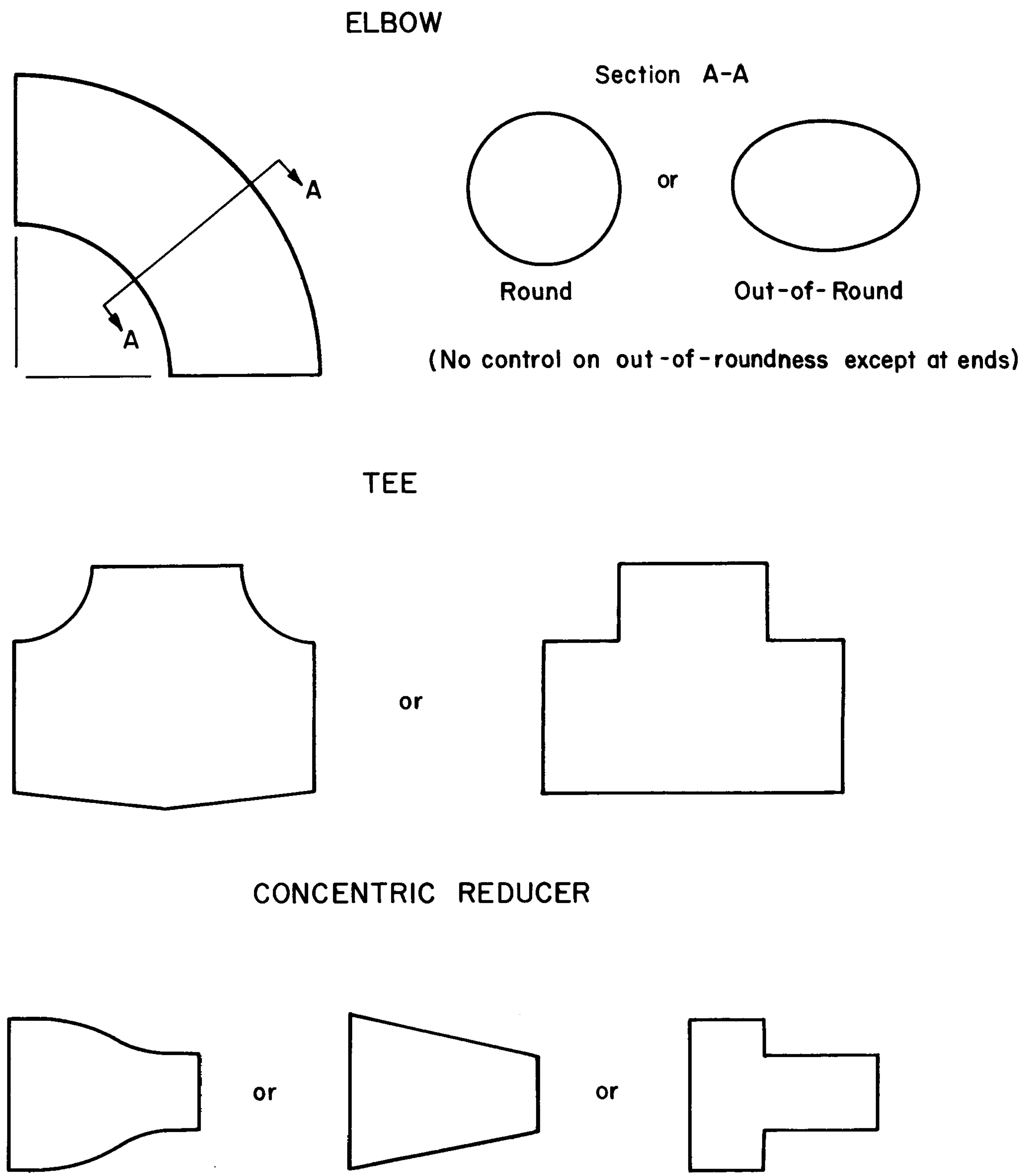


FIGURE 5.1: EXAMPLES OF SHAPE VARIATIONS PERMITTED BY USAS B16.9

elbow but he may actually receive a 12" XS elbow, tapered bored at the ends to meet end dimensions and tolerances. Here, from a structural analysis standpoint, the problem is that the piping system flexibility-analysis can significantly under-estimate loads developed by the piping system.

Standard USAS B16.9 was discussed as an example. Most other piping-component standards are similar. An exception occurs in flanged joints for which almost complete dimensional control is contained in the standards (USAS B16.5, MSS-SP44, API-600, etc.). Another exception, of course, is for straight pipe, in which straightness, out-of-roundness, minimum and maximum wall-thickness (the latter by a weight tolerance) are reasonably well controlled by pipe standards.

CHAPTER 6

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# 6. STRAIGHT PIPE AND WELDS THEREIN

Wrought steel* pipe, by definition and long established practice is generally furnished to nominal dimensions as given in USAS B36.10. This standard identifies certain classes of wall thickness as standard weight (Std.), extra strong (XS), and double extra strong (XXS) along with the more recent designations by "Schedule Number", i.e., Schedule 40, 80, and 160, respectively. Pipe can be differentiated from "tubing" in that pipe conforms to B36.10 dimensions while tubing has an outside diameter equal to the nominal size, with no generally accepted classes of wall thickness. From a structural analysis standpoint, there is no difference between pipe and tubing, except for certain tolerance considerations.

Pressure piping** is generally purchased to one of some 30 presently active ASTM pipe specifications. The ASTM specifications cover chemical and mechanical properties of the material, hydrostatic testing, tolerances, finish, marking and other similar requirements.

Pressure piping may be either "seamless" or "rolled-and-welded". Seamless pipe, as a standard product, is available in sizes up to 26" O.D. In larger sizes, or for heavy wall thicknesses in smaller sizes, seamless pipe can be obtained as "forged and bored" (ASTM-A369). Seamless pipe may also be centrifugually cast (ASTM-A426, -A451). Rolled-and-welded*** pipe is made, as the term implies, by rolling a plate to a cylinder and joining the edges with a longitudinal weld. While, in general, seamless pipe is preferred for critical-service piping, considerations of availability and cost may lead to the use of welded pipe; particularly for relatively thin-wall pipe.

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* Cast iron pipe is dimensionally described by USAS Standards of the A21. series.

** The term "pressure piping" is used herein to distinguish such pipe from that used for structural purposes.

***Spiral-welded pipe is available but is not generally used for critical-service pressure piping.

At the present time, pipe for the primary coolant loops of water cooled reactors is sometimes required in large sizes and heavy wall thicknesses such that it is beyond the range of normally furnished standard piping dimensions. These may be special products to the extent that they are machined, inside and outside, to final dimensions. At present, primary coolant loop piping is made either of solid stainless steel or of carbon steel with an internal stainless-steel cladding.

Ideally, pipe may be considered as a uniform-wall, cylindrical shell and, as such, is amenable to quite exact analysis in the elastic region and relatively exact analysis in the plastic or creep region. Because analysis is relatively exact for uniform wall cylindrical shells, there are many hundreds of published papers dealing with various aspects of cylindrical shells. Many of the problems of static, linear-elastic behavior of such shells have been solved. Present day papers are devoted more to the subjects of buckling, vibration, large plastic deformations, creep, anisotropic behavior, etc. This Chapter will not attempt to cover all this work, but rather touch on certain aspects of particular significance to piping.

Actual commercial pipe is normally not an idealized uniform-wall cylindrical shell. Commercial pipe is furnished to specified tolerancers which permit out-of-roundness and variable wall thickness. ASTM Specification A530, "General Requirements for Specialized Steel Pipe", gives tolerances applicable to most pressure-piping. As mentioned previously, large-size primary coolant loop piping is not necessarily a "standard" product and may be subject to special tolerances.

## 6.1 Internal Pressure Theory

### 6.11 Theory - Round, Uniform Wall Pipe

The round, uniform wall straight pipe may be considered as the basic piping component. Almost all codes and standards design equations are related to this component.

#### 6.111 Elastic Theory

The membrane stresses in a thin-wall pipe with closed ends are:

$$\sigma_h = \frac{d_m P}{2t} \tag{6.1}$$

$$\sigma_a = \frac{d_m P}{4t} \tag{6.2}$$

where $\sigma_h$ = stress in hoop direction

$\sigma_a$ = stress in axial direction

$P$ = internal pressure

$d_m$ = mean pipe diameter

$t$ = pipe wall thickness.

In these equations (whose origins are lost in antiquity), the pipe is assumed to be closed at the ends; the stresses are those in the pipe remote from the end closures.

For pipe without limitation to "thin-wall", the stresses in the elastic regimes are given exactly by Lame'(6.1):

$$\sigma_h = P\frac{\left[d^2 + 0.25(d\ D/R)^2\right]}{D^2 - d^2} \tag{6.3}$$

$$\sigma_a = P \frac{d^2}{D^2 - d^2} \tag{6.4}$$

$$\sigma_r = P \frac{[d^2 - 0.25(dD/R)^2]}{D^2 - d^2} \tag{6.5}$$

where the symbols used are defined in Figure 6.1.

An extensive review of formulas for pipe thickness, subject only to internal pressure loading, is given by Buxton and Burrows(6.2). In particular, the genesis of equations given in present Codes is discussed, i.e., equation (21) of Reference 6.2.

$$\frac{P}{S} = \frac{1}{0.5(D/T) - 0.4} \tag{6.6}$$

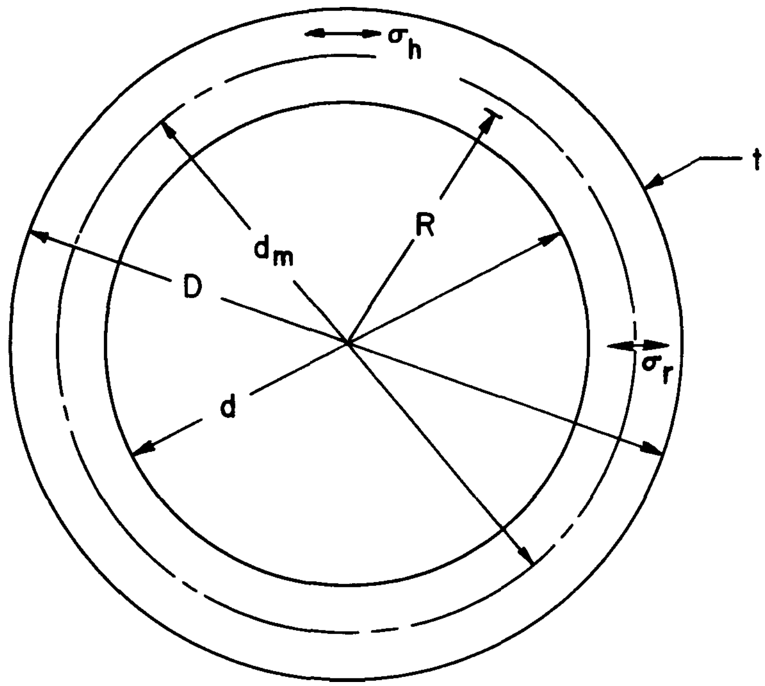
Equation (6.6) is a close approximation to Equation (6.3) for $R = d/2$. Equation (6.3) is the exact solution for the maximum elastic stress in the pipe; that stress occurs in the circumferential direction on the inside surface. Equation (6.6) was developed for Code use because it is simpler than Equation (6.3), although for $R = d/2$ Equation (6.3) reduces to:

$$\frac{P}{S} = \frac{D^2 - d^2}{D^2 + d^2} \tag{6.7}$$

6.112 Plastic Theory

<u>Yielding</u>

For thin wall pipes the pressure to produce yielding, according to the maximum shear failure theory, is simply




P = internal pressure
D = outside diameter
d = inside diameter
$d_m$ = mean diameter
t = wall thickness
R = variable radius

$\sigma_h$ = stress in hoop direction
$\sigma_a$ = stress in axial direction
$\sigma_r$ = stress in radial direction

FIGURE 6.1 NOMENCLATURE; ROUND, UNIFORM-WALL PIPE, ELASTIC STRESSES DUE TO INTERNAL PRESSURE

$$P_y = \frac{2\ S_y t}{d_m} \tag{6.8}$$

where $P_y$ = yield pressure

$S_y$ = yield strength of material (assumed to be isotropic)

For thin wall pipes, according to the octahedral shear stress failure theory, the pressure to produce yielding for a closed-end pipe is:

$$P_y = \frac{2\ S_y t}{d_m} \cdot \frac{2}{\sqrt{3}} \tag{6.9}$$

Marin and Sharma[6.3] extend the thin-wall theory to pipe material with anisotropic properties, using the octahedral shear stress failure theory. Rodabaugh, Atterbury, and McClure[6.4] also consider anisotropy in the yielding of gas transmission pipe line* under combined bending and pressure.

For thick wall pipe, the yield pressure must be more clearly defined. While the Lame′ equations, in combination with a selected failure criterion, will predict the pressure $P_{yi}$ that causes yielding on the inside of the pipe, the remainder of the pipe wall is still elastic. Accordingly, there is a redistribution of stresses as pressure is increased above $P_{yi}$. For piping, a more significant pressure is that which produced yielding "through-the-wall"; higher pressures then produce large (for perfectly

---

* Gas transmission pipe line can have significant anisotropic properties because of the cold expansion of the pipe in the manufacturing process.

plastic material, unbounded) increases in the pipe diameter. This aspect has lead to the development of the Nadai[6.5] equation:

$$P_y = S_y \cdot \frac{2}{\sqrt{3}} \ln \frac{D}{d} \qquad (6.10)$$

Equation (6.10) is for an ideally plastic material. For strain hardening materials, or for materials in the creep range, the Bailey-Nadai formula (see Reference 6.2) is:

$$\frac{P}{S} = \frac{n}{2}\left[1 - \frac{1}{\left[1 + \frac{2}{(D/T) - 2}\right]^{2/n}}\right] \qquad (6.11)$$

where T is the wall thickness and n is defined by the material-descriptive equation:

$$e_s = AS_s^{\,n} \qquad (6.12)$$

$e_s$ = major principal shear strain

$S_s$ = major principal shear stress

A,n = constants (material and temperature dependent).

It can be shown that for large values of D/T (thin wall pipe), Equation (6.11) reduces to P = 2ST/D.

Additional references on yielding of thick wall cylinders are listed herein, References (6.6) through (6.10). References (6.11) through (6.22) are primarily concerned with the maximum pressure capacity of thick-wall pipe, however, yielding is also considered.

## Maximum Pressure Capacity

From a theoretical aspect, the maximum pressure of a pipe may involve large deformations and strain hardening of the pipe material. The increase in pipe diameter causes increasing stresses per unit pressure; which is balanced by the strain hardening of the material. At some stage of increasing pressure the strain hardening is insufficient to compensate for the increasing diameter. This pressure represents the maximum the pipe can withstand.

Cooper(6.23) gives an equation for the maximum pressure of thin wall cylinders, which is:

$$P_u = (S_u t/r)\ [2/(3)^{\frac{n+1}{2}}] \qquad (6.13)$$

where $P_u$ = maximum pressure capacity

$S_u$ = material ultimate strength (nominal)

$t$ = initial wall thickness

$r$ = initial pipe radius (thin-wall theory)

$n$ = strain hardening exponent in the equation, $\sigma = Ke^n$.

For carbon steel such as A106 Gr. B, the value of n is about 0.2. For n = 0.2, Equation (6.13) gives a maximum pressure of 1.035 times $S_u t/r$. The theory indicates, therefore, that the maximum pressure for such pipe can be calculated with adequate accuracy (well within the tolerances on pipe wall thickness and ultimate strength) by a simple membrane stress equation. The theory, in this aspect, agrees with burst tests of thin-wall carbon steel pipe,

Marin and Sharma(6.3) extend the theory for thin wall pipe to include anistropy of the pipe material.

The maximum pressure capacity of thick-wall cylinders has been investigated by a number of authors, References (6.11) through (6.22). Marin and Rimrott[(6.18)] review much of this prior history and present a theory for the maximum pressure of thick-wall cylinders. Crossland[(6.20)] suggests that the material properties should be obtained from torsional tests whereas Marin[(6.21)] prefers to use the true stress-strain tensile relationship for the material.

Langer[(6.139)] reviews a number of the theories referenced above, shows comparisons with test data, and suggest appropriate design formulas for calculating the burst pressures of cylinders and spheres. These equations require knowledge of only two properties of the material; the nominal ultimate tensile strength and the uniform elongation.

## Creep and Creep-Rupture

The Code design of piping (and pressure vessels) for high-temperature operation takes creep into account by basing allowable stresses on the uniaxial tensile properties of the material at temperature. In the ASME Code*, the following creep or stress-to-rupture criteria are used for establishing allowable stresses.

| Section 1 | Section VIII | |
|---|---|---|
| | Ferrous | Nonferrous |
| Conservative average of stress to produce a creep rate of 0.01% per 1000 hr | Same as Sect. 1** | Same as Sect. 1 |
| 60% of average or 80% of minimum stress to produce rupture in 100,000 hr | 100% of estimated minimum stress to produce rupture in 100,000 hr*** | Same as for ferrous |

* 1968 editions

** Usually not greater than 60% of average stress-to-rupture in 100,000 hr

*** Only in some cases where successful service experience can be used as a guide

The allowable stress is then used to determine a minimum wall thickness by the equation*:

$$t = \frac{PR_o}{SE + yP} \tag{6.14}$$

where $t$ = minimum wall thickness

$P$ = internal pressure

$R_o$ = outside radius

$S$ = allowable stress

$E$ = joint efficiency (1.0 for seamless)

$y$ = a plastic flow coefficient, with values from 0.4 to 0.7 in ASME Section I; 0.4 is used in ASME Code Section VIII.

As discussed previously, Equation (6.14) with $y = 0.4$ is an approximation of the circumferential elastic stress on the inside surface of a thick wall cylinder. Accordingly, the Code design method implies that creep or stress-to-rupture is a function of the principal stress and that the elastic stress on the inside surface of a pipe may be compared with stresses in a tensile creep test in order to obtain a design formulation.

Before considering more sophisticated theories of creep in pipe, some general discussion of the problems involved is appropriate. Figure 6.2 shows a typical strain-time curve for a tensile creep test and serves to illustrate the three stages of creep. Figure 6.2 illustrates what is typically available with regard to creep characteristics of a material. It is developed by a test in which the material is subjected to a constant,

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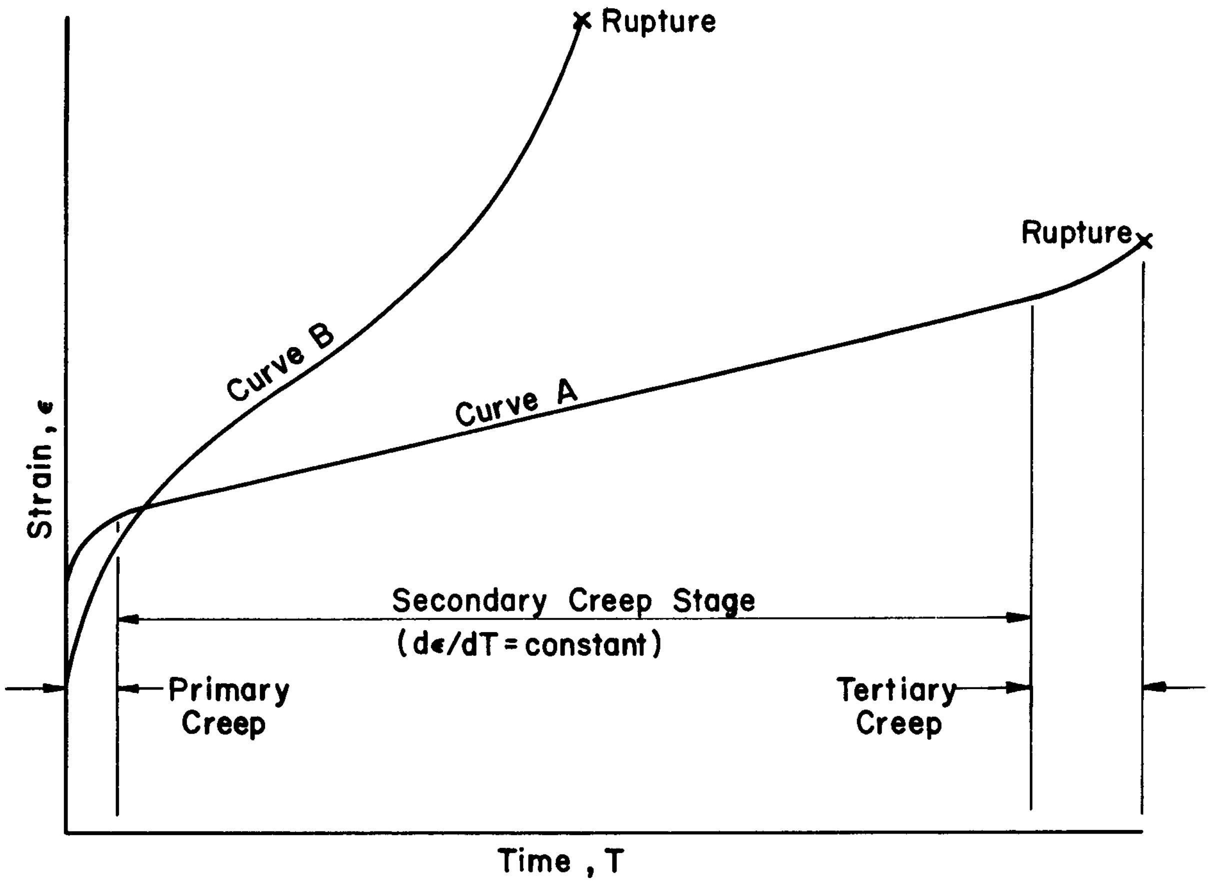
* In ASME Section VIII, the equation is limited to cylindrical shells in which $t < 0.5\ R_i$, where $R_i$ = inside radius.




CURVE A: Material, stress, temperature combination such that a well defined secondary creep stage occurs.

CURVE B: Material, stress, temperature combination such that stages of creep are not distinguishable.

FIGURE 6.2 ILLUSTRATION OF TYPES OF CREEP CURVES uniform (in the test section) stress at a constant temperature. The strain is measured as a function of time. As applied to design of a pipe, the following questions arise:

(1) What is the effect of a bi-axial or tri-axial stress field on the creep strain rate, and what effect on the total strain to produce fracture initiation?

(2) What is the effect of variable temperature; e.g., periodic shut-downs?

(3) What is the effect of variable loads?

These and similar questions have led to a significant amount of work in an attempt to establish reliable design methods under creep conditions. Finnie(6.24) gives a brief discussion of some aspects raised by the above questions and gives a list of 111 references in this field of work.

One important aspect for design under creep conditions is that the total strain at fracture or initiation of third stage creep can be relatively small under long-time creep conditions. At life-times of 10,000 to 100,000 hrs*, rupture at strains of around 1% may occur in some materials; those materials having quite large elongations (30-50%) in a short-time tensile test at temperature. The two criteria used in the ASME Code for establishing stresses at high temperatures implies that, in some cases, 100% of the rupture strength is less than 60% of the stress to produce 1% secondary creep in 100,000* hours; i.e., rupture occurs at

* Most data on creep or stress to rupture is extrapolated from shorter-time tests to a 100,000 hour estimate.

less than 1% secondary strain. Accordingly, in design of pipe for long service life at high temperatures, it is necessary to consider the magnitude of accumulated strains.

For thin wall pipe, the ASME Code design approach appears to be conservative because the principal stress is conservative (in this application) with respect to the octahedral shear stress theory for creep or stress-to-rupture.* Also, because creep strains are limited to 1% in 100,000 hours, the increase in diameter and decrease in thickness would only increase the stress about 2% as the result of finite deformations. A life of 100,000 hours corresponds to 11.4 years of continued operation. There are usually a number of conservative factors** involved in actual pressure vessel or piping design; e.g., the actual wall is usually thicker than the minimum required by Equation (6.14), the actual operating temperatures are usually less than the design temperature, and actual operating pressures are usually less than the design pressure. For creep considerations, even a small difference between actual operating temperature and design temperature can introduce significant conservatism in the design.

The Code design approach is presumed to be applicable to thick wall pipe (up to a wall of 1/2 the pipe radius). Some of the more sophisticated analysis methods for design of pipe under creep conditions will be discussed before commenting on this aspect.

The analysis of a thick wall tube under creep conditions is complicated by the presence of stress variations through the wall thickness.

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* Most references assume that creep, in analogy to plastic flow, is a function of the octahedral shear stress or maximum shear stress. However, in Reference (6.29), it is postulated that creep strain may be a function of principal stress and creep fracture may be a function of octahedral shear stress.

** Thermal gradients and external loads, particularly if cyclic, may reduce or eliminate the conservatism for pressure loading only.

Upon reaching temperature (usually assumed to occur in a short time, but without thermal gradients) the problem involves not only creep but relaxation; i.e., a conversion of stored elastic strains into plastic strains. Bailey[6.25], in 1935, presented the earliest known solution to the problem, neglecting elastic strains and first-stage creep. He also assumed that: (a) plane sections remain plane, (b) axial creep deformation is zero, (c) total strains are small with respect to the overall dimensions of the cylinder, and (d) the creep-rate/stress relationship is given by:

$$\dot{\epsilon}_i = A\,\sigma_i^{\,n}\, f(\sigma_s) \tag{6.15}$$

where

$\dot{\epsilon}_i$ = principal creep strain rate = $d\epsilon/dt$, t = time (i = 1,2,3)

A,n = material and temperature dependent constants, from tension-creep tests.

$\sigma_i$ = principal stress (i = 1,2,3).

$f(\sigma_s)$ = a function of the octahedral shear stress field.

The use of Equation (6.15) implies that the strain rate is independent of time; like the secondary creep section of Curve A shown in Figure 6.2.

The analysis by Bailey has been repeated with only minor variations by many authors.

The above analysis has been extended by Rimrott[6.27] to include the effect of finite deformations. Explicit equations and graphs for time-to-failure for both thin- and thick-wall cylinders are given by Rimrott, Mills, and Marin[6.28]. These analyses assume that the material can withstand large strains without fracture; they should be used with caution for pipe with material-temperature-time combinations for which strain at fracture may be small.

For material-temperature-time creep characteristics such as illustrated by curve B of Figure 6.2, Equation (6.15) is obviously inappropriate because there is no secondary creep stage. The following equation, proposed by Johnson[6.26], can be used.

$$\dot{\epsilon}_i = A_i \, \sigma_i^{\,n} \, f(\sigma_s) \; T^m \qquad (6.16)$$

Equation (6.16) is the same as Equation (6.15), with the addition of the time term, $T^m$. This equation can then be used to develop an equation analogous to those by Rimrott[6.27], this has been done by King and Mackie[6.29].

Pai[6.30] extends the theory further to consider anisotrophy, using a time-dependent creep relationship.

As compared to the more sophisticated analyses cited above, the ASME Code Section VIII design approach is conservative for thick-walled cylinders because it assumes that the elastic stress at the inside of the cylinder is not reduced by plastic flow due to creep. ASME Code Section I, by using a y-factor in Equation (6.14) which is greater than 0.4, removes part of this conservatism. The ASME design approach may be unconservative if, in fact, creep strain and stress-to-rupture are functions of the maximum shear stress or the octahedral shear stress. At the upper limit of thickness permitted in Section VIII ($t = 0.5R_i$), the stress per unit pressure happens to check exactly with the Lame′ equation for inside hoop stress, i.e., $\sigma/P = 2.60$. The effective shear stress* at the bore is 2.60P + P = 3.60 P. This is 1.38 times the principal stress. The effective octahedral shear stress* at the bore is 3.105 P, which is 1.19 times the principal

* Effective stresses are defined by Equations (3.1) and (3.2) of Chapter 3.

stress. However, creep-relaxation will tend to even out the hoop stresses across the wall thickness; they will tend to reach an asymptotic value of 2.0P. At this stage, the effective shear stress will be 3.0P at the bore. The effective octahedral shear stress will be 2.61P at the bore. Accordingly, the ASME Code design approach appears to be conservative, when considered in conjunction with a limiting creep strain of 1% which limits finite deformations to negligible amounts.

## 6.12 Theory, Out-of-Round Pipe

An analysis of stresses in out-of-round pipe was published by Haigh[6.31] in 1936. A similar analysis is given by Schmidt[6.32]. In these theories, the initial cross-sectional shape is assumed to be described by the equation:

$$R = R_m + \sum U_n \cos n\theta \qquad (6.17)$$

where $R$ = initial pipe radius

$R_m$ = mean pipe radius.

Equation (6.17) can, of course, be used to describe any cross-sectional shape of the pipe. It is assumed that this shape persists for a long distance along the pipe axis. If the out-of-roundness is small ($U_n/R \ll 1$), the membrane stress is still essentially PR/t. The bending stress is given by:

$$\sigma_b = \frac{PR}{t} \sum \left(\frac{U_n}{t}\right) \left[\frac{6}{1 + \frac{12PR^3\,(1 - \nu^2)}{Et^3(n^2 - 1)}}\right] \cos n\theta \qquad (6.18)$$

For a simple ovality of the cross section, n is equal to 2, the maximum bending stress (at $\theta = 0, \pm \pi/2$ and $\pi$) as given by Equation (6.18) is essentially the same as that given in the ASME Code[6.33].

It can be seen that Equation (6.18) is non-linear with pressure. Figure 6.3 shows an example of this non-linear effect for the particular case of $n = 2$, $R/t = 40$, $U_2/R = 0.01$. Figure 6.4 shows how the total stress (bending plus membrane) compares with the membrane stress for a range of values of R/t and for $U_2/R = .01$ or $U_2/R = .005$. It might be remarked that the non-linear effect is quite significant for large values of R/t. For example, at $R/t = 60$, $U_2/R = .01$, a linear analysis would give $\sigma_{max}/(PR/t) = 4.6$, as compared to the non-linear analysis result of $\sigma_{max}/(PR/t) = 1.37$ for P such that $PR/t = 20{,}000$ psi.

Pipe made from a rolled-and-welded plate may have a local out-of-roundness at the longitudinal weld because of either under-rolling or over-rolling the abutting plate edges. The description of this shape by Equation (6.17) involves higher values of n. It can be seen in Equation (6.18) that as n increases, the non-linear effect decreases. At this time, no calculations for local out-of-roundness using Equations (6.17) and (6.18) are available; however, a theory and sample calculation given by Schmidt[6.34] are pertinent. Schmidt develops the linear theory for a deformed shape as shown in Figure 6.5. In a specific example in which $R/t = 45.5$, and u is such that the out-of-roundness is 2% of the diameter, Schmidt reports that the maximum bending stress is 11.1 times the nominal membrane stress, PR/t. Such high bending stresses are presumably relieved by yielding, however, cyclic pressure tests on pipe with longitudinal welds indicates

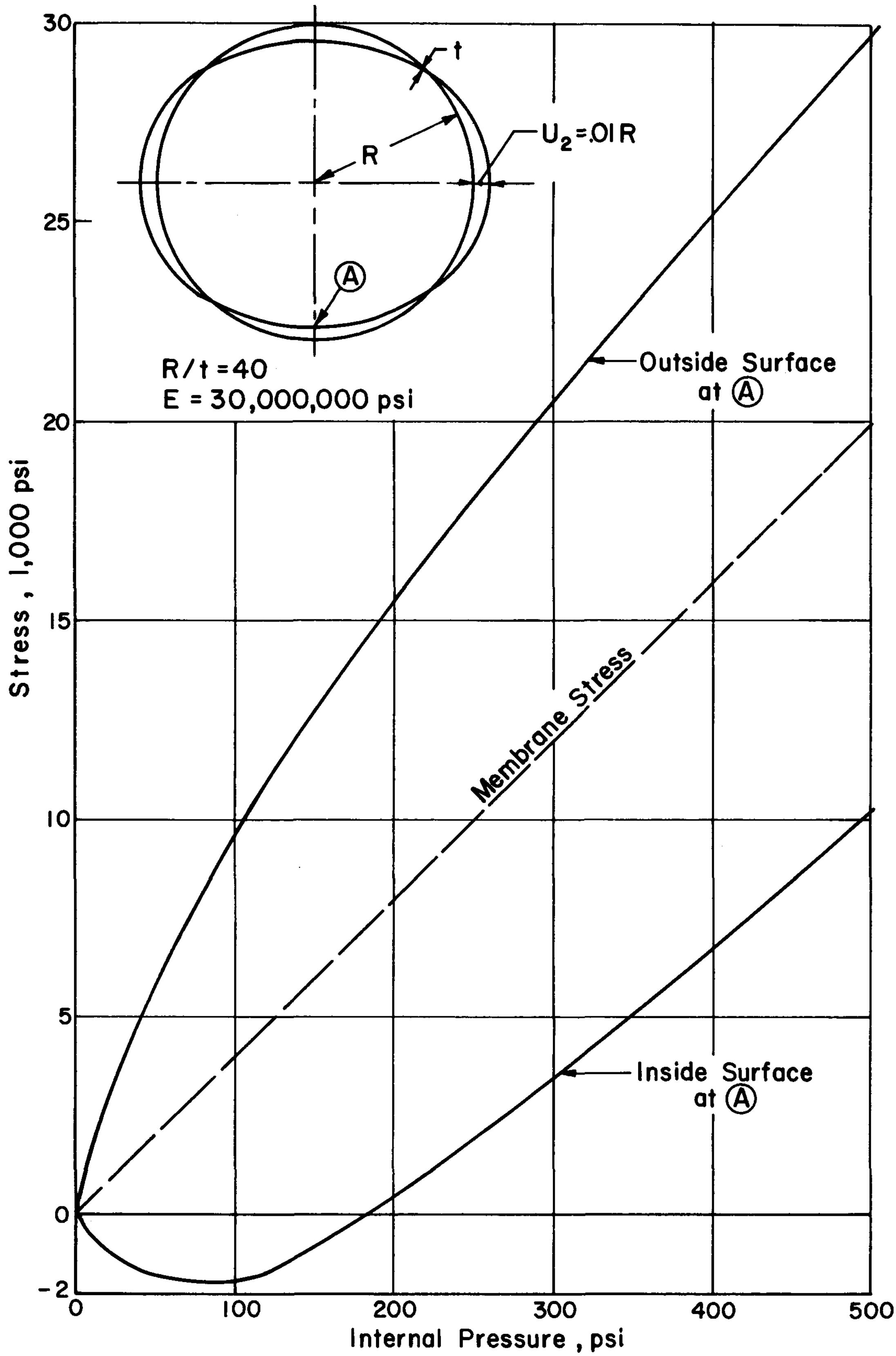



FIGURE 6.3 EXAMPLE OF PRESSURE VS. STRESS FOR AN OUT-OF-ROUND PIPE, n = 2

$P = S_a t/R$, $S_a$ = 20,000 psi

R = mean pipe radius

t = pipe wall thickness

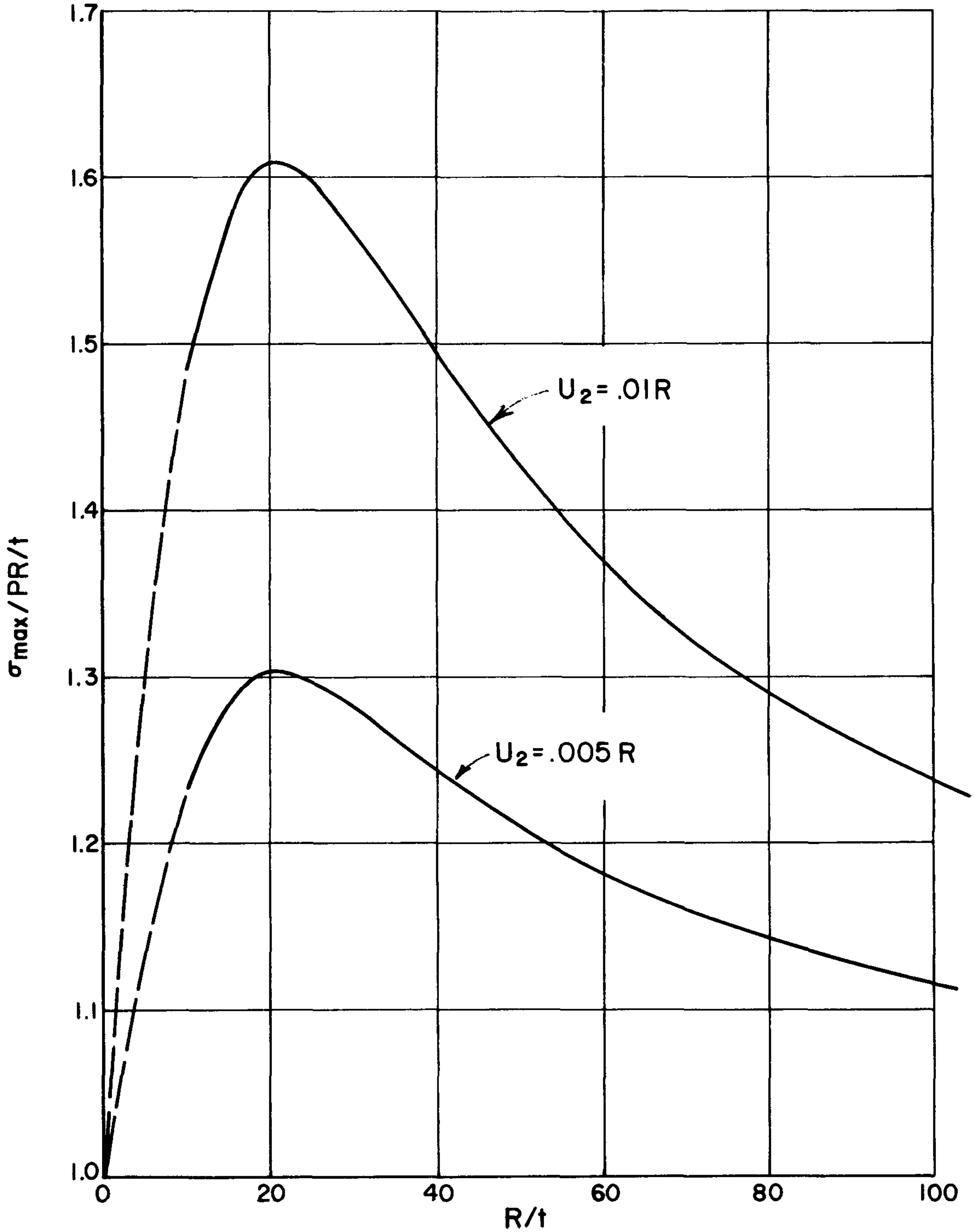
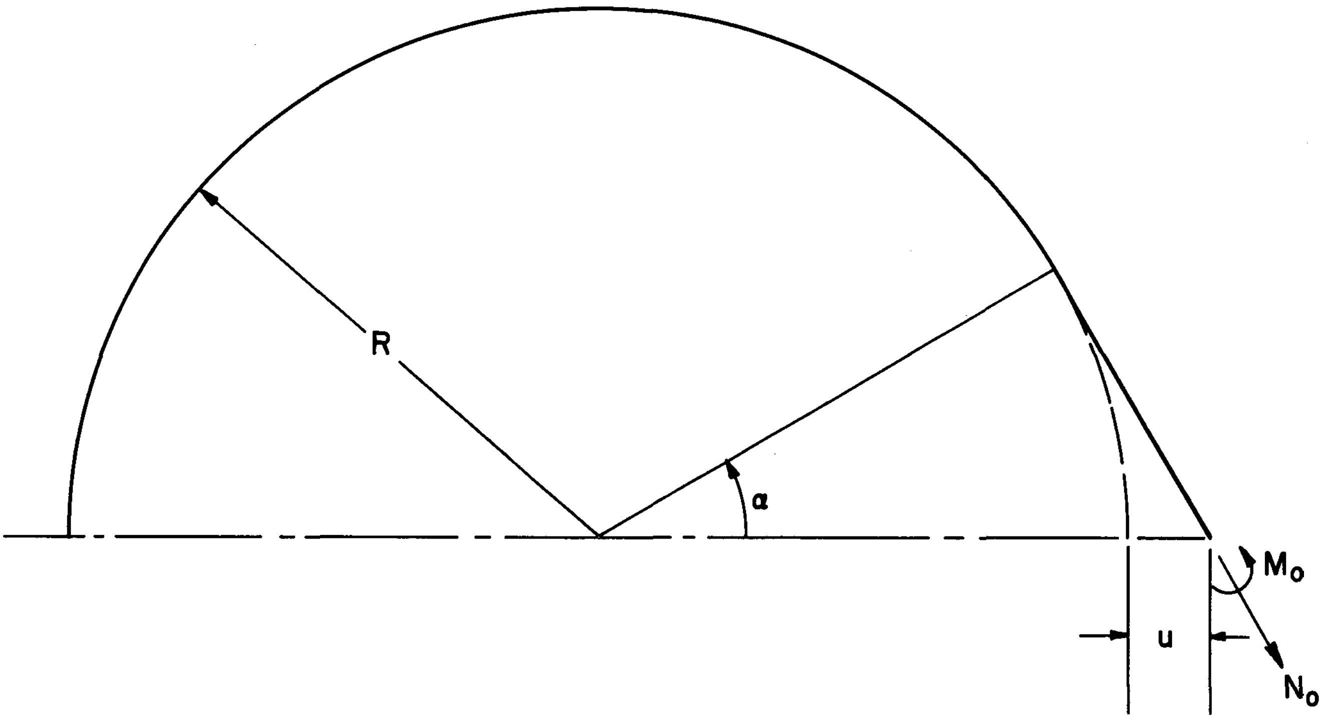



FIGURE 6.4 MAXIMUM STRESS AS A FUNCTION OF R/t, OUT-OF-ROUND PIPE, n = 2




$$u = 2(\sec\alpha - 1)/(\sec\alpha + 3)$$

$$M_o = PR^2\, G(\alpha) \qquad N_o = PR\, H(\alpha)$$

$$G(\alpha) = [3(\pi - \alpha)^2 \tan^2\alpha + (\pi - \alpha)\tan^3\alpha\,(3 + \sin^2\alpha + 0.5\tan^2\alpha) + \tan^6\alpha\,(1 + 29\cos^2\alpha)/12]/I(\alpha)$$

$$H(\alpha) = [6(\pi - \alpha)^2 \sec^2\alpha + (\pi - \alpha)\sin\alpha\,(12 + 16\tan^2\alpha + 2.5\tan^4\alpha) + \tan\alpha\,\sin\alpha\,(6 + 10\tan^2\alpha + 0.5\tan^4\alpha)]/I(\alpha)$$

$$I(\alpha) = 6(\pi - \alpha)^2 + (\pi - \alpha)\sin 2\alpha\,(6 + 9\tan^2\alpha + 2\tan^4\alpha) + \sin^2\alpha\,(6 + 12\tan^2\alpha + \tan^4\alpha)$$

FIGURE 6.5 CROSS SECTION OF PIPE WITH AN ANGULAR BUCKLE, ANALYSIS BY SCHMIDT(6.34) that the combination of out-of-roundness and weld irregularities may constitute a significant weakness in pipes subjected to cyclic internal pressure.

The theoretical effect of out-of-roundness on maximum pressure capacity or creep is not known. For a pipe made of a material with a reasonable amount of ductility, and with initial out-of-roundness not greater than about 2%, it would seem unlikely that the out-of-roundness would have any practical significance for these aspects.

## 6.13 Wall Thickness Variations

Calculations of stresses in pipe are normally based on the minimum specified wall thickness. For most standard pipe, the average wall thickness is about 12% thicker than the minimum specified. Seamless pipe quite often has a zone of minimum thickness which spirals through the pipe length. The wall thickness variation in pipe is normally quite gradual and does not significantly increase stresses. Accordingly, stresses calculated on the basis of minimum wall thickness are conservative. In-so-far as the writer is aware, no one has attempted to make any corrections to calculated stresses in straight pipe due to variations in wall thickness.

## 6.2 Moment Loading, Theory

Pipe is often subjected to significant moment loadings which arise due to thermal expansion of the pipe, movement of end anchors or weight of the pipe and its contents. The term "moment loading" used herein is intended to describe either bending or torsion of the pipe as a beam.

Axial loads and shear loads may also be applied, but these are usually negligible in a piping system.

## 6.21 Elastic Theory

For moment loads which do not produce large deformations, and for which elastic buckling does not occur, the load stress and load-displacement relationships are given by the usual beam equations.

$$S_a = (Mr/I) \cos \theta \tag{6.19}$$

$$\tau = Tr/J \tag{6.20}$$

$$\theta = M\ell/EI \tag{6.21}$$

$$\delta = M\ell^2/2EI \tag{6.22}$$

$$\gamma = T\ell/GJ \tag{6.23}$$

where

$S_a$ = axial stress (hoop stress = 0)

$\tau$ = shear stress

E = modulus of elasticity

G = shear modulus = $E/2(1 + \nu)$

I = moment of inertia = $(\pi/64)(D^2 - d^2)$

J = polar moment of inertia = 2I

M, T, r, $\ell$, $\theta$, $\delta$, and $\gamma$ are defined in Figure 6.6

Small out-of-roundness of the cross section, which can have a large effect on stresses due to internal pressure, enters into stresses due to moments only to the extent that it changes the value of I, J, and r by a small amount.

Elastic buckling of pipe due to moments is not ordinarily a problem. Some aspects are discussed under Section 6.4 herein.

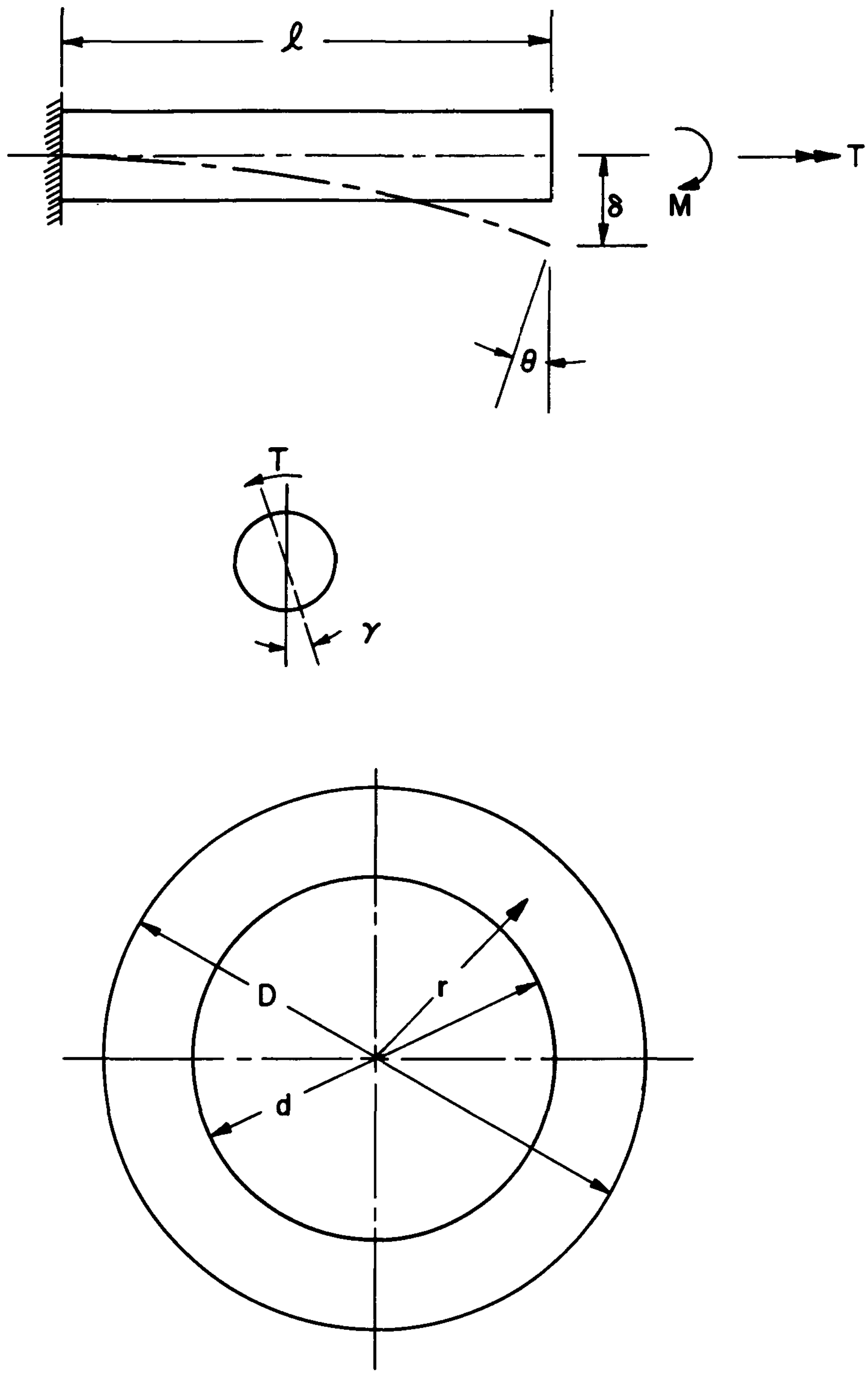



FIGURE 6.6 NOMENCLATURE, MOMENT LOADS ON PIPE

## 6.22 Plastic Theory

The theory for beams subjected to a pure moment loading is discussed in several books(6.35, 6.36, 6.37). These theories involve the assumptions that:

(a) Material has the same stress-strain relationship in tension as in compression.

(b) Plane sections remain plane.

(c) By symmetry (for beams as shown in Figure 6.7), the neutral axis is the plane of symmetry axis.

For either an elastic-perfectly plastic or a rigid-perfect plastic idealization of the material properties, a limit moment can be established. The limit moment $M_1$ is given by the general equation:

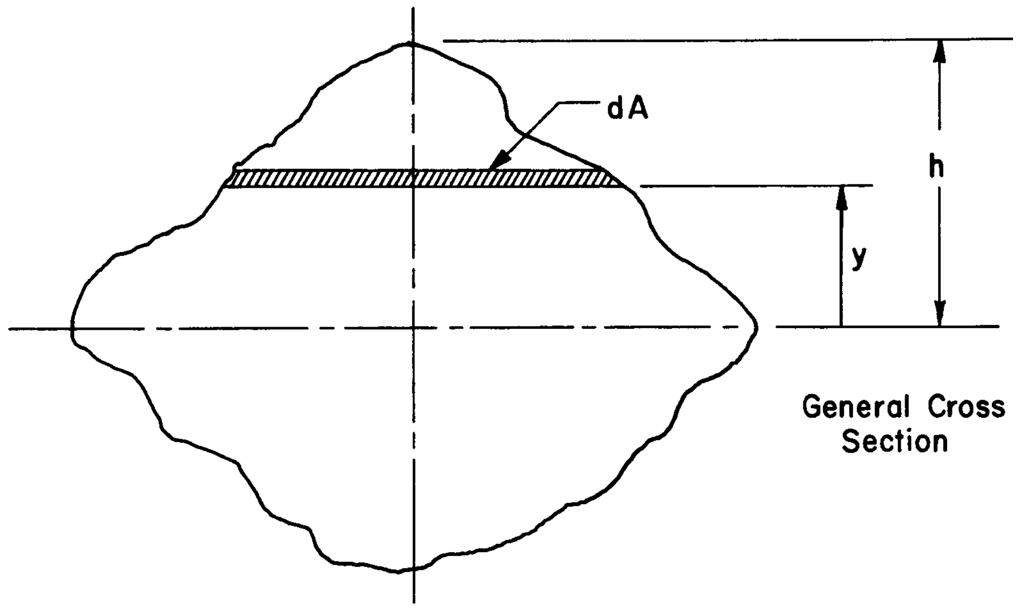
$$M_1 = 2\int_o^h S_o \, y \, dA \qquad (6.24)$$

where $S_o$ = yield stress

$y$, $dA$ defined in Figure 6.7.

As applied to a beam of rectangular cross section with width b and depth 2h; $dA = bdy$ and $M_1 = S_o \, b \, h^2$. The value of $M_1$ is 1.5 times the moment which produces outer fibre yield stress based on elastic theory. For a thin-wall cylinder, $y = r \sin \varphi$ (see Figure 6.7), $dA = tr \, d\varphi$, and equation (6.24) gives $M_1 = 4S_o \, r^2 t$. In this case $M_1$ is equal to $4/\pi$ times the moment that produces outer fibre yield stress based on elastic theory.

Most piping materials strain harden to some extent. The stress-strain relationship can often be satisfactorily approximated by the equation:



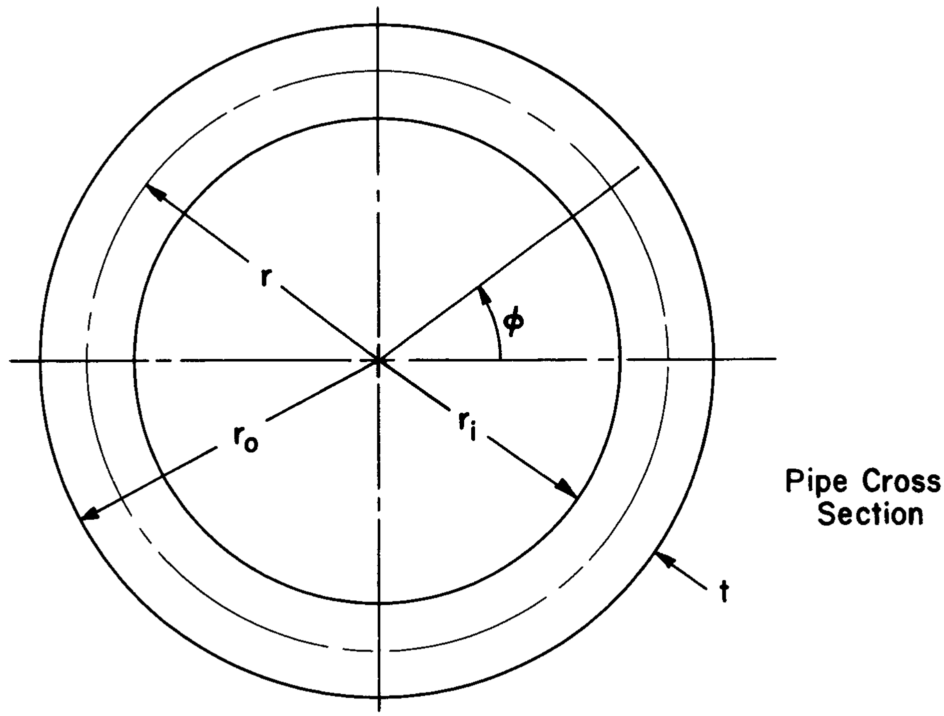




FIGURE 6.7 NOMENCLATURE, PLASTIC BENDING OF PIPE

$$S = C\, e^{n} \tag{6.25}$$

S = stress

e = strain

C and n are material-dependent constants

The general moment-strain relationship is given by:

$$M = 2\int_{o}^{h} Ce^{n}\, y\, dA \tag{6.26}$$

With the assumption that plane sections remain plane, $e = y/\rho$ where $\rho$ = radius of curvature of bent beam. The general moment-curvature relationship is then:

$$M = 2\int_{o}^{h} \frac{Cy^{n+1}}{\rho^{n}}\, dA \tag{6.27}$$

As applied to a beam with rectangular cross section with width b and depth 2h, dA = bdy and the moment-curvature relationship from equation (6.27) is

$$M = \frac{2C\, b(h)^{n+2}}{(n+2)\, \rho^{n}} \tag{6.28}$$

As applied to a thin-wall cylinder, $y = r \sin \varphi$, $dA = tr\, d\varphi$ and equation (6.27) gives, according to Rodabaugh, et al[(6.4)],:

$$M = \frac{2\sqrt{\pi}\, C(r)^{n+2} t}{\rho^{n}} \left[ \frac{\Gamma\left(\frac{n}{2} + 1\right)}{\Gamma\left(\frac{n+3}{2}\right)} \right] \tag{6.29}$$

where $\Gamma$ = gamma function

Equation (6.29) includes the elastic and limit-moment solutions as special cases. For the elastic range, C = E (modulus of elasticity) and n = 1; $\Gamma\,(3/2) = \sqrt{\pi}/2$ and $\Gamma\,(2) = 1.0$ from which $M = E\,(\pi r^3 t)/\rho$. For the rigid-perfectly plastic material, $C = S_o$ (yield stress), n = 0, $\Gamma\,(1.0) = 1.0$, and $M_1 = 4\, S_o r^2 t$.

For a thick-wall cylinder, the moment-curvature relationship is given by:

$$M = \frac{2\sqrt{\pi}}{(n+3)} \frac{C}{\rho^n} \left[\frac{\Gamma\left(\frac{n}{2}+1\right)}{\Gamma\left(\frac{n+3}{2}\right)}\right] \left(r_o^{\,n+3} - r_i^{\,n+3}\right) \tag{6.30}$$

where $r_o$ = outside radius of cylinder

$r_i$ = inside radius of cylinder.

Equation (6.30) also contains the elastic and limit-moment solutions as special cases.

Equations (6.29) and (6.30), for $n > 0$, predict a monotonically increasing moment as the radius of curvature decreases. The limit load would then correspond to the ultimate tensile strength of the material. An implied assumption is that the cross section remains circular. Ades[(6.38)] considers the effect of flattening of the cross section and shows, by means of a minimized work approach, that there is a value of $\rho$ (radius of curvature) which corresponds to a maximum value of applied moment, M. This value of M is then a "limit moment" considering strain hardening. The method presented by Ades involves numerical integration; hence, the results cannot be expressed in a simple closed-form expression. Further work with this approach is required to permit comparisons with Equation (6.24), (6.29), and (6.30).

For very thin cylindrical shells, a further limitation due to elastic buckling exists. This aspect is discussed by Timoshenko and Gere[(6.39)]. For steel pipe with yield strength of 35,000 psi, this type of local buckling is controlling for D/t ratios of around 200 and larger; hence, this is not usually a problem in typical piping. Elastic-plastic buckling may intervene at some smaller of D/t; however, presumably the flattening type of deformation discussed by Ades[6.38] will predominate in typical piping.

For bending loads applied to pipe under secondary stage creep conditions, a solution parallel to that for plastic bending may be obtained*. The assumption is made that:

$$S = C(\dot{e})^n \tag{6.31}$$

where S = stress

$\dot{e}$ = strain rate = de/dT, T = time

C, n = material, temperature dependent constants.

This then leads to Equation (6.30), except that $\dot{\rho}$, the time rate of change of curvature, is substituted for ρ. For a constant applied moment, the radius of curvature decreases inversely with time. The radius of curvature at time T is given by:

$$\rho = \left\{\frac{2\sqrt{\pi}C}{(n+3)M}\left[\frac{\Gamma\left(\frac{n}{2}+1\right)}{\Gamma\left(\frac{n+3}{2}\right)}\right]\left(r_o^{\,n+3} - r_i^{\,n+3}\right)\right\}^{\frac{1}{n}}\frac{1}{T} \tag{6.32}$$

Equation (6.32) may also be applied to the relaxation case in which a moment is rapidly applied a time T = 0. This moment induces a radius of curvature which can be calculated on an elastic basis. Assuming this radius of curvature remains fixed, the decrease in the value of M as a function of time can be calculated.

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* A somewhat different formulation of the solution to this problem is given by Robinson[6.40]. The equivalence can be shown by noting that Robinson's n is equivalent to 1/n herein, and that Robinson's $r_o^{\,n} = (S/C)(\dot{\rho})^n$ herein.

The preceding analysis for creep-bending assumes that the cross section remains circular. Presumably, as in the case of plastic bending, the cross section will tend to flatten for relatively small values of $\rho$. A theoretical development including this effect is not known to the writer.

## 6.3 Combined Pressure and Moment Loading

In the elastic regime, for round, uniform-wall pipe, stresses and/or deflections can be obtained by linear superposition of the individual loads. The stresses in out-of-round pipe are non-linear with pressure. For large deformations due to bending moments, some flattening of the pipe cross section may occur, leading to a non-linear interaction between pressure and bending. Further study of this aspect is required; however, for most piping systems this will probably have a negligibly small effect.

In the plastic regime, the paper by Stokey, Peterson, and Wunder[6.41] gives limit load combinations. The loadings consist of internal pressure, axial force, bending moment and torque. The analysis is based on the maximum shear stress yield criterion with the material assumed to be rigid-perfectly plastic. Figure 6.8 illustrates the combinations of moment and torque which can be applied for a specific case of internal pressure that produces a hoop stress (PD/2t) equal to two-thirds of the material yield strength. It can be seen that the pressure does not reduce the limit load capacity very much. For example, for a torque, T = 0, M is 85% of the limit moment in the absence of internal pressure. Similarly, for M = 0, T is 95% of the limit torque in the absence of internal pressure.

Theories for creep under combined pressure with bending, torsion or axial load have been developed by several authors under the general subject of creep in combined stress fields; e.g., Nadai[6.35] and Johnson[6.42]. Finnie[6.43] has investigated the particular case of pressure combined with a bending moment.

## 6.4 Elastic or Plastic Instability

Piping is sometimes subjected to lateral external pressure and must be designed to support such pressure. The design of cylindrical shells

$P = St/r$, $S = 0.667\ S_o$

$S_o$ = yield strength of pipe material

r = pipe radius

t = pipe wall thickness

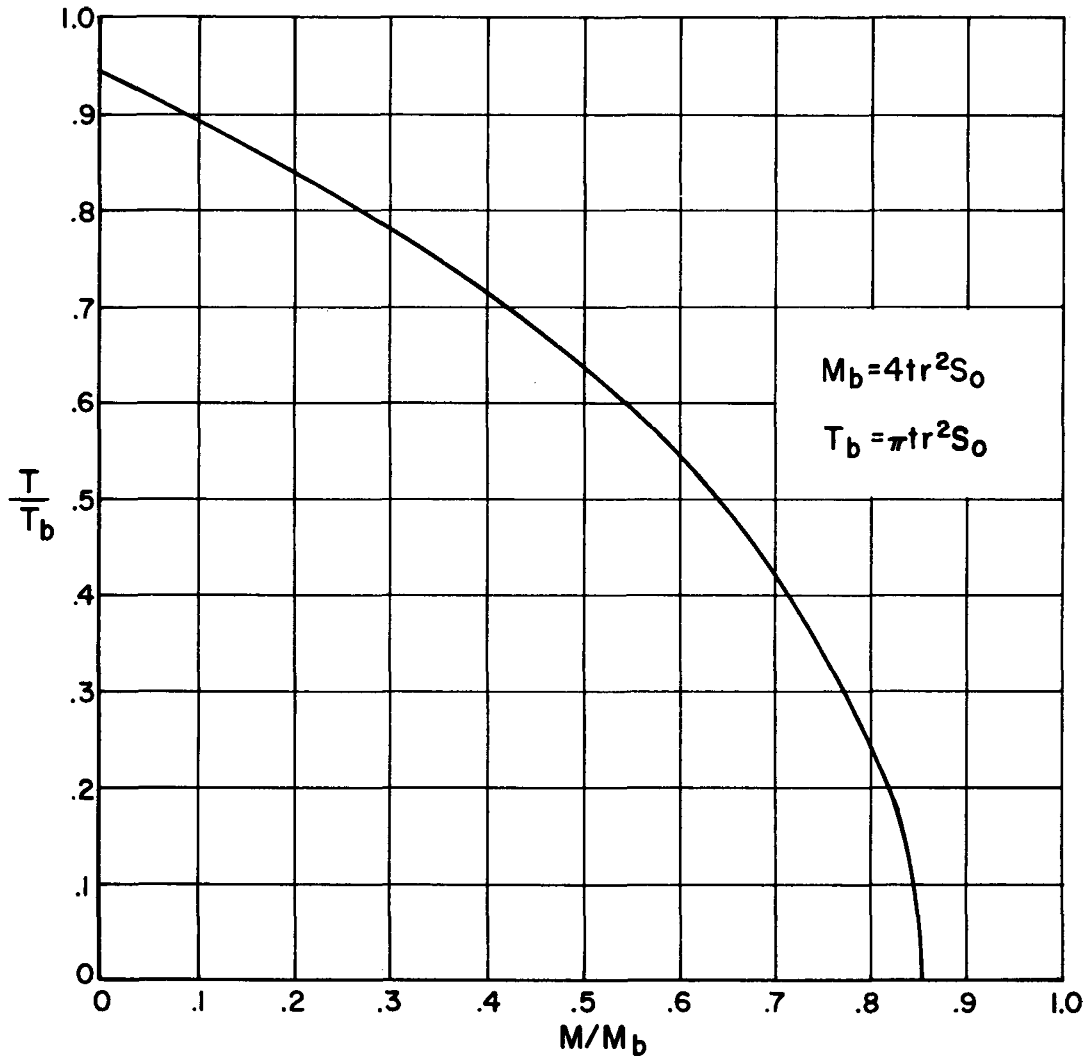


FIGURE 6.8 LIMIT LOAD COMBINATIONS ON THIN-WALL PIPE

for external pressure loading is covered in the ASME Boiler Code. The background of these design methods are given in References (6.44) through (6.49).

Internal pressure loading can lead to instability of a pipe as a beam-column. This kind of instability arises in piping systems in which the axial pressure load is restrained by some structure other than the pipe itself. The most common examples occur in piping systems using either bellows expansion joints or packed slip-joints. Haringx[6.50] presents the theory for this type of instability; the theory is essentially that of a column loaded in axial compression by the pressure-end-force, $\pi r^2$ P. In installing such piping systems, care must be taken that sufficient guides (not hangers) are placed along the pipe length.

As discussed in the last part of sub-section 6.22, for very thin pipe the application of a bending moment can produce local buckling. An analogous kind of buckling can occur for axial compressive loads or a torsional moment. The existing theory and test data suggest that these kinds of buckling are controlling only for larger D/t ratios than normally encountered in piping.

United Nuclear Corporation has made a study of stability analysis of piping; see Reference (6.140). Much of the material in this reference is restricted in application to relatively thin-wall piping for low pressure, high temperature service, but the report is a fairly comprehensive study of stability analysis methods. Also, the NASA Shell Analysis Manual[6.141] contains much information on buckling of cylindrical shells subjected to pressure, moments, and torsion. Interaction curves are also presented for combined loading. Most of the NASA manual pertains to D/T ratios greater than 100. Some information on creep-buckling is contained in Reference (6.140).

## 6.5 Test Data

### 6.51 Elastic Stresses

Elastic stresses in a round, uniform-wall cylindrical shell are quite firmly established on the basis of theory. Experimental verification of such stresses would be somewhat academic. Presumably, for this reason, the literature does not contain data of this type except as by-products of other tests. For example, Leven[6.51] ran photoelastic tests on cylindrical shells with nozzles subjected to internal pressure loading. Where test data are given at points remote from the nozzles, the reported stresses agree adequately with Lame equations for stresses in a cylinder. Leven[6.52] ran photoelastic tests on cylindrical shell nozzles in spherical shells in which a bending moment was applied to the cylindrical nozzle; the stresses show reasonable agreement with the usual equation $S = (Mc/I) \cos \theta$. Accordingly, while there aren't many test data, there is no reason to doubt the validity of the usual equations for calculating stresses in the elastic region.

Measured stresses due to internal pressure in pipe (which is not necessarily either round nor of uniform wall thickness) are quite often found to be quite different than predicted by Lame' equations, for the average diameters and wall thickness of the test pipe. For example, tests[6.53] of 8.625" O.D. x 0.219" wall pipe, in which strain gages were placed on the outside surface at 6 locations around the pipe circumference at 4 planes along the pipe axis (24 gages) gave hoop stresses which ranged from 0.78 to 1.09 times the theoretical (Lame') hoop stress. Similar tests on 12.75" O.D. x 0.25" wall pipe and 24" O.D. x 0.250" wall pipe gave hoop stresses of 0.67 to 1.05 (12" pipe) and 0.97 to 1.33 (24" pipe) times the theoretical hoop stress. Similar deviations between measured stresses and round-cylinder theory are reported by Kilpi[6.54] in tests on a large cellulose digester pressure vessel.

There are a few other isolated pieces of test data on the deviation of measured stress in cylindrical shells from theoretical values of round cylinders. Presumably, these deviations are due to out-of-roundness of the vessel; however, accurate quantitative data are meager. Kilpi[6.54], in order to obtain a quantitatively controlled test of the effect of out-of-roundness, performed tests on a ring with a local, shallow inward "buckle". The ring was loaded with simulated internal pressure. According to Kilpi, the stresses determined by this test agree quite well with Equation (6.18).

### 6.52 Yield Loads

The thin-wall tube has been extensively used as a test specimen to investigate yield and plastic flow criteria. Many of these tests indicate that yielding starts somewhere between the maximum shear stress failure criterion and the octahedral shear stress criterion; on the average the test results agree better with the latter criterion[6.35, 6.55]. This implies that a thin-wall cylindrical shell with closed ends will yield at a pressure about 15% higher than the pressure required to cause the hoop stress to equal the yield strength of the material. In many cases, effects of anisotropy, along with vagueness in the definition of yield strength of a material and yield pressure of the cylinder, are sufficient to introduce uncertainties in the results which are greater than the difference between the maximum shear theory of yielding and the octahedral shear stress theory of yielding.

No test data have been found which indicate the effect of out-of-roundness on yielding with internal pressure or other types of loading.

One type of experimental data, of significance in piping design, would give limit bending loads with various magnitudes of internal pressure.

Ades[6.38] implied that he had such data for zero internal pressure, but test results were not given. Otherwise, no experimental data of this type are known.

## 6.53 Maximum Pressure Capacity

The maximum pressure capacity of cylindrical shells has been a matter of practical significance for many years; considerable experimental data exist in the literature. The earliest known tests were published by Cook and Robertson[6.56] in 1911. Additional data are given in References (6.57) through (6.69) and in (6.139). These tests cover a wide range of O.D./I.D. ratios; from 1.07 to 12. These tests were used, in part, to evaluate the accuracy of theoretical methods for calculating the "instability pressure" of thick-wall cylinders. A practical observation, noted by several of the authors and discussors, is that the test data* correspond about as well with the mean diameter formula as with any of the theoretical equations. The mean diameter formula is simply:

$$P_u = 2\ S_u\ t/D_m \tag{6.33}$$

where

$P_u$ = ultimate pressure capacity

$S_u$ = nominal tensile strength of the material

$t$ = wall thickness

$D_m$ = mean (average of inside and outside) diameter

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* While the test data covers a wide variety of materials, they do not cover "brittle" materials. For such materials, particularly in thick-wall cylinders, Equation (6.33) may be unconservative.

With one exception, all of the data in References (6.56) through (6.69) are on seamless cylindrical shells. Maximum-pressure-capacity test data on cylinders with longitudinal welds are apparently quite meager. Griffis, et.al.(6.57) include data on 6 test specimens with simulated longitudinal welds; two of which were tested as closed-end cylinders. The maximum pressures of these two cylinders were essentially the same as the seamless cylinders. The "weld" was simulated by machining longitudinal slots, 180° apart, along the entire length of the solid bar stock and filling those slots with weld metal. Machining and finishing of the tubes proceeded so that the soundest part of the weld thickness was within the final wall thickness.

No quantitative data on the effect of out-of-roundness on maximum pressure capacity of pipe are available. There are some data on burst tests of pipe and additional data on burst tests of piping components attached to straight pipe during the test. Presumably many of these pipes were typically out-of-round. There is no evidence that such out-of-roundness has any significant effect on the burst pressure of pipe made of a reasonably ductile material.

## 6.54 Creep and Creep Rupture

Tubular specimens have been used by several investigators; in part to investigate the relationship between creep strain in combined stress fields. Some of these are listed as reference (6.42) and (6.70) through (6.72).

Of more direct interest, in the present context, are several investigations of creep in closed-end pipe subjected to internal pressure.

Earliest known tests of this type were reported by Clark[6.73], Clark and White[6.74], Van Duzer and McCutchan[6.75] and Norton[6.76]. Later tests on stress-to-rupture are given by Kooistra, Blaser and Tucker[6.77], Tucker, Coulter and Kooistra[6.78], King and Mackie[6.29], Lee[6.142], and Davis[6.143].

Several pertinent papers describing service experience with piping operating at high temperatures are listed in references (6.79) through (6.82). These service experiences indicate, as was pointed out before herein, that the limited strain capacity of metals at high temperatures and long life is a significant aspect of designing piping systems for high temperature operation.

## 6.55 Fatigue

Available fatigue test data is almost entirely limited to:

(1) Tests run at room temperature

(2) Tests in environments such as air, water, or oil. Corrosive effects are presumably small.

(3) Fatigue failure is defined as a crack which has propagated through the wall thickness.

Unless specifically stated otherwise, these conditions apply to all of the fatigue data discussed below.

### 6.551 Cyclic Internal Pressure

The thin cylindrical shell has been used as a test specimen to determine the effect of combined stresses and mean stresses on fatigue life of materials. Marin[6.83], Morikawa and Griffis[6.84], Majors, Mills, and MacGregor[6.85] and Bundy and Marin[6.86] give results of tests in which cylindrical tubes were subjected to combinations of internal pressure and axial loads. The interpretation of these results with respect to combined-stress-fatigue-failure theory is obscured by the presence of anisotropy in some, if not all, of the test specimen materials. With respect to pipe, the results indicate that for a ratio of $\sigma_h/\sigma_a$ of 2 (closed-end pipe), the value of $\sigma_h$ to produce fatigue in less than 100,000 cycles is greater than the yield strength and the value of $\sigma_h$ corresponding to the endurance limit (large number of cycles) is about equal to the yield strength. This observation applies to tubes made of low carbon steel (e.g., SAE 1020) with polished surfaces. Ruiz[6.87] arrives at the same qualitative conclusions in fatigue tests of AISI type 321 stainless steel, again with polished surfaces. Ruiz was able to produce fatigue failures in less than 100,000 cycles only by using pressures in excess of 85% of the burst pressure; corresponding to about 1.6 times the yield pressure.

Because piping systems are generally limited to pressures corresponding to some fraction of the yield pressure, it seems safe to assume that for materials with a reasonable ratio of ultimate strength to yield strength (e.g., a ratio of 1.5 or larger), fatigue failure will not occur in a pipe with $D/t > 10$ due to cyclic pressure in the absence of notches or out-of-roundness of the pipe.

Test data on fatigue of thick wall cylinders are given by Morrison, Crossland, and Parry[6.88]. Data are presented for cylinders with O.D./I.D. ratios from 1.2 to 3.0, made of seven different materials. It was found, for most of the seven materials, that the test results correlated fairly well with the magnitude of the maximum shear stress at the bore of the cylinders, i.e.,

$$\tau = \frac{1}{2}\left[\frac{K^2 + 1}{K^2 - 1} + 1\right]P \qquad (6.34)$$

where $\tau$ = maximum shear stress at bore

$K$ = ratio of O.D. to I.D. of cylinder

$P$ = internal pressure.

However, the magnitude of the shear stress range at the endurance limit in the cyclic pressure tests was only about one-half of the shear stress range for the material endurance strength when tested as a solid bar in reversed torsion. The cyclic pressure tests involve a mean stress equal to one-half of the stress amplitude; one might ascribe some part of the discrepancy noted above to the mean stress. This aspect does not appear to be sufficient to explain the entire discrepancy because:

(1) Cyclic pressure tests on thin-wall cylinders, in which the same ratio of mean stress to variable stress is involved, do not show this large discrepancy.

(2) Generally valid methods of accounting for mean stress would not be sufficient to account for the discrepancy. For example, the modified Goodman diagram equation (which usually overestimates the mean stress effect) is:

$$S_{eq} = \frac{S_a}{1 - \frac{S_m}{S_u}} \qquad (6.35)$$

$S_{eq}$ = equivalent completely reversed stress amplitude

$S_a$ = stress amplitude

$S_m$ = mean stress

$S_u$ = ultimate strength of the material.

Equation (6.35), as applied to Morrison's test data, would give a correction for mean stress of about 20%.

Possible reasons for this seeming anamoly are discussed by Morrison, et. al.[6.88], without coming to a firm conclusion. Among other aspects of these high pressure tests is that of "hydrowedging". The hypothesis is that at high pressures the test fluid penetrates into microcracks that may be initially present, or into the small cracks formed in the early stages of the test. As the cracks close, the fluid is partially trapped in the cracks thereby causing a large increase in stress at the root of the cracks. This hypothesis is supported to some extent by the sensitivity of Morrison's result to surface finish and/or heat treatment of the bore. On the other hand, Morrison's direct test data on this effect, in which he obtains the fatigue life of a solid test specimen surrounded by the test fluid (oil) at 45,000 psi, indicates that the effect is small.

Parry[6.89] gives additional test data, similar to those given by Morrison, Crossland, and Parry[6.88] on 6 additional materials. These results also indicate that the shear stress range in the cyclic pressure tests on thick cylinders is about one-half of the shear stress range for the material.

Hannon[6.90] attempts a correlation of the test data of Morrison, et al. for one of the materials tested. Hannon introduces semi-empirical correction factors for hydrowedge, mean stress, and size effect. Some of these factors are plausible, but not convincing. Accordingly, it appears that the test data in References (6.88) and (6.89) on the fatigue of thick wall cylinders indicate that fatigue analysis based on maximum shear theory can be quite unconservative for this specific application, for reasons not clearly established at this time.*

The above discussion has been concerned with "ideal" cylinders; i.e., round, uniform-wall, free of surface defects. Quantitative data on pipe, with typical surface defects, out-of-roundness and welds are apparently rare or non-existent. However, some qualitative data exist, which are discussed in the following.

Dubuc and Welter(6.91) and Welter and Dubuc(6.92) ran tests on cylinders made of A201-GrA, A302-GrB or T1 plate. The primary purpose of the tests was to determine the fatigue life under cyclic pressure of nozzles in these cylindrical shells. However, the test models included longitudinal welds, girth welds to heads and intentional surface defects in the form of milled notches and partial holes. The data give some information (mostly lower bound) with respect to the fatigue strength of these details. The cylindrical shell was made from 0.75" thick plate, rolled to half-cylinders and made into cylinders with two longitudinal welds. Tests consisted of application of cyclic pressure from about zero (presumably) to a maximum pressure about equal to the yield pressure of the cylinder.

The failures of interest herein are those that occurred in longitudinal welds, girth welds or notches. Only one failure occurred in a longitudinal weld, in vessel M, at 39,100 cycles of 0 to 5700 psi pressure (max. pressure corresponds to 95% of yield pressure of cylinder). Several end closures were used: flat heads, elliptical heads and spherical heads. The

* Reviewers of this report have suggested that exhaustion of ductility by the building up of large cumulative mean strains may be the reason for the seeming discrepancy.

first tests were run with flat heads; these were abandoned after the first pair of tests because failure occurred "very soon" under some (unspecified) cyclic pressures. Several failures occurred in girth butt welds to elliptical heads; in the last pair of vessels tested in Reference (6.91), spherical heads were used, with no girth weld failures. However, in Reference (6.92) spherical heads were used and failures in the shell-to-head welds occurred. Only one failure occurred in the intentional surface defects; this was in one of six notches, 0.06" deep, made with a standard Charpy V-notch cutter and oriented with their lengths parallel to the vessel axis. This failure occurred in vessel M at 46,000 cycles of 0 to 5700 psi cyclic pressure (max. pressure corresponds to 95% of yield pressure of cylinder).

As remarked earlier, the above tests were run primarily to determine the fatigue life of nozzles in cylinders; insufficient details of failures at other points negate any quantitative conclusions.

Rodabaugh and George[(6.93)] ran a series of cyclic pressure tests, including straight pipe with a longitudinal weld. The pressure cycle was from about 50% to 90% of the yield pressure of the pipe. Fatigue failures occurring at the longitudinal welds ranged from failure at 144,000 cycles up to 900,000 cycles without failure. These longitudinal welds were made by the pipe manufacturers, using either automatic resistance welding or automatic submerged arc welding. The large spread in the test results presumably is due to:

(1) On some of the specimens, the external weld flash had been partially removed by a planing cutter. At some areas along the weld, this cutter formed a sharp groove in the pipe surface parallel to the weld [photos are shown in Reference

(6.93)]. Fatigue failures started in these grooves.

(2) There was some local out-of-roundness of variable severity in the weld region of the pipes.

These tests also involved a large number of girth butt welds between straight pipe and ASA B16.9 welding caps. In the writer's recollection, no fatigue failures occurred in these girth welds.

Pickett and Grigory[(6.94)] and Pickett, et. al.[(6.95)] give results of cyclic pressure tests on "large size pressure vessels". These vessels consist of cylinders with longitudinal welds. In the large size vessel tests, no fatigue failures developed in the longitudinal welds, per se, however in some of the vessels (e.g., Vessel No. 5, Vessel No. 6, Nozzle N-9), the fatigue failures at nozzles may have been influenced by out-of-roundness associated with the longitudinal weld. In the half scale vessel tests, the longitudinal weld of Model F failed at 11,707 cycles of 0 to 3500 psi pressure (nominal hoop stress at 3500 psi pressure is 33,000 psi). In the SwRI half scale model, failure of the longitudinal weld occurred in 227,685 cycles of 0 to 4000 psi, however, at 225, 240 cycles a pressure in some unknown amount above 4000 psi was applied. Apparently, there was a significant amount of out-of-roundness associated with these longitudinal weld failures.

Morikawa and Griffis[(6.84)] include some results on cylinders with a simulated longitudinal weld. The "welded" specimens were prepared by rough-turning the specimen and milling a 60° V-shaped slot longitudinally along the full length of the unbored specimen at opposite extremities of a diameter. This slot was 1/2 inch deep and was filled with weld metal prior to boring and finish machining to 1 inch I.D. x 0.050 inch wall. The slot depth was such that the final .050 inch of wall thickness was located at approximately the center of the weld. Welding rod was AWS E 6010. These are significant tests in that they represent an "ideal" weld, finish machined inside and outside, with no out-of-roundness. The results indicate a decrease in fatigue strength as compared to seamless specimens; by a factor of around 0.85 on stress.

Ruiz(6.87) compares low-cycle fatigue test data for cyclinders having a uniform wall thickness with data for much thicker cylinders having a longitudinal notch, which reduced the remaining wall thickness at the notch to that of the uniform wall cylinders. The specimens were made of type 321 stainless steel. For failure in less than $10^5$ cycles, cyclic pressures corresponding to 70% of the burst pressure (~ 1.33 times nominal yield pressure) were required, even for the cylinders with severe notches.

The cyclic-pressure fatigue test data can be summarized as follows:

(1) For thin-wall cylinders, without notches, the value of $P_{max}$ (in a pressure cycle from 0 to $P_{max}$) must exceed the yield pressure in order to obtain failures in less than 100,000 cycles. The endurance value of $P_{max}$ is about equal to the yield pressure.

(2) Thick-wall cylinders present an anomaly in that the endurance shear stress range is only about one-half of the endurance shear stress range expected from material tests.

(3) The available data on welds and notches are too limited to reach in general conclusions. For longitudinal welds, the out-of-roundness near the weld may be significant in addition to irregularities of the weld.

6.552 Cyclic Moments

Tubular test specimens, subjected to combinations of bending moment and torsion, have been used to investigate combined stress theories of fatigue failure. These tests, on specimens with polished surfaces, by-and-large indicate that the fatigue failure of such tubes can be estimated by use of the maximum shear criteria, obtaining stresses from the equations $\sigma_b = Mc/I$ and $\sigma_s = Tc/2I$.

Commercial pipe does not have polished surfaces, nor is it round nor of uniform wall thickness. A summary of bending tests of straight pipe available at the time (1952) is given by Markl[6.96]. Markl's tests were run on forged carbon steel (comparable to ASTM A106 Gr B properties) transition pieces, with a gradual taper between the "pipe" section and a heavier section used as the anchor in a cantilever beam test. Markl compares test results of pipe with that of polished bars of the same (carbon steel) material. In the failure cycle range of $10^3$ to $10^5$, the pipe shows about the same fatigue strengths as the polished bars; i.e., the surface effect was negligible. At lower cycles, the pipe fatigue strength was higher than that of the polished base; probably because the pipe tests were deflection controlled whereas the polished bar tests were load controlled. At higher cycles, the pipe fatigue strength is lower than that of the polished bars; i.e., as normally the case, the surface finish is significant for a large number of cycles.

Additional tests on straight pipe are reported by Newman[6.97] and O'Toole and Rodabaugh[6.98]. Newman's tests (his Series A) were run on 6.625 inch outside diameter by 0.375 inch wall made of carbon steel comparable to ASTM A106 GrB. O'Toole and Rodabaugh's tests were run on 4-inch std. wt. pipe, 8 specimens made of A106 GrB material, 4 of ASTM A312 type 304 material. Both Newman, and O'Toole and Rodabaugh used resonance bending testing in which the pipe is vibrated in the "free-free" first mode, supported at the node points. The maximum bending stress occurs in the center of the pipe length; remote from any entraneous stress raisers. O'Toole and Rodabaugh[6.98] plot test results from Markl[6.96], Blair[6.99], and Newman[6.97], along with their own results for carbon steel pipe. These are all quite adequately represented by the equation:

$$SN^{0.2} = 383,000 \tag{6.36}$$

where S = nominal bending stress amplitude (Mc/I)

N = cycles-to-failure (data covers range of $2 \times 10^2$ to $4 \times 10^6$).

There is some small evidence of a knee in the S-N data at around $S = 18,000$ psi, $N = 4 \times 10^6$. At high cycles, the stress intensification factor with respect to polished bar tests is about two.

O'Toole and Rodabaugh[6.98] results for A312 TP 304 pipe indicate that such pipe has a significantly longer fatigue life than carbon steel pipe. This difference presumably arises from two sources: (1) the relatively better surface finish of stainless steel pipe and (2) the better fatigue strength of 304 stainless steel. The tests do not cover a sufficient range of stress to construct an S-N curve, however if expressed in the form of equation (6.36), the S-N equation would be:

$$SN^{0.2} = 445,000 \tag{6.37}$$

Markl[6.96] also gives extensive data on the fatigue strength of typical butt welds in 4-inch standard weight pipe. At about $10^5$ and higher cycles (where polished bar load-controlled and pipe deflection tests are directly comparable, because the nominal stresses are elastic) the fatigue-effective stress intensification factor of welds without a backing ring is about two. Welds with a backing ring had lower fatigue lives.

Some additional bending fatigue tests on girth butt welds in pipe have been reported by Newman[6.97], O'Toole and Rodabaugh[6.98], Meister, et.al.[6.100], and Dawes[6.101]. Markl's test results are represented by the equation:

$$iSN^{0.2} = 245,000 \tag{6.38}$$

where i = stress intensification factor (Markl's data, i = 1.0)

S = nominal bending stress, Mc/I, psi

N = cycles-to-failure (Equations valid for N in the general range of $10^2$ to $10^6$ cycles).

The results of subsequent test data can be compared in terms of the i-factor. Such a comparison is shown in Table 6.1. Also shown in Table 6.1 is an $i_1$-factor; this represents the stress intensification factor of the weld with respect to typical pipe without a weld.

Meister, et.al.,[6.100] also includes a rather extensive series of tests on defects introduced in the welds. Consistently harmful defects were found to be root concavities and root undercuts. This agrees with the findings of Newman[6.97] that root defects are most significant in typical pipe girth butt welds subjected to cyclic bending loads.

TABLE 6.1. SUMMARY OF TEST DATA ON FATIGUE OF BUTT GIRTH WELDS IN PIPE UNDER REVERSED BENDING LOADING

| Reference | Pipe Size | Material | i (1) | $i_1$ (2) |
|---|---|---|---|---|
| Markl (6.96) | 4.5 x 0.237 | A106 GrB | | |
| No backing rings | | | 1.00 | 1.56 |
| With backing rings | | | 1.22 | 1.91 |
| Newman (6.97) | 6.625 x 0.375 | B.S. 806, Class B | | |
| Control welds | | | 1.19 | 1.86 |
| Porosity Series G | | | 1.49 | 2.33 |
| Transline Slag H | | | 1.38 | 2.16 |
| Gross Defects I | | | 1.54 | 2.40 |
| Lack of Fusion J | | | 1.28 | 2.00 |
| Lack of Penetration K | | | 3.45 | 5.40 |
| Piping L | | | 1.57 | 2.45 |
| O'Toole and Rodabaugh (6.98) | 4.5 x 0.237 | | | |
| Fusion, As welded | | A106 - GrB | 0.82 | 1.28 |
| Fusion, Overlay Ground Flush | | | 0.78 | 1.22 |
| Conventional, Acceptable | | | 0.89 | 1.39 |
| Conventional, Rejectable | | | 0.99 | 1.54 |
| Fusion, As welded | | A312 TP 304 | 0.66 | 1.20 |
| Fusion, Overlay Ground Flush | | | 0.69 | 1.25 |
| Conventional, Rejectable | | | 0.81 | 1.47 |
| Meister, et.al. (6.100) (3) | | | | |
| Backing rings | 2.375 x 0.154 | A106 GrB | 1.00 | 1.56 |
| Cons. Insert | " " | A106 GrB | 0.77 | 1.20 |
| Backing rings | " " | 70-30 Cu-Ni | 1.27 | -- |
| Cons. Insert | " " | Monel | 0.80 | -- |
| Backing rings | 4.5 x 0.237 | A106 GrB | 1.16 | 1.81 |
| Cons. Insert | " " | A106 GrB | 0.89 | 1.39 |
| Backing rings | " " | 70-30 Cu-Ni | 1.48 | -- |
| Cons. Insert | " " | 70-30 Cu-Ni | 0.93 | -- |
| Cons. Insert | " " | Monel | 0.74 | -- |
| Dawes (6.101) (3) | 4.5 x 0.25 | Mild steel | | |
| TIG root runs | | | 1.12 | 1.75 |
| TIG root runs, C.B.R. (4) | | | 0.86 | 1.34 |
| MIG root runs, C.B.R. | | | 0.96 | 1.50 |

(1) i in equation $iSN^{0.2} = 245{,}000$

(2) $i_1$ in equation $i_1SN^{0.2} = 383{,}000$ for carbon steel

$i_1SN^{0.2} = 445{,}000$ for TP 304 steel

(3) Comparisons made at 5 x $10^5$ cycles-to-failure.

(4) C.B.R. ≡ Ceramic Backing Ring

It is beyond the scope of this report to discuss in detail the tests summarized in Table 6.1; the reader should consult the references cited for a number of pertinent factors not covered herein. However, the following general observations can be made:

(1) The test data on girth butt welds are in general agreement with the data presented by Markl and used in ASA Piping Codes.

(2) The data indicate that 304 stainless steel is a little more tolerant of the notches associated with welds than is carbon steel.

(3) By careful control of weld root conditions, such as obtained by use of a consumable insert or a weld-land fusion, along with a reasonably smooth overlay, it is possible to produce a girth butt weld almost equal in fatigue strength to that of the pipe in which it is placed.

(4) Table 6.1 indicates that a girth butt weld is always weaker in fatigue than the pipe in which it is used. On the average, this is true, however three cases are reported by O'Toole and Rodabaugh in which failures occurred in the pipe rather than in the butt weld. One case occurred in carbon steel pipe at a metal-stamped identification number. The other two occurred in 304 pipe, at "draw drags" in the outside surface of the pipe.

The data on fatigue strength of girth butt welds in pipe with wall thickness in the range of 1/4 to 3/8 inch appear to be adequate for design purposes. Two questions arise:

(1) What stress intensification factors should be used for thicknesses beyond the range of the test data?

(2) What stress intensification exists for torsional loading?

With respect to the first question, it might be noted that weld irregularities are, in general, not proportional to the thickness. For example, in a pipe with 2-inch wall thickness one might reasonably expect the overlay and root defects to be smaller in proportion to the pipe thickness than for a typical weld in 3/8-inch wall pipe. In this sense, the stress intensification factors given by test data may be unduly conservative when applied to thick wall pipe. Some guidance in this area can be obtained from extensive literature of the fatigue strength of butt welds in plates. Newman[(6.102)] gives a survey of the available literature (1959) on effects of defects on such joints. Since 1959 considerable additional data have been published, mainly by the British Welding Research Association. References (6.103) thru (6.116) are related to this work.

The second question concerning the fatigue strength of girth butt welds with torsional loading is more difficult. The writer has not found any published test data which would bear directly on the question.

There does not appear to be any test data on the effect of longitudinal welds on the fatigue strength of pipe subjected to bending or torsion loads. For bending, presumably the effect would be minor; the same may not be true for torsion.

## 6.56 Fracture Behavior of Defects

### 6.561 Axially-Oriented Cracks, Internal Pressure Loading

In paragraph 6.53, test data on the maximum pressure capacity or burst pressure of straight pipe without intentional defects are referenced and briefly discussed. A perhaps more important aspect concerns the failure pressure of pipe with defects; more important because in the presence of defects the failure pressure may be within the range of normal design pressure of the pipe.

Fractures in piping with controlled defects have been studied by a number of investigators; References (6.117) through (6.125). The studies discussed below are those concerned with fractures in the low to medium strength structural materials.

The conditions governing the initiation of fracture have been studied by Duffy, et. al.[6.126] for piping made of plain carbon and low-alloy steels with tensile strengths generally below 120,000 psi. The pipe diameter to thickness ratios (d/t) ranged from 20 to 100. Both ductile and brittle fracture initiation were studied over a temperature range from -100 to +150 F.

Data on relatively heavy-walled piping (d/t from 8 to 15) are given by Eiber, et. al.[6.127]. These tests were run at elevated temperature (500-700 F) to determine critical crack size and the extent of crack propagation under various subcooled water and boiling water conditions such as those employed in boiling water and pressurized water cooled reactors.

Based on the tests described above, equations have been developed for predicting the failure pressure of a pipe containing either surface flaws or through-wall flaws oriented parallel to the pipe axis (normal to the hoop stress direction). These equations are given for general information, however, the references cited should be consulted concerning their detailed background and potential limitations.

I. For through-the-wall flaws

$$K^2 = \frac{\pi c\, \sigma_h^2}{\cos\,\theta}\left[1 + 1.61\,\frac{c^2}{rt}\right] \qquad (6.39)$$

where:

$K$ = stress intensity factor, Ksi $\sqrt{\text{in.}}$

$c$ = half of the crack length (through-the-wall cracks), in.

$r$ = mean radius of pipe, inches

$t$ = average wall thickness of pipe, inches

$\theta = \frac{\pi}{2}\frac{\sigma_h}{\sigma_c}$

$\sigma_h$ = nominal hoop stress at failure pressure of flawed pipe,

$\sigma_h = P_f r_i/T$, psi

$P_f$ = failure pressure of pipe with a flaw, psi

$r_i$ = inside radius of pipe, inches

$\sigma_c$ = nominal hoop stress at failure pressure of unflawed pipe,

$\sigma_c = P_u r_i/t$

$P_u$ = failure pressure of pipe without flaws, psi.

Some comments by the writer concerning equation (6.39):

(1) The stress intensity factor K, when defined for the critical stress at which point a defect has just become unstable and is starting to propagate, is commonly referred to as $K_c$ for the plane stress condition or $K_{Ic}$ for the plane strain condition.

(2) Equation (6.39) is explicit for the value of K; however, the piping designer is not directly interested in the value of K anymore than he is interested in the tensile strength of the pipe material. What the designer ordinarily needs is the value of failure pressure, $P_f$, as a function of crack length, c, or possibly c as function of $P_f$ so that he can examine his design and/or operating pressures in terms of potential defects in the pipe. Having established values of K and $\sigma_c$, these relationships are implicit in equation (6.39); they cannot be expressed explicitly because $P_f$ is involved in the cos θ term.

(3) To use equation (6.39), a value of $\sigma_c$ or $P_u$ must be established. Equation (6.39) cannot be used to obtain this value by letting $c = 0$ because $\sigma_h/\sigma_c = 1.0$, $\cos [(\pi/2)(\sigma_h/\sigma_c)] = 0$ with an indeterminate result. In paragraph 6.53, it was noted that the fracture pressure of an unflawed pipe is reasonably well predicted by the mean diameter equation using the material ultimate tensile strength; i.e.,

$$P_u = \frac{S_u t}{r} \tag{6.40}$$

from which:

$$\sigma_c = S_u \frac{r_i}{r} \tag{6.41}$$

where $S_u$ = ultimate tensile strength of pipe material, psi

$r$ = mean pipe radius

t and r as defined under equation (6.39).

In tests run at Battelle-Columbus, a suitable value for $\sigma_c$ has been found to be $(S_u + S_y)/2$, where

$S_y$ = yield strength of pipe material, psi.

II. For surface flaws

$$\frac{\sigma_{hs}}{\sigma_c} = \frac{A_o - A}{A_o - A\left(\frac{\sigma_h}{\sigma_c}\right)} \qquad (6.42)$$

where

$\sigma_{hs}$ = nominal hoop stress at failure pressure of flawed pipe,

$\sigma_{hs} = P_{fs} r_i / t$, psi

$P_{fs}$ = failure pressure of pipe with a surface flaw, psi

$\sigma_c$ = nominal hoop stress at failure pressure of unflawed pipe, psi

$A_o$ = 2ct, area of through wall defect of same length as surface defect, sq in.

A = cross sectional area of surface defect, sq in.

$(\sigma_h/\sigma_c)$- obtained from equation (6.39)

III. For through-the-wall defects, ductile fracture

$$P_f = \frac{S_f t}{r_i}\left[1 + \frac{1.61\ c^2}{rt}\right]^{-1/2} \qquad (6.43a)$$

or

$$c^2 = \left[\left(\frac{S_f t}{P_f r_i}\right)^2 - 1\right]\frac{rt}{1.61} \qquad (6.43b)$$

where

$S_f = (S_u + S_y)/2$, psi

$S_u$ = material ultimate tensile strength, psi

$S_y$ = material tensile yield strength, psi

other symbols as defined under equation (6.39).

Some comments by the writer on Equation (6.43):

(1) It should be emphasized that equation (6.43) is limited to temperatures, materials, thicknesses, etc. where the failure is ductile.* However, the formulation is quite useful, where applicable*, because the failure pressure or critical crack size can be calculated with material properties from an ordinary tensile test.

(2) It may be noted that by letting c = 0 in equation (6.43a), the burst pressure is given by:

$$P_f = \frac{(S_u + S_y)}{2} \frac{t}{r_i} \tag{6.44}$$

This equation, for $S_y$ considerably less than $S_u$, and for $r_i$ approximately equal to r, would give lower predicted failure pressures than obtained from equation (6.40). The writer is not aware of any test data that can be used to check equation (6.43a) for very small crack sizes; its use for very small crack sizes appears to be conservative.

Equation (6.39) described above has been compared with test data by Getz, et. al.[6.123] and Lake, et. al.[6.128]; both of these sets of data being on pipe made of aluminum alloys. The correlations were quite good.

General Electric (APED-San Jose, California) has been conducting and are continuing similar tests on critical crack size. The latest available quarterly report[6.129] gives results in general agreement with the formulations discussed above. It was reported that in

* Failure pressure calculated by equation (6.43a) is an upper bound value and the pressure may not be reached for non-ductile failure conditions.

equation (6.39) the 1.61 factor should be 0.4 to produce $K_c$ values that are constant for a wide range of crack lengths.

Kihara, et. al.[6.120] conducted a series of tests on gas transmission pipe at -320 F. The purpose of the tests was to determine critical crack sizes and the effect of pipe diameter and thickness in the brittle fracture regime. The formulation developed in the study is similar to that proposed in Reference (6.127).

Irvine, et. al.[6.130] conducted a series of tests on 9'-6", 5' and 3' diameter cylindrical vessels over a range of temperature to determine critical crack sizes. The tests covered the transition from ductile to brittle behavior. As a result of the study they developed an empirical formulation which uses Charpy impact energy as a measure of the toughness of the material and also employs semi-empirical constants.

The controlled defects used in the experimental work discussed above were almost always made by machining or sawing. The question arises as to whether cracks produced by fatigue would exhibit behavior similar to the machined flaws. Two cyclic pressure tests of thin-walled pipes have been reported where flaws were extended to determine their critical size for comparison with the same size machined flaws. In the ductile range of behavior for the low alloy, no difference in the critical crack size was observed.

Hahn, et. al.[6.132] has prepared a report which abstracts much of the data discussed above and compares the data with equation (6.39) and similar correlation equations.

6.562 Circumferentially Oriented Cracks, Internal Pressure Loading

For internal pressure loading, the circumferentially oriented crack would be expected to be less critical than the axially oriented crack because the nominal axial stress is about one-half of the hoop stress.

Test data on circumferentially oriented (along with cracks at other orientations) are given in References (6.133), (6.134), and (6.129). Apparently no general correlation formulas have been made for these crack orientations.

6.563 Critical Crack Size, External Moment Loading

For piping systems subjected to high external moment and/or torsional loadings, the critical crack size may not be axially oriented; the crack orientation and critical size may depend upon the particular combination of internal pressure and moment applied to the pipe. In-so-far as the writer is aware, no tests have been run directly to study this aspect. The bending fatigue tests being run at General Electric[6.129] in which circumferential notches are placed in the pipe and fatigue crack growth determined, should give some guidance in this area.

6.564 Propagation of Fractures

An important aspect of fracture concerns the extent of the fracture propagation. This depends to some degree on the loading involved; e.g., a pipe pressurized with a gas will generally result in greater propagation of fracture than the equivalent pipe pressurized with a liquid. The conditions involved in the propagation of fractures in thin-wall, carbon steel piping have been examined in over 100 tests employing combinations of air and water, and all natural gas as pressurizing media[6.135].

A general overall approach to the design of pressure vessels and structures employing "stiff" loading systems (e.g., a pipe with essentially incompressible liquid as a pressurizing medium) has been developed by Pelleni[6.136] and co-workers. They have developed a series of diagrams that outline an engineering approach to the selection and evaluation of materials for both fracture initiation and propagation.

6.565 Thermal Stresses and Residual Stresses

While there have been speculations that thermal stresses and/or residual stresses have been significant in some service-experienced brittle fractures, the writer does not know of any test data which either proves or disproves the significance of these kinds of stresses in establishing critical crack sizes.

6.566 Effect of Pre-Service Test

An area which has recently been investigated concerns the effect of a pre-service hydrostatic test on the subsequent behavior of flaws in the vessel or piping. Two reports have been published. Reference (6.137) summarizes the results of experiments with thin-walled piping. Nichols[6.138] summarizes the results of an extensive literature review on all types of structures and vessels. Both reports indicate that a pre-service pressure test in the range of 1.25 times the operating pressure can have beneficial effects in service. These studies are mostly concerned with internal pressure loading following the pre-service hydrostatic test. The possible effect of other service loadings (e.g., external moments, thermal stresses) does not appear to be included in the evaluation although it seems likely that the pre-service hydrostatic test would be beneficial even for other-than-pressure loadings.

## 6.6 Local Loads

Piping systems must be supported, braced, guided, anchored, etc. Such restraints may introduce local loads into the pipe, particularly where "integral attachments" such as those shown in Chapter 15 Figure 15.1 are used. Alternately, "non-integral attachments" can be used where, by definition, the attachment is not welded to the pipe. These may consist of bolted-clamps, slings, clevises, or saddle supports. Such non-integral attachments are ordinarily used for moderate service conditions. At elevated temperatures, and particularly for support of vertical piping, it is difficult to maintain the necessary frictional forces and some type of integral attachment is necessary.

Integral attachments must be carefully designed in order to avoid failures at the attachment. Thielsch[6.144] and Hahn[6.145] discuss service failures of integral attachments and give some qualitative suggestions for improved designs.

Integral attachments are subjected to three types of loadings:

(1) Internal pressure in the pipe

(2) External loads (weight, restraint of movement, etc.) applied to the attachment from the supporting structure

(3) Thermal gradients, in particular a temperature difference between the pipe wall and the attachment.

Pressure vessel and piping codes, with one exception, do not give specific rules for design of attachments. The one exception is in the ASME Boiler Code, Section I, Power Boilers, Par UW-43, wherein a specific method of determining the size of integral lugs for supporting tubes is given. The background of this design procedure is discussed by Melworm and Berman[6.146].

Reference (6.146) also gives the results of four series of tests run at the Foster Wheeler Corporation.

The analysis of stresses at attachments presents a rather formidable problem because the structure and loadings are not axisymmetric. Further, maximum stresses usually occur at fillet welds joining the attachment to the structure and the local stresses depend upon the detailed configuration of the weld. The kind of "fit-up" between attachment and pipe may also significantly affect the maximum stresses. In principle, finite-element computer programs such as those discussed in Chapter 3, Par 3.12, could be used to evaluate stresses at integral attachments. In addition, there are a number of analysis methods which can be applied to analysis of stresses at attachments. Roark[6.147] gives formulas for stresses in cylinders with a radial load distributed over a small area and for a horizontal cylindrical shell on saddle supports. Hoff, Kempner, and Pohle[6.148] and Cooper[6.149] give analyses for cylindrical shells under a line load. Bijlaard[6.150] published a series of papers on stresses in cylindrical shells with local loadings. Bijlaard's work, along with correlations with experimental data, forms the basis for a widely used design procedure given by Wichman, Mershon, and Hopper[6.151]. Melworm, Patel, and Berman[6.152] give analyses for balanced radial loads on long cylinders.

Experimental data on stresses at attachments are given by Mehringer and Cooper[6.153] and by Cranch[6.154]. Test data on branch connections discussed in Chapter 8 are also applicable, to some extent, to attachments; the difference herein is that "attachments" are considered as welded to the pipe with no opening in the pipe.

All of the theories and test data discussed above are concerned with internal pressure and/or external loads. Design methods or test data for thermal gradients have not been found in the literature.

## 6. REFERENCES


(6.1) Lame', G., "Lecons sur la Theorie Mathematique de l'Elasticite des Corps Solides", Bachelier, Paris, France, pp 188-191 (1852).

(6.2) Buxton, W. J. and Burrows, W. R., "Formula for Pipe Thickness", Trans. ASME, Vol. 73, p. 575 (1951).

(6.3) Marin, J. and Sharma, M. G., "Design of Thin-Walled Cylindrical Pressure Vessels Based Upon the Plastic Range and Considering Anisotropy", Welding Research Council Bulletin No. 40, May, 1958.

(6.4) Rodabaugh, E. C., Atterbury, T. J., and McClure, G. M., "Yielding of Thin-Wall Pipe Under Combined Pressure and Bending", Battelle Memorial Institute, Columbus, Ohio, May, 1962.

(6.5) Nadai, A., Theory of Flow and Fracture of Solids, McGraw-Hill Book Co., New York (1931).

(6.6) MacGregor, C. W., Coffin, L. F., and Fisher, J."The Plastic Flow of Thick-Walled Tubes with Large Strains", J. Applied Physics, Vol. 9, No. 3, p. 291 (1947).

(6.7) MacGregor, C. W., Coffin, L. F., and Fisher, J. C., "Partially-Plastic Thick-Walled Tubes", J. of the Franklin Institute, Vol. 245, p. 135 (1948).

(6.8) Hill, R., Lee, E. H., and Tupper, S. J., "Plastic Flow in a Closed-End Tube with Internal Pressure", Proc. of First U.S. National Congress of Applied Mechanics, p. 561 (1952).

(6.9) Bland, D. R., "Elastoplastic Thick-Walled Tubes of Work-Hardening Material Subject to Internal Pressures and to Temperature Gradients", J. of the Mechanics and Physics of Solids, Vol. 4, p. 209 (1956).

(6.10) Hannon, B. M., and Sidebottom, O. M., "Plastic Behavior of Open-End and Closed-End Thick Wall Cylinders", ASME Paper No. 67-WA/PVP-8.

(6.11) Turner, L. B., "The Stresses in a Thick Hollow Cylinder Subjected to Internal Pressure", Trans. Cambridge Publ. Society, Vol. 21, No. 14, pp. 377-396 (1910).

(6.12) Manning, W.R.D., "The Overstrain of Tubes by Internal Pressure", Engineering, Vol. 159, pp. 101-102, 183-184 (1945).

(6.13) Nadai, A., Plasticity, McGraw-Hill Book Company, New York (1931).

(6.14) Bailey, R. W., "Thick-Walled Tubes and Cylinders under High Pressure and Temperature", Engineering, Vol. 129, pp 772-777, 785-786, 818-819 (1930).

REFERENCES (contd.)

(6.15) Jorgensen, S. M., "Overstrain and Bursting Strength of Thick-Walled Cylinders", ASME Paper 57-PET-4.

(6.16) Beeuwkees, B., and Laning,J. H., "Strains in Gun Tubes Calculation of Pressure Expansion Curve of Circular Cylinders", Watertown Arsenal Lab., Experimental Report 730/111, 1944.

(6.17) Hu, L. W., "Finite Deformation of Thick-Walled Cylinders Under Radial Pressure", Dept. of Engr. Mechanics, Penn. State U., 1953.

(6.18) Marin, J. and Rimrott, F.P.J., "Design of Thick-Walled Pressure Vessels Based Upon the Plastic Range", Welding Research Council Bulletin No. 41, May, 1958.

(6.19) Marin, J. and Weng, Tu-Lung, "A Critical Evaluation of the Strength of Thick-Walled Cylindrical Pressure Vessels", Welding Research Council Bulletin No. 74, Jan., 1962.

(6.20) Crossland, B., "The Design of Thick-Walled Closed-End Cylinders Based on Torsion Data", Welding Research Council Bulletin No. 94, February, 1964.

(6.21) Marin, J., Discussion of Reference (6.19).

(6.22) Marin, J. and Weng, Tu-Lung, "Strength of Thick-Walled Cylindrical Pressure Vessels", Trans. ASME, J. of Engineering for Industry, November, 1963, p. 405.

(6.23) Cooper, W. E., "The Significance of the Tensile Test to Pressure Vessel Design", Welding Research Supplement, Jan., 1957, p. 49-s.

(6.24) Finnie, I., "Stress Analysis in the Presence of Creep", Applied Mechanics Reviews, Vol. 13, No. 10 (1960).

(6.25) Bailey, R. W., "The Utilization of Creep Test Data in Engineering Design", Proc. Instn. Mech. Engrs., Vol. 131, p. 131 (1935).

(6.26) Johnson, A. E., Henderson, J., and Khan, B., "Behavior of Metallic Thick-Walled Cylindrical Vessels or Tubes Subject to High Internal or External Pressures at Elevated Temperatures", Proc. Instn. Mech. Engrs., Vol. 175, p. 1043 (1961).

(6.27) Rimrott, F.P.J., "Creep of Thick-Walled Tubes Under Internal Pressure", J. of Applied Mechanics, Trans. ASME, Vol. 81, Series E, p. 271 (1959).

(6.28) Rimrott, F.P.J., Mills, E. J., and Marin, J., "Prediction of Creep Failure Time for Pressure Vessels", J. of Applied Mechanics, Trans. ASME, June, 1960, p. 303.

REFERENCES (contd.)

(6.29) King, R. H., and Mackie, W. W., "Creep of Thick-Walled Cylinders", J. of Basic Engineering, Trans. ASME, Vol. 89, Series D, p. 877 (1967).

(6.30) Pai, D. H., "Steady-State Creep Analysis of Thick-Walled Orthotropic Cylinders", International J. of Mech. Sciences, Vol. 9, p. 335 (1967).

(6.31) Haigh, B. P., "An Estimate of the Bending Stresses Induced by Pressure in a Tube that is Not Initially Quite Circular; Proc. Instn. Mech. Engrs., Vol. 133, pp. 96-98. (1936)

(6.32) Schmidt, K., "Stress in an Out-of-Round Pressure Vessel, V.D.I., Vol. 102, No. 1, p. 11 (1960).

(6.33) ASME Boiler and Pressure Vessel Code, Section VIII, Par. UF-27, (1965 Edition), Published by the American Society of Mechanical Engineers, 345 E. 47th St., New York, N. Y. 10017

(6.34) Schmidt, K., "Calculation of Stress for an Out-of-Round Tube Under Internal Pressure", V.D.I., Vol. 98, No.4, p. 121 (1956).

(6.35) Nadai, Theory of Flow and Fracture of Solids, Vol. 1, McGraw-Hill Book Company, New York, New York. (1950)

(6.36) Hoffman, O. and Sachs, G., Introduction to the Theory of Plasticity for Engineers, McGraw-Hill Book Company, New York, N.Y.

(6.37) Phillips, A., Introduction to Plasticity, The Ronald Press Company, New York, New York.

(6.38) Ades, C. S., "Bending Strength of Tubing in the Plastic Range", J. of the Aeronautical Sciences, August, 1957, p. 605.

(6.39) Timoshenko, S. P. and Gere, J. M., Theory of Elastic Stability, 2nd Edition, p 482, McGraw-Hill Book Company, New York, N. Y.

(6.40) Robinson, E. L., "Steam-Piping Design to Minimize Creep Concentrations", Trans. ASME, Vol. 77, p. 1147 (1955).

(6.41) Stokey, W. F., Peterson, D. B., and Wunder, R. A., "Limit Loads for Tubes Under Internal Pressure, Bending Moment, Axial Force and Torsion", Nuclear Engineering and Design, Vol. 4, p. 193 (1966).

(6.42) Johnson, A. E., "Creep Under Complex Stress Systems at Elevated Temperatures", Proc. Instn. Mech. Engrs., Vol. 164, p. 432 (1951).

(6.43) Finnie, I., "Steady Creep of a Tube Under Combined Bending and Internal Pressure", Trans. ASME, Vol. 82D, p. 462 (1960).

REFERENCES (contd.)


(6.44) Saunders, H. E. and Windenburg, D. F., "Strength of Thin Cylindrical Shells Under External Pressure", ASME Trans., Vol. 53, Paper No. APM-53-17a (1931).

(6.45) Windenburg, D. F. and Trilling, C., "Collapse by Instability of Thin-Cylindrical Shells Under External Pressure", ASME Trans., Vol. 56, p. 819 (1934).

(6.46) Windenburg, D. F., "Vessels Under External Pressure-Theoretical and Experimental Equations Represented in Rules for the Construction of Unfired Pressure Vessels Subjected to External Loading", Mechanical Engineering, Vol. 59, p. 601 (1937).

(6.47) Sturm, R. G., "A Study of the Collapsing Pressure of Thin-Walled Cylinders", University of Illinois Engineering Experimental Station, Bulletin No. 329, 1941.

(6.48) Bergman, E. O. "The New-Type Code Chart for the Design of Vessels Under External Pressure", ASME Trans., Vol. 74 (1952).

(6.49) Holt, M., "A Procedure for Determining the Allowable Out-of-Roundness for Vessels Under External Pressure", ASME Trans., Vol. 74 (1952).

(6.50) Haringx, J. A., "Instability of Thin-Walled Cylinders Subjected to Internal Pressure", Philips Research Reports, Vol. 7, pp. 112-118 (1952).

(6.51) Leven, M. M., "Photoelastic Determination of the Stresses in Reinforced Openings in Pressure Vessels", Welding Research Council Bulletin No. 113, April, 1966.

(6.52) Leven, M. M., "Photoelastic Determination of Stresses Due to the Bending of Thin Cylindrical Nozzles in Thin Spherical Vessels", Westinghouse Research Laboratories Report No. 63-917-514-R2, April 15, 1963.

(6.53) Rodabaugh, E. C., Atterbury, T. J., and Hinueber, G. L., "Experimental Evaluation of Simulated, Uncased Pipeline Crossings of Railroads and Highways", Battelle Memorial Institute, Columbus, Ohio, April 15, 1968.

(6.54) Kilpi, J., "The Effect of Out-of-Roundness in a Pressure Vessel on the Stresses and Changes of Shape, with Special Reference to a Cellulose Digester, V.D.I., Vol. 101, Nos. 11 & 14, pp. 421, 568 (1959).

(6.55) Marin, J., Mechanical Behavior of Engineering Materials, Prentice-Hall, Inc., Englewood Cliffs, N. J.

REFERENCES (contd.)


(6.56) Cook, G. and Robertson, A., "The Strength of Thick, Hollow Cylinders Under Internal Pressure", Engineering, Vol. 92, p. 786 (1911).

(6.57) Griffis, L. V., Morikawa, G. K. and Fraenkel, S. J., "Tests on Flow and Fracture of Welded and Unwelded Tubes of Steel", Welding Research Supplement, April, 1948, p. 161-s.

(6.58) Faupel, J. H., and Furbeck, A. R., "Influence of Residual Stress on Behavior of Thick-Walled Closed-End Cylinders", Trans. ASME, Vol. 75, p. 345 (1953).

(6.59) Deffet, L. and Gelbgras, J., "Le Compartment des Tubes a Parois Epaisses Soumis a des Pressions Elevees", Rev. Universelle Mines, Vol. 9, p. 725 (1953).

(6.60) Crossland, B. and Bones, J. A., "The Ultimate Strength of Thick-Walled Cylinders Subjected to Internal Pressure", Engineering, Vol. 179, p. 80, 114 (1955).

(6.61) Clark, J. W. and Woodburn, W. A., Discussion of Reference (6.23).

(6.62) Crossland, B. and Bones, J. A., "Behavior of Thick-Walled Cylinders Subjected to Internal Pressure", Proc. Instn. Mech. Engrs., Vol. 172 (1958).

(6.63) Marin, J. and Sharma, M. G., "Design of a Thin-Walled Cylindrical Pressure Vessel Based Upon the Plastic Range and Considering Anisotropy", Welding Research Council Bulletin No. 40, May, 1958.

(6.64) Crossland, B., Jorgensen, S. M. and Bones, J. A., "The Strength of Thick-Walled Cylinders", Trans. ASME, J. of Engrg. for Industry, May, 1959, p. 95.

(6.65) Marin, J. and Weng, T., "Strength of Aluminum Alloy 6061-T4 Thick-Walled Cylindrical Vessels Subjected to Internal Pressures", Welding Research Council Bulletin No. 58, March, 1960.

(6.66) Wellinger, K. and Uebing, D., "Festigkeitsuerhalten disckwandiger Hohlzylinder Unter Innerdruck im vollplastischen Bereich", M.D.V., Vol. 66, June, 1960.

(6.67) Marin, J. and Weng, T., "Strength of Thick-Walled Cylindrical Vessels Under Internal Pressure for Three Steels", Welding Research Council Bulletin No. 67, March, 1961.

(6.68) Crossland, B., "The Design of Thick-Walled Closed-Ended Cylinders Based on Torsion Data", Welding Research Council Bulletin No. 94, February, 1964.

REFERENCES (contd.)


(6.69) Jones, B. H., and Mellor, P. B., "Plastic Flow and Instability Behavior of Thin-Walled Cylinders Subjected to Constant-Ratio Tensile Stresses", J. of Strain Analysis, Vol. 2, p. 62 (1967)

(6.70) Finnie, I., "An Experimental Study of Multiaxial Creep in Tubes", Joint International Conference on Creep, 1963, Published by Instn. Mech. Engrs., London.

(6.71) Johnson, A. E., Henderson, J. and Khan, B., "Multiaxial Creep-Strain/Complex-Stress/Time Relations for Metallic Alloys With Some Applications to Structures", Ibid.

(6.72) Kennedy, C. R., Harms, W. O. and Douglas, D. A., "Multiaxial Creep Studies on Inconel at 1500 F", Trans. ASME, Vol. 82, Series D, p. 599 (1961).

(6.73) Clark, C. L., "Properties of Ferrous Metals at Elevated Temperatures as Determined by Short-Time Tensile and Expansion Tests", Trans. ASME, Vol. 50, p. 263 (1928).

(6.74) Clark, C. L., and White, A. E., "The Stability of Metals at Elevated Temperatures", Trans. ASME, Vol. 51, p. 670 (1929).

(6.75) Van Duzen, R. M. and McCutchan, A., "High-Temperature-Steam Experience at Detroit", Trans. ASME, Vol. 61, p. 383.

(6.76) Norton, F. H., "Progress Report on Tubular Creep Tests", Trans. ASME, Vol. 63, p. 239 (1941).

(6.77) Kooistra, L. F., Blaser, R. U., and Tucker, J. T.,"High-Temperature Stress-Rupture Testing of Tubular Specimens", Trans. ASME, Vol. 74, p. 783 (1952).

(6.78) Tucker, J. T., Coulter, E. E. and Kooistra, L. F., "Effect of Wall Thickness on Stress-Rupture Life of Tubular Specimens", Trans. ASME, Vol. 82, Series D, p. 465 (1960).

(6.79) Baker, R. A., and Soldan, H. M., "Service Experiences at 1050 F and 1100 F of Piping of Austenitic Steels", Joint International Conference on Creep, 1963, Published by Instn. Mech. Engrs., London.

(6.80) Wyatt, L. M., and Gemmill, M.G., "Experience with Power Generating Steam Plant and Its Bearing on Future Developments", Ibid.

(6.81) Harlow, J. H., "Metallurgical Experience with the Eddystone 5000 lb/in$^2$ 1200 F Unit No. 1", Ibid.

(6.82) Brister, P. M. and Bressler, M. N., "Long-Time Experience with Steel and Alloy Superheater Tubes in Power Boiler Service", Ibid.

REFERENCES (contd.)


(6.83) Marin J., "Strength of Steel Subjected to Biaxial Fatigue Stresses", Welding Research Supplement, November, 1942, p 554s.

(6.84) Morikawa,G. K. and Griffis, L., "The Biaxial Fatigue Strength of Low-Carbon Steels", Welding Research Supplement, March, 1945, p 167s.

(6.85) Majors, H., Mills, B. D., and MacGregor, C. W., "Fatigue Under Combined Pulsating Stresses", J. of Appl. Mechs., Vol 72, p 269 (1949).

(6.86) Bundy, R. W. and Marin, J., "Fatigue Strength of 14S-T4 Aluminum Alloy Subjected to Biaxial Stresses", Proc. ASTM, Vol 54, p 775 (1954).

(6.87) Ruiz, C., "High-Strain Fatigue of Stainless-Steel Cylinders: Experimental Results and Their Applications to Pressure-Vessel Design", J. of Strain Analysis, Vol 2, p 290 (1967).

(6.88) Morrison, J.L.M., Crossland, B., and Parry, J.S.C., "The Strength of Thick Cylinders Subjected to Repeated Internal Pressure", Proc. Instn. Mach. Engrs., Vol 174, No. 2, p 95 (1960).

(6.89) Parry, J.S.C., "Fatigue of Thick Cylinders: Further Practical Information", Proc. Instn. Mech. Engrs., Vol 180, pt. 1, No. 16 p 387 (1965-66).

(6.90) Hannon, B. M., "Correlation of Fatigue Tests of Thick-Wall Cylinders Subjected to Repeated Internal Pressure", Trans. ASME, Series D, Vol 87, p 405 (1965).

(6.91) Dubuc, J. and Welter, G., "Investigation of Static and Fatigue Resistance of Model Pressure Vessels", Welding Research Supplement, July, 1956, p. 329s.

(6.92) Welter, G. and Dubuc, J., "Fatigue Resistance of Simulated Nozzles in Model Pressure Vessels of T-1 Steel", Welding Research Supplement, August, 1962, p 368s.

(6.93) Rodabaugh, E. C. and George, H. H., "Design and Strength of Welded Pipe Line Branch Connections", Proc. ASCE, J. of Pipeline Division, Vol 83, p PL1 (1957).

(6.94) Pickett, A. G. and Grigory, S. C., "Cyclic Pressure Tests of Full Size Pressure Vessels", Southwest Research Institute, San Antonio, Texas, Sept. (1966); Welding Research Council Bulletin 135, Nov. 1968.

(6.95) Pickett, A. G., et al, "Studies of the Fatigue Strength of Pressure Vessels", Progress Reports 1 - 24, January, 1963 through December, 1967, Southwest Research Institute, San Antonio, Texas.

REFERENCES (contd.)

(6.96) Markl, A.R.C., "Fatigue Tests on Piping Components", Trans ASME, Vol 74, p 402 (1951)

(6.97) Newman, R. P., "The Influence of Weld Faults on Fatigue Strength with Reference to Butt Joints in Pipe Lines", Trans. Institute of Marine Engineers, Vol 68, p 153 (1956).

(6.98) O'Toole, W. G. and Rodabaugh, E. C., "Fatigue Tests on 4" Std. Wt. Carbon and Stainless Steel Pipe and Circumferential Butt Weld Joints", Tube Turns, Louisville, Ky., Report No. 8.031, October, 1961.

(6.99) Blair, J. S., "Reinforcement of Branch Pieces", Engineering, Vol 162, pp 1, 217, 508, 529, 553, 577, 588, 605 (1946).

(6.100) Meister, R. P., et al, "Effects of Weld-Joint Imperfections on Piping Performance", Battelle Memorial Institute, Columbus, Ohio, July 31, 1964.

"Effect of Multiple Weld Repairs and 400 F Interpass Temperature on Piping Performance", Battelle Memorial Institute, Columbus, Ohio, October 30, 1964.

(Reports to Dept. of Navy, Portsmouth Naval Shipyard)

(6.101) Dawes, M. G., "Fatigue Strength of Tubes Butt Welded Using a Ceramic Coated Backing Member", British Welding Journal, June, 1967, p 304.

(6.102) Newman, R. P., "Effect on Fatigue Strength of Internal Defects in Welded Joints", British Welding Journal, February, 1959, p 59.

(6.103) Newman, R. P. and Gurney, T. R., "Fatigue Tests of Plain Plate Specimens and Transverse Butt Welds in Mild Steel", British Welding Journal, December, 1959, p 569.

(6.104) Newman, R. P., "Fatigue Strength of Butt Welds in Mild Steel", British Welding Journal, March, 1960, p 169.

(6.105) Gunn, K. W. and McLester, R., "Effect of Mean Stress in Fatigue Properties of Aluminum Alloy Butt-Welded Joints", British Welding Journal, March, 1960, p 201.

(6.106) Gilde, W., "Increasing the Fatigue Strength of Butt-Welded Joints", British Welding Journal, March, 1960, p 208.

(6.107) Tomlinson, J. E. and Wood, J. L., "Factors Influencing the Fatigue Behavior of Welded Aluminum", British Welding Journal, April, 1950, p 250.

(6.108) Stallmeyer, J. E. and Munse, W. H., "Fatigue of Welded Joints in High-Strength Steels", British Welding Journal, April, 1950, p 281.

REFERENCES (contd.)

(6.109) Wood, J. L., "Flexural Fatigue Strength of Butt Welds in N.P. 5/6 Type Aluminium Alloy", British Welding Journal, May, 1960, p 365.

(6.110) Gurney, T. R., "Fatigue Tests in Butt and Fillet Welded Joints in Mild and High Tensile Structural Steels", British Welding Journal, November, 1962, p 614.

(6.111) Dinsdale, W. O., "Effect of Reinforcement Shape on Fatigue Behavior of Butt Welds in N P 5/6", British Welding Journal, May, 1964, p 233.

(6.112) Newman, R. P. and Gurney, T. R., "Fatigue Tests on 1/2" Thick Transverse Butt Welds Containing Slag Inclusions: 1st Interim Report, British Welding Journal, July, 1964, p 341.

(6.113) Gurney, T. R., "Effect of Stress Relief on Fatigue Strength of 1/2" Thick Transverse Butt Welds Containing Slag Inclusions", British Welding Journal, November, 1964, p 593.

(6.114) Gurney, T. R. and Smith, G. C., "Fatigue Tests on 1-1/2" Thick Transverse Butt Welds Containing Slag Inclusions", British Welding Journal, January, 1967, p 17.

(6.115) Harrison, J. D. and Gurney, T. R., "Some Fatigue Tests on 1-1/2" Thick Butt Welds Containing Continuous Slag Lines", Brisith Welding Journal, March, 1967, p 121.

(6.116) Frost, N. E. and Denton, K., "The Fatigue Strength of Butt-Welded Joints in Low-Alloy Structural Steels", British Welding Journal, April, 1967, p 157.

(6.117) Peters, R. W., and Kuhn, P., "Bursting Strength of Unstiffened Pressure Cylinders with Slits", NACA TN 3993, 1957.

(6.118) Anderson, R. B., and Sullivan, T. L., "Fracture Mechanics of Through-Cracked Cylindrical Pressure Vessels", NACA TN D-3252, 1966.

(6.119) Nichols, R. W., Irvine, W. H., Quirk, A., and Bevitt, E., "A Limit Approach to the Presentation of Pressure Vessel Failure", Proc. First Int. Conf. on Fracture, Sendai, Japan, 1673, 1966.

(6.120) Kihara, H., Ikeda, K., and Iwanga, H., "Brittle Fracture Initiation of Line Pipe", Document X-371-66, Presented at Int. Inst. Welding, Delft, 1966.

(6.121) Crichlow, W. J., and Wells, R. H., "Crack Propagation and Residual Static Strength of Fatigue-Cracked Titanium and Steel Cylinders", ASTM-STP 415, 25, 1967.

(6.122) Duffy, A. R., "Studies of Hydrostatic Test Levels and Defect Behavior", Symposium on Line Pipe Research, A.G.A., Dallas, Texas, (Nov. 17-18, 1965).

REFERENCES (contd.)


(6.123) Getz, D. L., Pierce, W. S., and Calvert, H., "Correlation of Uniaxial Notch Tensile Data with Pressure-Vessel Fracture Characteristics", ASME, 1964.

(6.124) Sechler, E. E., and Williams, M. L.,"The Critical Crack Length in Pressurized Monocoque Cylinders", Graduate Aero, Lab. Cal. Inst. Tech. Report, Sept., 1959.

(6.125) Almond, E. A., Petch, N. J., Wraith, A. E., and Wright, E. S., "The Fracture of Pressurized Laminated Cylinders" (to be published).

(6.126) Duffy, A. R., McClure, G. M., Eiber, R. J., and Maxey, W. A., "Fracture Design Practices for Pressure Piping", Chapter to Microscopic and Macroscopic Aspects of Brittle Fracture, to be published by Office of Naval Research.

(6.127) Eiber, R. J., Maxey, W. A., Duffy, A. R., and Atterbury, T. J., "Investigation of the Initiation and Extent of Ductile Pipe Rupture", Quarterly progress Reports, BMI-1793, BMI-1813, BMI-1817, BMI-1836, BMI-1847 (May 1966 to June 1968).

(6.128) Lake, R. L., DeMoney, F. W., and Eiber, R. J., "Performance of 7039 Aluminum Alloys in Vessel Burst Tests and Notched Panel Tests at -220 F. Proc. 1st Int. Congress on LNG, Chicago, Ill., April 7-12, 1968; sponsored by Institute of Gas Technology.

(6.129) General Electric Co., APED, San Jose, California, Reactor Primary Coolant Rupture Study, Quarterly Progress Reports Nos. 1-12, April 1965 to March 1968.

(6.130) Irvine, W. H., Quirk, A., and Bevitt, E., "An Elastic-Plastic Approach to Fracture Mechanics", AHSB(S) R142, United Kingdom Atomic Energy Authority, 1968.

(6.131) "The Effect of Through Crack Size on the Initiation of Fracture in Line Pipe", prepared by AISI Line Pipe Research Committee, presented at AIME Metalworking Conference, Dec. 6, 1966.

(6.132) Hahn, G. T., Sarrate, M., and Rosenfield, A. R., "Criteria for Crack Extension in Cylindrical Pressure Vessels", International Journal of Fracture Mechanics, Vol 5, No 3, Sept., 1969.

(6.133) Sechler, E. E., and Williams, M. L., "The Critical Crack Length in Pressurized Monocoque Cylinders", Final Report on Contract NAw-6525 (Galcit 96), Sept. 1969, Guggenheim Aeronautical Lab., Calif. Inst. Tech., Pasadena, Calif.

(6.134) Eiber, R. J., Maxey, W. A., Duffy, A. R., and McClure, G. M., "Behavior of Through-Wall and Surface Flaws in Cylindrical Vessels", Paper presented at National Symposium on Fracture Mechanics, June 17-19, 1968, Lehigh University.

REFERENCES (contd.)

(6.135) Symposium on Line Pipe Research, March, 1966, Available from the American Gas Association, 605 Third Avenue, New York, New York, 10016, A.G.A. Catalogue No. L30000, pp 83-118.

(6.136) Pellini, W. S., "Advances in Fracture Toughness Characterization Procedures and in Quantitative Interpretation to Fracture-Safe Design for Structural Steels", Welding Research Council Bulletin No. 130, May, 1968.

(6.137) Study of Feasibility of Basing Natural Gas Pipeline Operating Pressures on Hydrostatic Test Pressure, February, 1968, Available from American Gas Association, 605 Third Avenue, New York, New York, 10016, A.G.A. Catalog No. L30500.

(6.138) Nichols, R. W., "The Use of Overstressing Techniques to Reduce the Risk of Subsequent Brittle Fracture", British Welding Journal, Part I - January 1968; Part II - February, 1968.

(6.139) Langer, B. F., "PVRC Interpretive Report of Pressure Vessel Research, Section 1 - Design Considerations", Welding Research Council Bulletin 95 (April, 1964).

(6.140) "A Study of Instability Analysis Methods as Related to LMFBR Piping Systems", Technical Report 240, United Nuclear Corp., Research and Engineering Center, Elmsford, New York.

(6.141) "NASA Shell Analysis Manual", Prepared by North American Aviation Space and Information Systems Division under Contract NAS 9-4387, June 27, 1966.

(6.142) Lee, W. T., "Biaxial Stress-Rupture Properties of Austenitic Stainless Steels in Static Sodium", AI-AEC-12694, June, 1968, Contract AT(04-3)-701.

(6.143) Davis, E. A., "Creep Rupture Tests for the Design of High Temperature Steam Equipment", ASME Journal of Basic Engineering, pp 453-461, June, 1960.

(6.144) Thielsch, H., "Defects and Failures in Pressure Vessels and Piping", Reinhold Publishing Corp., New York, 1965.

(6.145) Hahn, F. P., "Failures of Header Support Bracket Assemblies", Materials Technology in Steam Reforming Processes, Pergammon Press, pp 249-253 (1966).

(6.146) Melworm, R. F., and Berman, I., "Welded Attachments to Tubes - Experimentation, Analysis and Design", Welding Research Supplement, Oct. 1966, pp 454s-464s.

REFERENCES (contd.)

(6.147) Roark, R. J., "Formulas for Stress and Strain", McGraw-Hill Book Company, New York, 4th ed., 1965.

(6.148) Hoff, H. J., Kempner, J., and Pohle, F. V., "Line Load Applied Along Generations of Cylindrical Shells", Quarterly of Applied Mathematics, Vol 11, 1954, pp 411-425.

(6.149) Cooper, R. M., "Cylindrical Shells Under Line Load", J. of Applied Mechanics, Dec. 1957, pp 553-558.

(6.150) Bijlaard, P. P.

"Stresses from Radial Loads in Cylindrical Pressure Vessels", The Welding Journal Research Supplement, December, 1954.

"Stresses from Local Loadings in Cylindrical Pressure Vessels", Trans. ASME, vol. 77, p 805 (1955).

"Stresses from Radial Loads and External Moments in Cylindrical Pressure Vessels", The Welding Journal Research Supplement, p 608s, December, 1955.

"Computation of the Stresses from Local Loads in Spherical Pressure Vessels or Pressure Vessel Heads", Welding Research Council Bulletin No. 34 (March, 1957).

"Local Stresses in Spherical Shells from Radial or Moment Loadings", The Welding Journal Research Supplement, p 240s, May, 1957.

"Stresses in a Spherical Vessel from Radial Loads Acting on a Pipe"; "Stresses in a Spherical Vessel from External Moments Acting on a Pipe"; "Influence of a Reinforcing Pad on the Stresses in a Spherical Vessel Under Local Loading", Welding Research Council Bulletin No. 49 (April, 1959).

"Stresses in Spherical Vessels from Local Loads Transferred by a Pipe"; "Additional Data on Stresses in Cylindrical Shells Under Local Loading".

(6.151) Wichman, K. R., Hopper, A. G., and Mershon, J. L., "Local Stresses in Spherical and Cylindrical Shells Due to External Loadings", Welding Research Council Bulletin No. 107, August, 1965.

(6.152) Melworm, R. F., Patel, P. D., and Berman, I., "Shell Theory Solution for Asymmetric Balanced Radial Loads on Long Cylinders", J. Engineering for Power, Trans. ASME, April, 1958, pp 177-185. "Solutions for Distributed Loads on Long Cylinders", J. Engineering for Industry, Trans. ASME, August, 1969, pp 623-635.

(6.153) Mehringer, F. J., and Cooper, W. E., "Experimental Determinations of Stresses in the Vicinity of Pipe Appendages to a Cylindrical Shell", Proc. Society Experimental Stress Analysis, Vol. 14, No. 2, pp 159-174.

REFERENCES (contd.)

(6.154) Cranch, E. T., "An Experimental Investigation of Stresses in the Neighborhood of Attachments to a Cylindrical Shell", Welding Research Council Bulletin No. 60, May, 1960.

CHAPTER 7

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# 7. CURVED PIPE AND MITERS

The term "curved pipe" as used herein is intended to cover both shop or field bent pipe and welding elbows manufactured to meet butt-welded fitting specifications such as USAS B16.9 or B16.28. Analytically, such components may be considered as sections of an annular torus. They are basically defined by a bend radius R, a cross section radius r, wall thickness t, and arc angle $\alpha$; as shown in Figure 7.1.

Miters consist of segments of straight pipe, cut at a miter angle $\beta$ and welded to produce the desired direction change. Miters are basically defined by a miter angle $\beta$, miter spacing s, cross section radius r, and wall thickness t, as shown in Figure 7.2. For certain combinations of $\beta$ and s, the miter bend approaches the configuration of a curved pipe; theories and tests for curved pipe provide some guidance to the characteristics of such miters.

## 7.1 Internal Pressure Loading

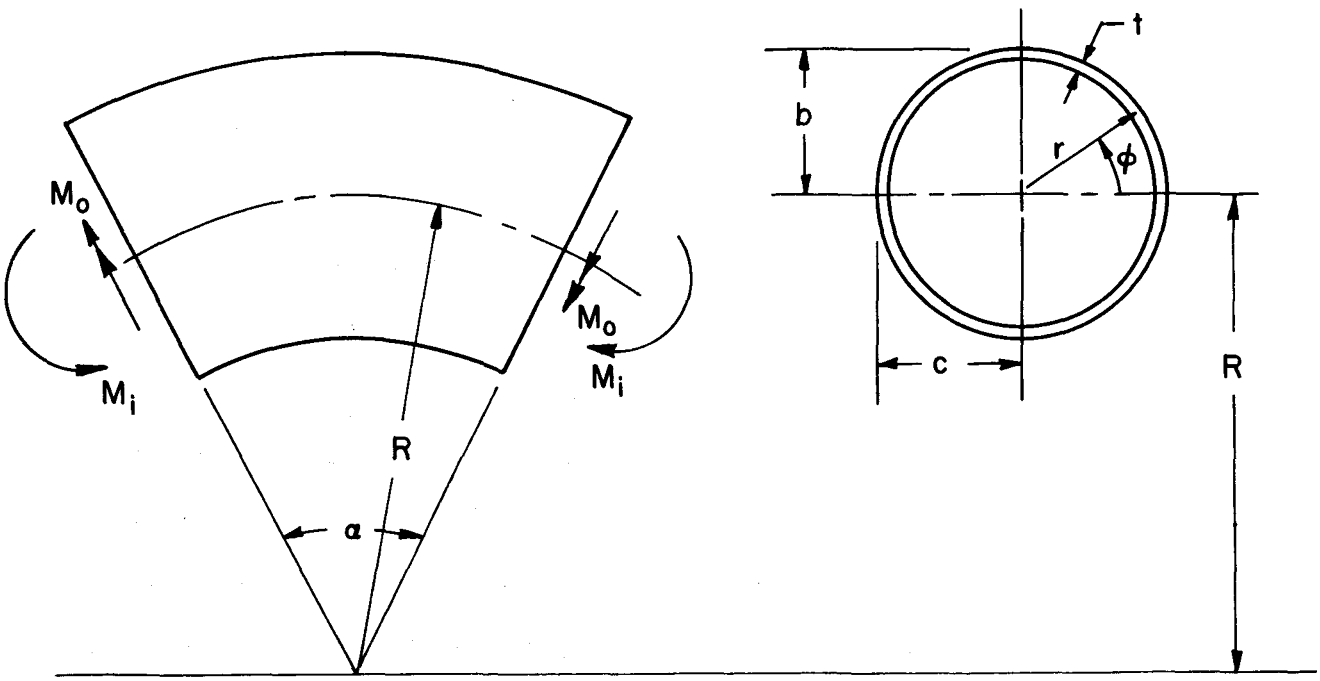
### 7.11 Theory - Curved Pipe, Circular Cross Section

The membrane stresses in a section of a thin-wall, circular cross section torus are given by the equations:

$$\sigma_{\varphi m} = \frac{rP}{t}\left[\frac{1 + 0.5\ (r/R)\ \sin\varphi}{1 + (r/R)\ \sin\varphi}\right] \tag{7.1}$$

$$\sigma_{\alpha m} = \frac{rP}{2t} \tag{7.2}$$

where P = internal pressure and $\sigma_{\varphi m}$ $\sigma_{\alpha m}$, r, R, t, and $\varphi$ are defined in




$\sigma_{\phi m}$ = membrane stress in $\phi$-direction; hoop or transverse stress

$\sigma_{\phi b}$ = bending stress in $\phi$-direction

$\sigma_{\alpha m}$ = membrane stress in $\alpha$-direction; axial or longitudinal stress

$\sigma_{\alpha b}$ = bending stress in $\alpha$-direction

b = ellipse radius in plane of the bend

c = ellipse radius normal to plane of the bend

h = $tR/r^2$

$\mu$ = $\sqrt{12(1 - \nu^2)}\ r^2/Rt$

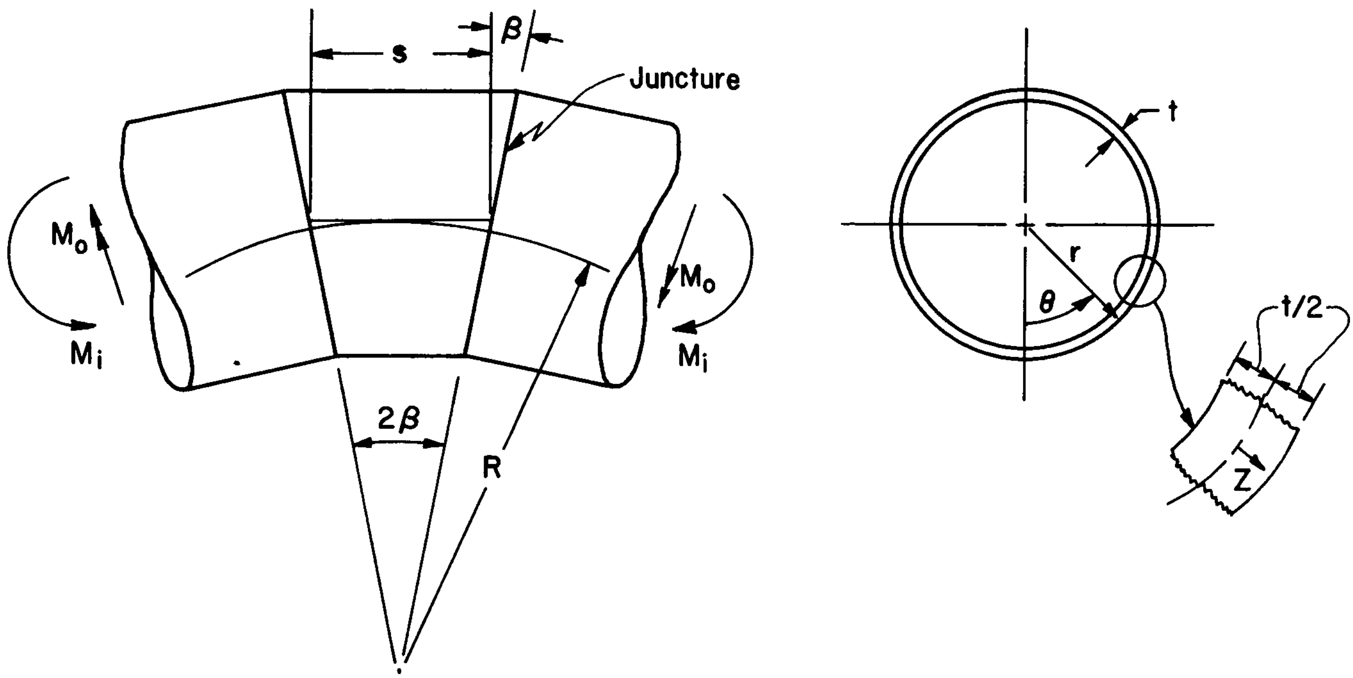
$\nu$ = Poisson's ratio

E = modulus of elasticity

$M_i$ = in-plane bending moment

$M_o$ = out-of-plane bending moment

FIGURE 7.1: NOMENCLATURE FOR CURVED PIPE




$\sigma_{\theta m}$ = membrane stress in θ-direction; hoop or transverse stress

$\sigma_{\theta b}$ = bending stress in θ-direction

$\sigma_{\beta m}$ = membrane stress in β-direction

$\sigma_{\beta b}$ = bending stress in β-direction

Single miter = miter with only one juncture

Multiple miter = miter with more than one juncture

Segments = number of straight pipe pieces in miter bend

= 3 in above sketch

$s = 2R \tan \beta$ for multiple miters

FIGURE 7.2: NOMENCLATURE FOR MITERS

Figure 7.1. Equations (7.1) and (7.2) were first published* by Lorenz[7.1] in 1910. The factor in brackets in Equation (7.1) is sometimes referred to as the "Lorenz factor" and will be so referred to herein. (It is frequently called the "Lorenz effect".) The derivation of these equations is shown by Timoshenko[7.2].

The $\sigma_{\alpha m}$ - stress, as shown by Equation (7.2), is the same as the axial stress in straight pipe with the same r and t. The value of $\sigma_{\varphi m}$ depends upon both r/R and $\varphi$. The maximum occurs at $\varphi = -\pi/2$ and is:

$$(\sigma_{\varphi m})_{max} = \frac{rP}{t}\left[\frac{1 - 0.5\ (r/R)}{1 - (r/R)}\right] \quad . \tag{7.3}$$

For r/R = 3 (roughly corresponding to USAS B16.9 elbows), the maximum value of $\sigma_{\varphi m}$ is 1.25 times the hoop stress in straight pipe with the same r and t.

Bending stresses in a section of a thin-wall, circular cross-section torus, in which the ends of the torus section are free to deform, are zero. This result is given by Clark, Gilroy, and Reissner[7.3], Equation (110) therein.

The derivation of Equations (7.1) and (7.2) involve the usual assumptions of thin-wall shell, linear-elastic theory. In addition, the analysis is applicable to a section of a torus, without consideration of the restraints imposed by attachments to the ends of the torus section. In piping application, the curved pipe (torus section) terminates at straight pipe attached to the ends. The hoop stress in straight pipe is different than that of the torus section, hence a discontinuity exists

---

* Lorenz developed equations for an elliptical cross-section torus, such as represented by a Bourdon tube. They simplify to Equations (7.1) and (7.2) when the ratio of the major and minor axis of the ellipse is equal to unity; i.e., a circular cross section.

at the curved pipe to straight pipe junction. At this time, an acceptable and generally used theoretical analysis for these "end effects" does not exist. The theoretical analysis by Kalnins[7.4] may prove useful in this respect.

## 7.12 Theory - Curved Pipe, Non-Circular Cross Section

Curved pipe used in piping systems usually do not have a circular cross section. For such components, application of internal pressure can cause significant bending stresses. For components with a cross section describable as an ellipse having a major or minor axis in the plane of the bend, the theory developed by Clark, Gilroy, and Reissner[7.3] is pertinent. The maximum bending stress is given by:

$$(\sigma_{\varphi b})_{max} = \left(\frac{Pr}{t}\right) \times \frac{0.814}{\sqrt{1-\nu^2}} \frac{R}{r} \left(1 - \frac{b^2}{c^2}\right) \mu^{2/3} \qquad (7.4)$$

where

$\mu = \sqrt{12\ (1 - \nu^2)}\ r^2/Rt$

$\nu$ = Poisson's ratio

b and c are ellipse radii as shown in Figure 7.1.

Equation (7.4) is an asymptotic solution to the differential equations, valid for $\mu$ greater than about 10 and for b about equal to c. To illustrate the significance of the bending stress, consider the following example.

$R/r = 3.0$, $b = 0.99\ r$, $c = 1.01r$, $r = (b + c)/2$

$r/t = 10$, $\nu = 0.3$.

From Equation (7.4)

$$(\sigma_{\varphi b})_{max} = 0.51\ (Pr/t) \quad .$$

The out-of-roundness of ± 1%, for r/t = 10, produces a maximum bending stress of about one-half of the nominal stress in straight pipe. The out-of-roundness effect becomes even more significant as r/t increases. If, in the above example, r/t = 50, then

$$(\sigma_{\varphi b})_{max} = 1.48\ (Pr/t) \quad .$$

Equation (7.4) is based on linear elastic theory. As discussed in Chapter 6, for straight pipe the linear theory can be quite conservative for large values of r/t. An analogous nonlinear theory for curved pipe is not available; in its absence the nonlinear theory for straight pipe may serve as a design basis for curved pipe, provided that the linear-theory for curved pipe is about the same as for straight pipe. Reference (7.3) also gives an equation which compares the bending stress in curved pipe with the bending stress in straight pipe. The equation, for b/c close to unity and $\mu$ greater than about 10, is:

$$\frac{(\sigma_{\varphi b})_{max}}{(\sigma_{hb})_{max}} = \frac{1.87}{\mu^{1/3}} \tag{7.5}$$

where $(\sigma_{hb})_{max}$ = maximum bending stress in a straight pipe with the same cross section as the curved pipe.

For values of $\mu$ less than about 10, Equation (7.5) is not valid. Reference (7.3) gives series-form solutions* which are valid for small

* Actually, the series-form solutions are valid for all values of $\mu$, however, for large values of $\mu$, many terms in the series must be retained for adequate convergence. The asymptotic solutions are therefore much easier to use for large values of $\mu$.

values of $\mu$. The series-form solutions shows that as $\mu \longrightarrow 0$, the value of $(\sigma_{\varphi b})_{max}/(\sigma_{hb})_{max} \longrightarrow 1.00$. For a value of $\mu$ of 50 (representing about the highest value of $\mu$ encountered in curved pipe in piping systems), the value of $(\sigma_{\varphi b})_{max}/(\sigma_{hb})_{max}$ is 0.51. Accordingly, the bending stress in elliptical cross section curved pipe is not greater than the bending stress in elliptical cross section straight pipe; on the other hand it is not much less than in straight pipe.

One other aspect of the theory for internal pressure applied to an out-of-round curved pipe should be mentioned. When internal pressure is applied to such curved pipe, there will be a rotation of one end of the curved pipe with respect to the other end if one or both ends are free. If both ends are fixed, a moment will develop at the fixed ends. For values of $\mu$ greater than about 10 and for $b/c \approx 1$, the value of the moment is given in Reference (7.3) as:

$$M_p = \pi(1 - \frac{b^2}{c^2})\ (1 - 2/\mu)\ Rrt\ (\frac{Pr}{t}) \tag{7.6}$$

where $M_p$ = end moment due to internal pressure with the ends fixed.

The value of $M_p$ can be expressed in terms of nominal bending stress in straight pipe as:

$$S'_p = \frac{M_p}{Z} = \frac{M_p}{\pi r^2 t} = (1 - \frac{b^2}{c^2})\ \frac{R}{r}\ (1 - 2/\mu)\ (\frac{Pr}{t}) \quad . \tag{7.7}$$

Expressed in the form of Equation (7.7), it may be seen that for curved pipe not more than ± 1% out-of-round, the equivalent bending stress $S_p$ will not be more than about 10 or 12% of the nominal hoop stress, $Pr/t$. However, for a large, closely coupled piping system attached to load-sensitive equipment, the moment produced by internal pressure may not be negligible as shown by the following example.

$R = 45"$; $r = 15"$; $t = 1.5"$; $b = 0.99\ r$; $c = 1.01\ r$

Internal pressure is such that $Pr/t = 10{,}000$ psi.

From Equation (7.6):

$$M_p = \pi\ [1 - (.98)^2]\ (1 - 2/11)\ (45 \times 15 \times 1.5) \times 10{,}000$$

$$M_p = 1.04 \times 10^6 \text{ in-lb.}$$

The value of $M_p = 1.04 \times 10^6$ in-lb may be compared with a moment of $9 \times 10^4$ in-lb permitted by NEMA Standard No. SM 20-1958[7.5] on steam inlet, extraction or exhaust connections of steam turbines. The moment $M_p$ will, of course, be reduced by flexibility of the piping system; in addition the theory is linear, and nonlinear affects would reduce the moment. However, the calculations indicate that the pressure-generated moment in an out-of-round elbow may not always be negligible. The writer knows of one incident in which problems with a large centrifugal gas compressor probably arose from this effect.

The preceding theory is limited to the elastic small deformation regime, ignoring end effects. Apparently no theory exists as to the characteristics of curved pipe in the plastic and/or large deformation regime. Test data on burst strength discussed later herein indicate that, prior to occurence of a limit pressure as defined by rupture, large deformations and "end effects" play a significant role.

In field or shop bending of straight pipe into curved pipe, an increase in wall thickness near $\varphi = -\pi/2$ and decrease in wall thickness near $\varphi = \pi/2$ usually occurs. Weil, Brock, and Cooper[7.6] give equations for calculating wall thickness changes as a result of the bending process and corresponding equations for membrane stresses in the curved pipe of the resulting variable thickness. An interesting result is that the maximum membrane hoop stress, which occurs at $\varphi = -\pi/2$ for a uniform wall curved pipe, moves to the location $\varphi = \pi/2$, with its value given by:

$$(\sigma_{\varphi m})\ \max = \frac{Pr}{t_o}\left(1 + \frac{r/R}{2}\right) \tag{7.8}$$

where $t_o$ = initial (assumed uniform) wall thickness of the straight pipe.

7.13 Theory - Miters

The theory for miters with internal pressure loading is not completely developed. The single miter problem was approached by Van der Neut(7.7) and Murthy(7.8) starting with thin-shell equations. Green and Emmerson(7.9) developed an analogous theory starting with the equations of three-dimensional elasticity. Both References (7.7) and (7.9) arrive at an anomalous result; i.e., the longitudinal stresses at large distances from the miter juncture are given by the equation:

$$(\sigma_{\beta m}) = \frac{Pr}{2t}\left(1 + \tan^2 \beta \cos 2\theta\right) \tag{7.9}$$

where P = internal pressure. Other symbols are defined in Figure 7.2. One would expect that, at large distances from the miter juncture, the axial membrane stress would be just $Pr/2t$. Murthy(7.8) recognized the same problem but immediately limited his analysis to small values of $\beta$ such that $\tan^2 \beta$ is assumed negligible compared to unity. Green and Emmerson(7.9) also suggest that their development may be restricted to small values of the miter angle $\beta$; just how small is not established.

The results of interest in the present review are given by the following equations from Green and Emmerson for stresses at the mitered juncture.

$$(\sigma_{\theta m}) = \frac{Pr}{t}\left[1 + \frac{k\epsilon}{2}\cos\theta\right] \tag{7.10}$$

$$(\sigma_{\beta m}) = \frac{Pr}{2t} \tag{7.11}$$

$$(\sigma_{\beta b}) = \frac{Pr}{t}\left[\frac{k^3\epsilon\rho}{1-\nu^2}\cos\theta\right] \tag{7.12}$$

$$(\sigma_{\theta b}) = \nu\,(\sigma_{\alpha b}) \tag{7.13}$$

where

$P$ = internal pressure

$r$ = pipe radius

$t$ = pipe wall thickness

$k$ = $[0.75\,(1 - \nu^2)]^{1/4}$

$\epsilon$ = $\tan\beta/\lambda^{1/2}$

$\lambda$ = $t/2r$

$\rho$ = $z/(t/2)$

$z$ = variable through wall thickness, see Figure 7.2

$\rho$ = 1 at outside surface

$\rho$ = 0 at midsurface

$\rho$ = -1 at inside surface

$\nu$ = Poisson's ratio

$\theta$ = location angle as shown on Figure 7.2

Some comparisons of Equations (7.10) through (7.13) with test data are given later herein. It should be noted that the theory is supposed to be applicable only to single miters or widely spaced miters, i.e., where the miter spacing is sufficiently large so that deformations at one juncture do not extend to the adjacent juncture.

There are theoretical developments applicable to a "reinforced" miter; i.e., a miter in which the juncture is attached to a rib. In one approach to this structure, it is assumed that the membrane forces due to pressure in the pipe act undisturbed up to the rib. The analysis then gives stresses in the reinforcing rib but not in the shell segments. This approach is developed by Appleyard[7.10] and by Mackenzie and Beattie[7.11]. Another approach is to assume that the reinforcing rib is infinitely rigid, the analysis gives stresses in the shell at the juncture of the shell with the rib. These theories are developed by Kornecki[7.12], Estrin[7.13], and Corum[7.14], the last being more general in that in addition to internal pressure loading, the effect of moments or forces applied to the pipe remote from the oblique section are considered. Owen and Emmerson[7.15] also give the development of the theory for the clamped-juncture miter along with an alternate derivation of the theory given by Green and Emmerson[7.9].

## 7.2 Internal Pressure, Test Data

### 7.21 Test Data - Curved Pipe

#### 7.211 Elastic Stresses

Published test data on measured elastic stresses (e.g. as determined by strain gages) in curved pipe with internal pressure loading are quite limited.

The following test data represent results obtained by placing strain gages around the circumference of the curved pipe at a section midway between the ends. This gives stresses as a function of $\varphi$ at the center of the curved pipe. No significant test data are available for stresses as a function of $\alpha$; i.e., how the pipe or other closures attached to the ends of the curved pipe influences the stresses.

Experimental results for a curved pipe with $R/r = 9.6$ are given by de Leiris and Barthelemy(7.16). Good agreement between experimental and theoretical results are shown, however, the Lorenz effect is rather small and the comparison between test and theory deals mainly with the effect of ovality and variations in wall thickness.

Gross(7.17) gives test results for 6" - Sch. 80 welding elbows (R = 9", r = 3.172", t = 0.280", nominal dimensions). These results at least roughly confirm Equations (7.1) and (7.2), although irregularities in the cross section shape and wall thickness variations in the test model introduce uncertainties in the comparisons.

D. R. Zeno(7.18) gives test data on a curved pipe with R = 5", r = 2.57", t = .408", and $\alpha$ = 90 degrees. Strain gages were placed on the outside surface only. The Lorenz effect is quite significant for this test model, the membrane hoop stress at $\varphi = -\frac{\pi}{2}$ being about 1.5 times as high as the hoop stress in equivalent straight pipe. The test results confirmed this prediction, as well as the general form of variations of $\sigma_{\varphi m}$ as a function of $\varphi$. The longitudinal membrane stress, expected to be independent of $\varphi$ by Equation (7.2), was found to vary appreciably. No mention is made of thickness variations or ovality of the cross section; it may be speculated that a major part of differences between theory and test results arose from these aspects.

Rodabaugh, Melnick, and Atterbury(7.19) tested elbows with mean dimensions as follows:

| Model No. | R | r | t | $\alpha$ |
|---|---|---|---|---|
| 1 | 11.81 | 4.02 | .497 | 90° |
| 2 | 12.27 | 6.02 | .622 | 90° |
| 3 | 12.34 | 4.05 | .484 | 45° |
| 4 | 18.54 | 6.10 | .494 | 45° |

These tests were run to determine the flexibility and stress of the curved pipe subjected to moment, and combinations of pressure and moment loads. However, the test data includes (not published) measured strains due to internal pressure. These data for test models 1, 2, and 3 roughly confirm Equation (7.1), although ovality effects are significant. In model 4, ovality effects predominate over the Lorenz effect.

The preceding comprises the known, non-proprietary data on elastic stresses in curved pipe and welding elbows with internal pressure loading. In summary, the test data:

(1) Roughly confirms Equations (7.1) and (7.2) for membrane stresses.

(2) Indicate the significance of out-of-roundness or cross-sectional shape irregularities but do not give any quantitative information.

(3) Do not give any useful information on the significance of "end effects".

7.212 Cyclic Pressure Fatigue Tests

Lane(7.20) gives the results of cyclic internal pressure fatigue tests on a series of 7 - 6" Sch. 80 welding elbows (R = 9", r = 3.172", t = 0.280", nominal dimensions). The pressure was varied from 100 psi up to an upper pressure limit ranging from 1800 to 5000 psi. The nominal stress (Pr/t, but based on average measured dimensions of r and t) ranged from 1070 psi at the lower pressure to an upper stress limit ranging from 19,300 psi to 53,500 psi. The Lorenz factor at $\varphi = -\pi/2$ is about 1.28 for these elbows. All of the test specimens failed by a longitudinal fracture on the inner arc of the bend; as would be expected from Equation (7.1). An S-N curve is given in Reference 7.20 for this test series.

The results of the tests may be compared with the Nuclear Piping Code (USAS B31.7) analysis for cyclic operation as follows. From Figure 9.21 of Reference (7.20), the stress intensity amplitude for failure in $10^5$ cycles is:

$$\sigma_a = \frac{60,300 - 1,500}{2} = 29,400 \text{ psi}$$

The mean stress during the cycle is:

$$\sigma_m = \frac{60,300 + 1,500}{2} = 30,900$$

The values of both $\sigma_a$ and $\sigma_m$ include the Lorenz factor of 1.28 because failures occurred in the inner arc of the bend. The stress intensity as defined by Section III is taken as the hoop stress plus the internal pressure.

The equivalent completely reversed stress can be obtained by the modified Goodman diagram equation:

$$S_{eq} = \frac{\sigma_a}{1 - (\sigma_m/S_u)} \tag{7.14}$$

where

$S_u$ = ultimate tensile strength

= 75,000 psi for the test specimens

Hence:

$$S_{eq} = \frac{29,400}{1 - \frac{30,900}{75,000}} = 50,000 \text{ psi}$$

The value of $S_{eq}$ may now be compared with an appropriate S-N curve for the material. The design curves in USAS B31.7 cannot be used directly, since they include safety factors on stress and/or life. However, the S-N curve given in Reference (7.21), Figure 9 for carbon steels, is appropriate. The S-N curve gives S = 50,000 psi at N = $10^5$ cycles. The exact agreement between $S_{eq}$ and S from Reference (7.21) is no doubt a coincidence. However, the correlation may be taken as evidence of the validity of Equation (7.1), although other conditions such as surface finish and ovality may have had some affect on the results.

It may be noted that at the highest pressure test (5,000 psi), the calculated stress intensity, including the Lorenz factor, is 73,500 psi. This is well above the reported yield strength of the material of 46,000 psi. The elbow with this maximum pressure lasted for 36,500 cycles. There is no mention in Reference (7.20) of any deformation of the elbow during the fatigue tests; presumably such deformation was sufficiently small so that it was not noticeable.

## 7.213 Burst Tests

Published results of burst tests, in which pressure is increased until rupture occurs, are relatively limited. Gross[7.17] gives test results for 7 elbows: however, only one of those 7 elbows can be considered as typical of USAS B16.9 welding elbows. That test elbow (Experiment No. 15 in Reference 7.17) was a 6" Sch. 80; from the same batch as the elbows used in the cyclic pressure fatigue tests discussed above. The test elbow had a burst pressure of between 7100 and 7200 psi. This may be compared with a computed bursting pressure $P_c$ of:

$$P_c = \frac{S_u t}{r_m} = \frac{78,500 \times .290}{3.18} = 7150 \text{ psi}^*$$

The significant point in this comparison is that the actual burst pressure is given by $P_c$, not $P_c/1.28 = 5580$ psi which would be calculated if it were assumed that the Lorenz factor applies in the plastic region.

In a discussion of Reference (7.17), Blair (7.22) commented concerning a number of burst tests he had conducted. The test models consisted of 90 degree, 180 degree, and 360 degree (complete torus) welding elbows. At least some of the test models had an R/r ratio of 3.0. No information is given as to the values of r or t for the test models. Blair states that in twenty-four tests carried out, bursting occured at an average pressure equal to 1.04 times the (calculated?) burst pressure of the straight pipe.

A recent report by Rodabaugh, Duffy, and Atterbury (7.62) summarizes available data on experimentally determined yield pressure and burst pressure of B16.9 elbows. This includes some 15 tests performed by manufacturers of elbows.

While the Lorenz factor represents a membrane stress, there are two reasons why the burst pressure may not be reduced significantly as compared to the burst pressure of a straight pipe.

(1) Before the burst pressure is reached, the elbow shape changes significantly. In particular, the inner arc of the bend straightens or bulges out.

(2) During large plastic deformations, part of the load is transferred from the inner arc of the elbow to the straight pipe attached to the elbow.

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* $S_u$ = reported ultimate tensile strength of material in normalized heat treat condition, t = wall thickness at failure location, $r_m$ = mean radius (before test) of elbow cross section.

The preceding comments should also be taken as a warning against conditions in which it may not be safe to ignore the Lorenz affect in estimating the burst pressure of curved pipe or elbows; i.e.,

(1) Material of low ductility

(2) Curved pipe or elbows in which the inner arc length is relatively long; possibly relative to the parameter $\sqrt{rt}$. This would imply that the Lorenz factor is more likely to be significant for either:

(a) Large values of $\alpha$,

(b) Large values of r/t, or

(c) Large values of R/r*.

While the above summarizes the known, non-proprietary test data on burst pressures, it is pertinent to consider the bursting strength requirement given in USAS Standard B16.9, "Wrought Steel Buttwelding Fittings". This standard includes "long radius" elbows in which R/r $\approx$ 3**. An identical strength requirement is given in USAS Standard B16.28, "Wrought Steel Buttwelding Short Radius Elbows and Returns", which includes elbows with R/r $\approx$ 2. The bursting strength requirement is quoted below in its entirety because there are certain subtle but significant implications in the precise wording used.

"8. Bursting Strength.

The actual bursting pressure of the fittings covered by this standard shall at least equal the computed bursting pressure of seamless pipe of the schedule number (or nominal wall thickness) and material designated by the marking on the

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* This item may be self-compensating in the sense that as R/r increases, the inner arc length increases but the Lorenz factor itself decreases.

** Sizes 2" and larger.

fitting. To determine the bursting pressure of the fittings, straight seamless pipe of the designated schedule (or nominal wall thickness) and material shall be welded to each end; each pipe being at least equal in length to twice the outside diameter of the pipe and having proper end closures, applied beyond the minimum length of straight pipe; hydrostatic pressure shall be applied until either the fitting or one of the pipes welded thereto bursts.

"The computed bursting pressure of the seamless pipe, with which the actual bursting pressure of fittings shall be compared, shall be determined by the following formula:

$$P = \frac{2St}{D}$$

where:

P = bursting pressure of pipe, psi

S = minimum specified tensile strength of pipe or of material of an equivalent grade, psi

t = minimum pipe wall thickness, inches. For the purpose of this formula t is defined as 87-1/2 percent of the nominal thickness of the pipe for which the fitting is recommended for use.

D = specified outside diameter of pipe, inches."

"Since the above formula is applicable only to straight pipe, it cannot be used for a direct computation of the bursting pressure of fittings. Their ability to withstand bursting pressures shall be gaged only by comparing their behavior on test with the calculated bursting pressure of straight seamless pipe of the designated wall thickness and material."

The implication of the above bursting strength requirement is that ach manufacturer of welding elbows sold to USAS B16.9 or USAS B16.28 must omehow obtain assurance his products meet the bursting strength requirement. e might do this by running a series of prototype tests on his elbows, covering the range of dimensional parameters and materials that he sells. At east three major manufacturers of elbows have run such prototype tests and resumably so have others. One may postulate the existence of a considerable olume of test data indicating that the burst pressure of elbows is essentially the same as that of straight pipe. However, one should note that in alculating the required minimum burst pressure P, the value of S is the inimum specified tensile strength and the value of t is the minimum wall thickness. In testing a series of prototypes it would be unlikely that the manufacturer could or would select elbows with minimum wall thickness t and material with minimum tensile strength S. A series of prototype tests of "typical" elbows would only indicate that the burst pressure of elbows is not so much less than that of straight pipe, and that it is not compensated for by typical as compared to minimum thickness (particularly in the inner arc area) and by typical as compared to minimum material tensile strength.

The previous discussion considers data and burst-test requirements for welding elbows manufacturered to a standard such as USAS B16.9. Curved pipe may also be produced by a shop- or field-bending process applied to straight pipe. In general, such bending processes result in a thinning of the backwall and thickening of the crotch-wall. The only test data found on such bends are given by Feltz and Phillips[7.63]. These were tests on cold-formed pipe bends, sizes 3/4 through 4 inches, bent to a radius ration (R/r) of from about 8 to 9.6. The material was API 5L, Grade I. The 3/4, 1-1/4, and 2-inch sizes were standard weight wall; the 3 and 4-inch were 0.188-inch wall.

The significant results of the tests were that all burst ruptures were located in the straight pipe tangents to the curved pipe segment; not in the bent portion of the pipe. Feltz and Phillips attribute this to the strengthening of the material in the bent section by cold working.

## 7.22 Test Data - Miters

### 7.221 Elastic Stresses

Internal pressure test data for single miters, such that Equations (7.10) through (7.13) would be applicable thereto, are given by Owen and Emmerson[(7.15)]. Carefully machined models were made of an epoxy casting resin. Stresses were determined by using the stress-freezing technique of photoelasticity. Eight models were tested, with r = 2", t = 0.1" and 0.2", and $\beta$ = 15, 30, 37-1/2, and 45 degrees. The test results agree quite well with Equation (7.10) for the membrane hoop stress at the junction for all values of $\beta$ included in the test models. For other stresses, agreement is good for $\beta$ = 15 degrees, but for larger values of $\beta$, the theory appears to overestimate the bending stresses.

Mackenzie and Beattie[(7.11)] report results of internal pressure tests on a steel unreinforced single miter with r = 39.4", t = 1.375", and $\beta$ = 45 degrees. Stresses were determined by use of strain gages. Again, the test data agrees well with Equation (7.10) for the membrane hoop stress, but bending stresses are grossly overestimated.

Lane and Rose[(7.23)] report results of internal pressure tests on 3- and 4- segment miter bends with r = 6.09", t = .37", and $\beta$ = 30 degree (3 segment miter bend), $\beta$ = 15 degrees (4- segment miter bend). The miter spacing was made so that the miter bends simulate a curved pipe with $\alpha = 90°$, $R = 18''$, $R/r \cong 3$. Maximum values of membrane hoop stresses were found at the junctures. These are somewhat lower than predicted by Equation (7.10). The maximum measured hoop membrane stress indices were about 1.8 and 1.3 for the 3-segment and 4-segment bends, respectively. These may be compared with $(\sigma_{\varphi m})max = 1.25$ for the equivalent curved pipe with $R/r = 3$. Bending stresses were much smaller than predicted by Equations (7.12) or (7.13); however, the authors point out that their measured stresses probably underestimate the actual maximum stresses at the juncture.

### 7.222 Cyclic Pressure Fatigue Tests

Macfarlane[7.24] gives the results of cyclic pressure fatigue tests on five 3-segment miter bends with $r = 3.19$, $t = .278$, and $\beta = 22\text{-}1/2$ degrees. The combinations of $\beta$ and s used were such that the miter bends simulate a curved pipe with $\alpha = 90°$, $R = 9''$, $R/r \cong 3$. All five specimens failed by a crack across and transverse to a junction weld at $\theta$ between 11 degrees and 22 degrees. Theoretically, maximum stresses occur at $\theta = 0$. The maximum hoop stress index, by Equation (7.10), is 1.89. Macfarlane also ran cyclic pressure fatigue tests on straight pipe from the same lot of pipe as was used for making the miter bends. By comparing the S-N curve for the miter bends with the S-N curve for the straight pipe, a fatigue stress intensification factor of about 1.3 is obtained. This is considerably lower than the 1.89 hoop stress intensity obtained by Equation (7.10). A possible reason for the discrepancy is that the pipe itself was reported to have a poor surface finish. If the straight pipe is assigned a stress intensification factor of 1.3 or 1.4, then the fatigue tests would agree better with Equation (7.10) and with the measured stresses given by Lane and Rose[7.23] for a similar 3-segment bend. It should be noted that the direction of the fatigue cracks indicates high hoop stresses and indicates that the high bending stresses given by Equation (7.12) did not exist in the cyclic pressure fatigue test specimens.

#### 7.223 Burst Tests

Lane and Rose[7.23] give results of burst tests on 3-segment and 4-segment miters with $r = 6.18''$, $t \cong 0.37''$. The burst pressures were about 81% (3-segment miter) and 99% (4-segment miter) of the calculated burst pressure of equivalent straight pipe.

## 7.3 Moment Loading, Theory

### 7.31 Theory - Curved Pipe or Welding Elbows

#### 7.311 Elastic Characteristics

That a curved pipe subjected to a moment loading behaves differently than a curved solid bar was noted experimentally by Bantlin[7.25] in 1910. Because of the ability of the pipe cross section to deform, as shown in Figure 7.3, a curved pipe is more flexible than a curved bar (of the same moment of inertia); for the same reason high bending stresses can develop in the hoop-direction. These characteristics have since been identified by use of a flexibility factor K and a stress index i, defined as follows:

$$K = \frac{\theta_{ab}}{\frac{R}{EI}\int_o^\alpha M(d\alpha)} \qquad (7.15)^*$$

* It is assumed here that R, E, and I are constant over the arc length $\alpha$, M may vary along the arc length.

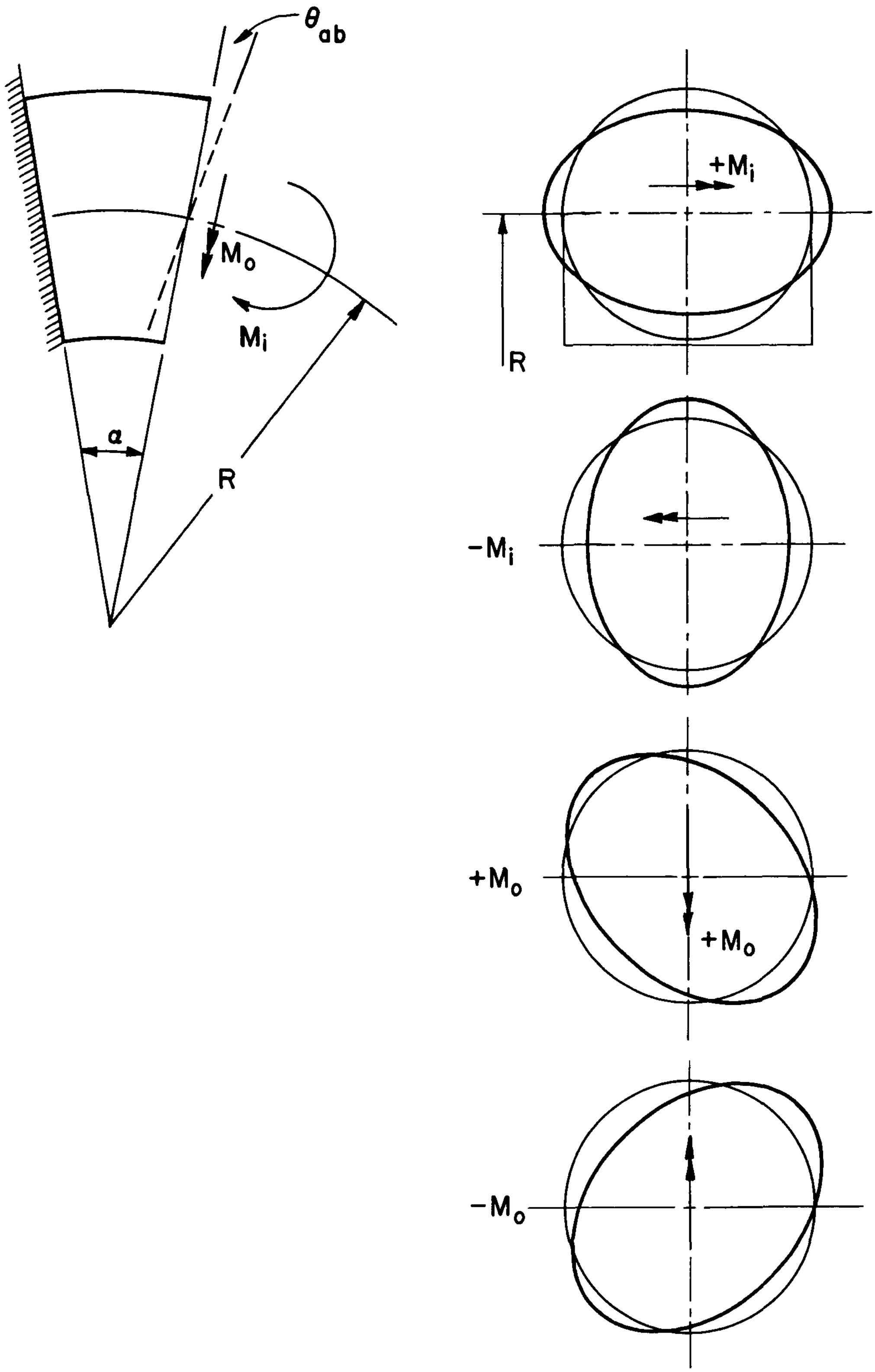


FIGURE 7.3 DEFORMATION OF CURVED PIPE CROSS SECTION UNDER BENDING MOMENTS

where $\theta_{ab}$ = rotation of end a with respect to end b of the curved pipe as shown by Figure 7.3.

R = bend radius

E = Modulus of elasticity

I = moment of inertia of pipe cross section

M = applied moment

$\alpha$ = curved pipe arc length

$$i = \frac{(\sigma_{\varphi b})\ max}{M/Z} \tag{7.16}$$

where $(\sigma_{\varphi b})_{max}$ = maximum bending hoop stress

Z = section modulus of curved pipe cross section

In 1911, Th. von Karman[7.26] published a theoretical analysis of the characteristics of curved pipe subjected to "in-plane" bending moments. (See Figure 7.1 for definition of in-plane moment.) A strain energy method which leads to a series solution was used. He gave only the first term in the series solution, which results* in following expressions for K and i:

$$K = \frac{12h^2 + 10}{12h^2 + 1} \tag{7.17}$$

$$i = \frac{18\ h}{12h^2 + 1} \tag{7.18}$$

where $h = tR/r^2$

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* In the development of the various theories for bending of curved pipe, an inconsistency occurs in some of the results with respect to the anticlastic behavior of the shell in the hoop direction. In Equation (7.17), h is more accurately defined as $tR/r^2 \sqrt{1-\nu^2}$. Also some papers on the subject give longitudinal stresses at the mid wall only; this has been misinterpreted as applying to the longitudinal surface stresses. Because of the anticlastic hoop bending, the longitudinal surface stresses are $\sigma_{\alpha m} \pm \nu\ \sigma_{\varphi b}$.

During the period from 1911 to 1943 several authors (7.27 through 7.30) arrived at essentially the same solutions as given by von Karman. During this period, numerous tests were run on pipe bends with both in-plane and out-of-plane bending. In at least one[7.31] of the reports on these tests, it was observed that curved pipe flexibility for out-of-plane bending was also higher than anticipated by curved beam theory. However, it was not until 1943 that Vigness[7.32] gave the development of a theory for out-of-plane moments. Vigness gave specific results for the first term of the series solution; the K- and i- factors for the first-term approximation are the same as given in Equations (7.17) and (7.18).

The first-term approximations of von Karman for in-plane bending, and Vigness for out-of-plane bending were sufficiently accurate for relatively heavy-wall pipe bends with large bend radii. However, with the increasing use of welding elbows having relatively thin walls, it became more apparent that the first-term approximations given by Equations (7.17) and (7.18) grossly underestimated both the flexibility and stress intensification present in curved pipe or welding elbows with small values of the parameter $h = tR/r^2$.

Shipman[7.33], in 1929, showed the value of K using the 1st and 2nd term of the series solution. Jenks[7.34], in a discussion of the paper by Shipman, gives equations for calculating flexibility factors and stress indices for all values of h. This was based on an "nth approximation" of von Karman's series solutions. Karl[7.35], also refined von Karman's analysis for in-plane bending by retaining more terms in the series solution.

In 1945 Beskin[7.36], again using a strain energy approach, extended both von Karman's and Vigness' analyses (in-plane and out-of-plane) bending, respectively) to include sufficient terms in the series solution so that the truncation error was less than 1 percent. Beskin plotted his results of a function of h, thereby showing that for values of h less than about 0.3, the value of the flexibility factor and stress intensification factors are given by the simple equations:

$$K = \frac{1.65}{h^{2/3}} \qquad (7.19)$$

$$i_i = \frac{1.89}{h^{2/3}} \qquad (7.20)$$

$$i_o = \frac{1.59}{h^{2/3}} \qquad (7.21)$$

where $i_i$ = stress index for in-plane bending

$i_o$ = stress index for out-of-plane bending

Clark and Reissner(7.37), in 1951, obtained solutions to the in-plane bending problem from the standpoint of the differential equations of shell theory. For their approximation consisting of one β-term, and two φ-terms, the resulting K- and i- factors are almost the same as those for von Karman's first-term approximation. Clark and Reissner show and discuss higher order approximations of their solutions, along with a general series solution for the flexibility factor. They then proceed to obtain an asymptotic solution for the differential equations. Their results for the K and i- factors are identical to Equations (7.19) and (7.20).

Clark and Reissner also investigated in-plane bending of a curved tube with elliptical cross section; with an important implication with respect to curved pipe or welding elbows. The asymptotic equations for the K and i-factors are:

$$K = \frac{4J(e)}{\pi} \frac{\sqrt{3(1-\nu^2)}c^2}{Rt} \qquad (7.22)$$

$$i = \frac{0.813}{\sqrt{1-\nu^2}} \frac{4J(e)}{\pi} \frac{u^{2/3}}{\sqrt{1-e^2}} \qquad (7.23)$$

where

$e = 1 - (b/c)^2$

b = ellipse semi-axis in plane of bend

c = ellipse semi-axis normal to plane of bend

$u = \sqrt{12(1-\nu^2)}\ bc/Rt$

J (e) = function of e given by

$$3e^2 J(e) = (1 + e^2)\ E(e) - (1 - e^2)\ I(e)$$

E(e) = complete elliptic integral of first kind

I(e) = complete elliptic integral of second kind

Considering curved pipe or elbows with ± 1 percent out-of-roundness, the value of b/c will range from 0.98 to 1.02 and the value of e will range from +.04 to -.04. The function J(e) is equal to $\pi/4$ at e = 0 and is within one percent of $\pi/4$ at e in the range from +.04 to -.04. Accordingly; Equations (7.22) and (7.23) show that the flexibility factor and stress index are only slightly changed by a small out-of-roundness of the section.

All of the theories discussed up to this point have one thing in common; i.e., they assume that $R/r \gg 1$. The validity of the application of such theories to welding elbows with R/r = 3 or less was questioned. Symonds and Pardue[(7.38)] developed* the theory for both in-plane and out-of-plane bending without the assumption that $R/r \gg 1$. Numerical comparisons show that the flexibility factor and stress index** obtained from the more refined analysis (R/r not assumed $\gg 1$) are within 5 percent of those obtained from the previously discussed theories.

Another aspect not included in theories discussed up to this point is that of the membrane hoop stress. Clark and Reissner[(7.37)] give values of

* This analysis is given in condensed form by Pardue and Vigness[(7.39)].

** The maximum value of the longitudinal stress is more affected by the R/r assumption, being of the order of 20 percent higher for R/r = 3 by Symonds and Pardue's analysis.

this membrane hoop stress for in-plane bending, both by a series solution and an asymptotic solution. The series solution, for "one β-term and two Ψ-terms retained", is

$$\frac{\sigma_{\varphi m}}{(M_i r/I} = - \left(\frac{r}{R}\right) \frac{\cos \varphi}{1 + (r/R) \sin \varphi} \left[\cos \varphi + \frac{1}{13.2h^2 + 1} \cos 3 \varphi\right] \qquad (7.24)$$

The asymptotic solution, valid for h < about 0.3, is:

$$\frac{(\sigma_{\varphi m}) \max}{(M_i r/I)} = \frac{r}{R} \cdot \frac{0.96}{h^{1/3}} \qquad (7.25)$$

where

$\sigma_{\varphi m}$ = membrane hoop stress

$(\sigma_{\varphi m})_{max}$ = maximum membrane hoop stress, at $\varphi = 0$.

Equations (7.20) and (7.25), valid for $h < 0.3$, give the surface stresses at $\varphi = 0$, i.e.:

$$\frac{\sigma_{\varphi}}{(M_i r/I} = - \frac{r}{R} \times \frac{0.96}{h^{1/3}} \pm \frac{1.89}{h^{2/3}} \qquad (7.26)$$

where $M_i$ has the direction shown in Figure 7.1, and the + part of the ± sign refers to the outside surface, - part to the inside surface.

Gross[(7.17)] also gives the in-plane theory for the membrane hoop stress along with explicit equations* for one, two, and three-term approximations for its calculation. The first-term approximation given by Gross is almost the same as Equation (7.24).

The inclusion of the direct stress places the maximum stress (for values of $h < 1.0$) on the inside surface at approximately $\varphi = 0$ or 180°. This corresponds to the location of the initiation of in-plane bending fatigue failures found by Markl[(7.40, 7.41)].

* There appears to be an error in the expression for the three-term approximation.

An analogous membrane hoop stress exists for out-of-plane bending. The theoretical equations for its evaluation are given by Rodabaugh[7.42]. For out-of-plane bending, the maximum membrane stress appears at $\varphi = \pi/4$, $5\pi/4$, $9\pi/4$, and $11\pi/4$. Its magnitude, in comparison to the maximum hoop bending stress, is less significant than for the in-plane bending case.

Turner and Ford[7.43] reviewed the various theories for in-plane bending of curved pipe. They listed the major assumptions and approximations as follows:

(1) $R \gg r$

(Symonds and Pardue[7.38] did not make this assumption.)

(2) Longitudinal strains constant through the wall, implying $t/r \ll 1$.

(3) Hoop stresses are due to bending only.

(Both Clark and Reissner[7.37] and Gross[7.17] included the membrane hoop stresses.)

(4) Hoop strains are due to bending only.

(See footnote on p. 24)

(5) Hoop bending stresses are distributed linearly through the wall thickness.

(6) In some cases, incomplete analysis leads to inconsistencies of a term $(1-\nu^2)$, and in evaluation of surface stresses.

(See footnote on p 24)

(7) Where series solutions have been used, insufficient terms have been retained for accuracy at small values of h.

(Jenks[7.34] and Beskin[7.36] covered this aspect.)

(8a) The shear stresses $\tau_{r\alpha}$ are neglected, and the solution is independent of the bend angle.

(This assumption is discussed subsequently herein.)

(8b) The shear stresses $\tau_{r\varphi}$ are neglected.

(Clark and Reissner[7.37] and Gross[7.17], in evaluating the direct hoop stress, include these.)

(9) All radial stresses are neglected.

(This assumption, with respect to internal pressure, is discussed subsequently herein.)

Turner and Ford developed a theory in which approximations 1 through 7 are not made. Assumption 8 in part and assumption 9 are still made. Turner and Ford made a number of numerical comparisons and, while theoretically the simplier theories can lead to cumulative errors, they concluded that:

> "Considered as an engineering problem, however, the range of practical pipe sizes, and the positioning of the peak stress values lead to the conclusion that the simpler theories (von Karman 1911; Beskin 1945) modified to allow for the transverse direct stress (Gross 1952-53) and taken to an adequate number of terms, are sufficiently accurate. The assumptions cause modifications which, by a combination of circumstances, do not affect the peak values of the stresses significantly; the cumulative effects are never more than 5-10 percent on the peak equivalent stresses."

In 1967 Smith[7.44] developed a theory for out-of-plane bending using the same basic assumptions as made by Turner and Ford for in-plane bending. Smith gives flexibility factors based on his analysis which are within 8 percent of those given by Equation (7.19). Values are also given for stresses, however, they are not in a form suitable for comparison with the simpler analyses. It appears, however, that the conclusion quoted above from Turner and Ford[7.43] for in-plane bending is also applicable to out-of-plane bending.

In a recent paper, Jones[7.45] extends von Karman's analysis but does not assume that $R/r \gg 1.0$. Symonds and Pardue[7.38] had previously developed this theory but Jones gives a more extensive description of the theory for in-plane bending and results obtained thereby. Jones states that the results of his work indicate that stresses in and flexibility of curved pipe bends are virtually independent of $R/r$ and depend almost entirely on the pipe factor $tR/r^2$.

The remainder of this discussion is concerned with the two assumptions made by Turner and Ford[7.43], both of which are quite significant. These assumptions are:

(8a) The shear stresses $\tau_{r\alpha}$ are neglected, and the solution is independent of the bend angle.

(9) All radial stresses are neglected.

The implication of the first assumption is briefly alluded to previously herein as "end effects". Consider, for example, a curved pipe with $R/r = 3$, $r/t = 20$, and $\alpha = 30$ degrees; pipe flanges are welded to both ends. The flexibility and stress increase in curved pipe arises from the flattening of the cross section; but in this example the ends cannot distort since the flanges are very rigid in a radial direction. The arc lengths between flanges are less than $\sqrt{rt}$, hence all points on the elbows will be affected by the flanges. Obviously, the flexibility and stress intensification factors given by the theories are incorrect when applied to such a structure.

As mentioned previously, an accepted and generally used theoretical analysis for these "end effects" does not exist. The theoretical analysis by Kalnins[7.4] may prove useful in this respect. There are, however, a significant amount of test data on end effects which will be discussed later herein.

The second assumption implies that the effect of internal pressure inside the curved pipe is negligible. That this might lead to a significant error was recognized for many years. Wahl[7.29] in 1928 applied a crude approximation to estimate the internal pressure effect. As later developments show, his approximation was qualitatively correct. In 1947 Barthelemy[7.46] published his development of the theory for in-plane bending with internal pressure. This paper apparently escaped attention in this country. Clark, Gilroy, and Reissner[7.3] developed a theory from which the pressure effect could be approximately deduced. Kafka and Dunn[7.47] directly developed the theory for in-plane bending. Rodabaugh and George[7.48] developed the theory for both in-plane and out-of-plane bending and also developed comparatively simple equations with adequate accuracy for purposes of practical piping problems. The theory of Kafka and Dunn is essentially the same as that of Rodabaugh and George for in-plane bending. The internal pressure has a significant practical effect only for small values of the parameter $tR/r^2$. An additional parameter is required; $\psi = PR^2/Ert = SR^2/Er^2$, where S = nominal pressure stress = Pr/t, E = modulus of elasticity. For curved pipe with sufficient internal pressure to give S = 20,000 psi, with R/r = 3.0, $tR/r^2 = 0.1$ and with E = 30,000,000 the flexibility factor is reduced from about 17 to 11, the in-plane stress index from 9 to 5, the out-of-plane stress index from 8 to 4.5 by the internal pressure effect.

### 7.312 Limit Bending Loads

In piping systems where the bending loads arise from imposed displacements, the limit load is not usually significant. However, for weight loads or where a displacement may result in strain concentration at an elbow, the limit load can be significant. No theoretical analysis of this problem has been found. Some test data exist which are discussed later herein.

## 7.32 Theory - Miters

Kitching[7.49] developed a theory for in-plane bending of a multiple segment miter. To obtain the flexibility factor, it was assumed that the deformed shape of the miter segments did not vary along the axis of the segment. The deformed shape was assumed to be given by a Fourier series in $2n\varphi$ ($n = 1,2,3$); an energy method was then used similar to von Karman's[7.26] analysis for curved pipe. To obtain stresses, an edge correction based on a modified "beam-on-elastic foundation" approach was used, making the edge conditions compatible and satisfying equilibrium at the edges.

Jones and Kitching[7.50] developed an analysis for a 2-segment miter bend. The development may be considered in two parts: (1) a strain-energy solution for that part of the cross sectional distortion with long decay length, (2) a shell theory solution for those distortions with short decay length, thus giving the local stresses at the discontinuity formed by the miter juncture.

These appear to be the only published theories on bending of miters. The theories are limited to in-plane bending. The theory developed by Corum[7.14] could be applied to a 2-segment miter with a rigid reinforcement at the juncture.

A semi-empirical equation was developed for miter bends by Markl(7.41), in analogy to his test results for curved pipe. Markl suggests that an equivalent bend radius R' be defined as

$$R' = \frac{s \cdot \cot \beta}{2} \text{ for } s < r\ (1 + \tan \beta) \tag{7.27}$$

$$R' = \frac{r\ (1 + \cot \beta)}{2} \text{ for } s > r\ (1 + \tan \beta) \tag{7.28}$$

The fatigue stress intensification factor* is then given by

$$i_f = \frac{0.9}{(tR'/r^2)^{2/3}} \tag{7.29}$$

and the flexibility factor by

$$K = \frac{1.52}{(tR'/r^2)^{5/6}} \tag{7.30}$$

## 7.4 Moment Loading, Test Data

### 7.41 Test Data, Curved Pipe

#### 7.411 Elastic Characteristics

As mentioned earlier, Bantlin(7.25) in 1910 first noted that the flexibility of curved pipe was significantly greater than that calculated for a curved beam of equivalent moment of inertia. During the subsequent 57 years a large amount of test data on the flexibility of, and stresses in, curved pipe has been accumulated.

Between 1910 and 1950, much of the test work was devoted to determination of the flexibility of curved pipe. Before the advent and general use

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* Markl's fatigue stress intensification factors are related to the fatigue strength of a "typical" girth butt weld in straight pipe taken as unity.

of SR-4 strain gages, the determination of stresses in a curved pipe was not a simple task. However, stresses due to in-plane bending were determined by Hovgaard(7.30) and by Wahl, Bowley, and Back(7.51) as early as 1929. Strains were determined by means of mechanical extensometers such as the Berry or Tuckerman types. Among other contributors to experimental determination of flexibility factors for in-plane bending are Hovgaard(7.30), Shipmen(7.33), Cope and Wirt(7.31) and Wahl(7.29). The question as to the out-of-plane flexibility factor, which most authors still believed to be unity, was raised by the test results of Cope and Wirt(7.31). Mayrose(7.52) gave results of out-of-plane bending tests from which the out-of-plane flexibility factor could have been fairly well established as roughly equal to the in-plane flexibility factor.

These early papers are mostly devoted to methods of making a piping system flexibility analysis (See Chapter 3). The experimental values for flexibility factors of curved pipe, almost all for test models with $tR/r^2$ greater than 0.5, were generally in adequate agreement with Equation (7.17). Relatively little work was done on stresses and none on stresses for out-of-plane bending. The paper by Wahl, Bowley, and Back(7.51) is worthy of note. In-plane bending tests were run on a bend with r = 5.13", t = .48", R = 54", $tR/r^2$ = .985. Both longitudinal and transverse strains were measured. The derived stresses agree reasonably well with von Karman's analysis for the stresses.

The paper by Vigness(7.53) is notable in that it specifically investigates the case of out-of-plane bending. The theory for out-of-plane bending was checked by:

(1) Measurements of displacements of one end of the curved pipe test assembly with respect to the other end. This gives a check of the out-of-plane flexibility factor.

(2) Measurements of the cross-sectional deformation. This gives an indirect check of the circumferential stresses.

(3) Measurements of the longitudinal strains (Tuckerman strain gages).

All of these measurements were compared with the theory and reasonable agreement was found. These tests also give evidence that the flexibility factor and stress index for torsional loading are equal to unity. The values of $tR/r^2$ for the test models were all equal to 0.6 or larger, hence the first-term series approximation of the theory was of adequate accuracy.

The paper by Pardue and Vigness[7.39] is the most significant of the available papers giving test data on the elastic characteristics of curved pipe. This paper gives:

(1) Flexibility factors of curved pipe with in-plane or out-of-plane bending.

(2) Maximum circumferential stresses for in-plane or out-of-plane bending,

(3) Maximum longitudinal stresses for in-plane or out-of-plane bending,

(4) Typical circumferential and longitudinal stresses as a function of $\varphi$.

The test models cover a range of $tR/r^2$ from 0.044 to 0.137. In addition to tests of 90° and 180° arcs with straight pipe attached, test data are also given for flanges attached to one or both ends. The paper is the best source of information on "end effects" at this time.

Companion papers by Gross[7.17] and Gross and Ford[7.54] published in 1953 provide additional test data on in-plane bending of "short-radius" pipe bends, i.e. curved pipe with $R/r \cong 3$. The paper by Gross and Ford gives test data on flexibility factors, cross-section deformations and stresses.

This paper gives the first set of test data in which strain gages were placed on both inside and outside surfaces. The flexibility factors agree adequately with the theory, as do the peak measured stresses. Gross and Ford also give data for combination of internal pressure and in-plane bending. Comparisons of test data with theory(7.48) are tabulated below.

Bend No. 5, R = 9", r = 3.183", t = 0.309", $tR/r^2 = 0.272$

| | P = 0 psi | P = 600 psi | $K_p/K_o$ or $i_p/i_o$ Test | $K_p/K_o$ or $i_p/i_o$ Theory |
|---|---|---|---|---|
| Flexibility Factor, K | 5.84 | 5.23 | .90 | .94 |
| Stress index, i | 3.7 | 3.0 | .81 | .89 |

Vissat and Del Buono(7.55) give test data on 180° arc short-radius ($R/r \cong 3$) elbows. Data are given for flexibility factors and for stresses on the outside surface only. The authors concluded that theories developed on the assumption that $R/r \gg 1$ were not applicable to elbows with R/r of 3. Discussions of the paper disagree with this conclusion. Apparently the authors were unaware of the theoretical work by Symonds and Pardue(7.38) and of the experimental results by Pardue and Vigness. Vissat and Del Buono also present data on 180° bends with both ends flanged; obtaining results roughly comparable to those obtained by Pardue and Vigness(7.39).

In 1956 and 1957, two papers were published on the effect of internal pressure on bending of curved pipe. The first, by Kafka and Dunn(7.47), gives in-plane bending test results on flexibility factors and longitudinal stresses (outside only). The theory and test data are in fairly good agreement; oddly, the largest disagreement occurs for zero pressure. The second paper, by Rodabaugh and George(7.48), gives in-plane bending test results on flexibility factors and a complete stress vs.

$\varphi$-location curve for an elbow with R = 30", r = 15", t = .515". This test data appears to be the first in which the strain gages were small enough to give reasonable assurance of accurate measurement of peak stresses. This is a problem with the transverse stress because it varies rapidly and unless the active strain gage length is less than about 5 percent of the radius, some error can occur due to averaging of stresses over the strain gage length. For a 15"-radius elbow, gages of .75" length could be used; for a 3" radius elbow, used in some previous tests, a strain gage length not over 0.15" would be necessary but probably considerably larger gages were used. The correlation of test data with theory is quite good for both flexibility factors and stresses. The test data shows quite clearly that the inside surface hoop stress is higher than on the outside surface thereby confirming the hoop membrane stresses as predicted by Clark and Reissner[7.37] or Gross[7.17].

Rodabaugh, Melnick, and Atterbury[7.19] give test results for flexibility and stresses in 90° and 45°, 8" and 12" elbows with R/r = 3 and $r/t \sim 10$. The elbows were subjected to both in-plane and out-of-plane bending, with and without internal pressure. The tests were primarily designed to investigate "end effects" for copper-nickel alloy elbows used in submarine piping. Empirical correlations are shown for both flexibility factors and stress indices. It was found that end effects were more significant for out-of-plane bending than for in-plane bending and more significant for 45° elbows than for 90° elbows. The theoretical internal pressure effect was small (less than 10 percent reduction in flexibility factor or stress index); the test results on the average agreed with the theoretically predicted reduction.

Smith and Ford[7.56] give test results for flexibility and stresses in 3 - 90° curved pipes of 6" nominal size with R/r = 5.8, 8.1 and 3.0; r/t = 13, 14, and 3.6. The elbows were subjected to both in-plane and out-of plane bending tests. Both in-plane and out-of-plane flexibility factors were found to be significantly lower than predicted by theory; 80 to 90 percent of theory for in-plane bending and about the same percentage for the major rotation due to out-of-plane bending. Stresses for both in-plane and out-of-plane bending were in reasonably good agreement with theory, except that transverse stresses in the bend with R/r = 3 were somewhat lower than theoretical stresses.

Findlay and Spence[7.57] give test results for no doubt the largest curved pipe ever tested; a piping system with two 90° elbows with r = 39.725", t = 1.45", and R = 117". One elbow was subjected to an in-plane bending load; the measured stresses agree quite well with theories including the transverse mean stress; i.e., Clark and Reisner[7.37] or Gross[7.17]. The authors are not too confident* of their experimental flexibility factor but give a value of 12.8, which is about 83 percent of the theoretical flexibility factor.

In summary of the test data on elastic characteristics of curved pipe:

(1) Test data generally indicate that actual flexibility factors for 90° bends with straight pipe tangents are around 90 percent of the theoretical flexibility factors.

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* It might be remarked that the accurate experimental determination of flexibility factors of curved pipe is not an easy task. Some problems involved are discussed in Reference (7.19).

(2) Stresses for sections of curved pipe which are not influenced by end effects are in fairly good agreement with the theory. The transverse direct stress is fairly well confirmed by test data. The theory which assumes $R/r \gg 1$ is adequate for curved pipe with $R/r = 3$.

(3) End effects are highly significant; more so for out-of-plane bending than for in-plane bending.

### 7.412 Cyclic Bending Fatigue Tests

The "stress intensification factors" given in the USASI Code for Pressure Piping (ASA B31.1-1955) for curved pipe are based on cyclic bending fatigue tests reported by Markl[7.40, 7.41]. The curved pipe tested by Markl was all 4" nominal size, made of carbon steel (A106 Gr. B). Values of R/r ranged from 2 to 10 with $tR/r^2$ from 0.06 to 1.5. Arc angles included in the tests ranged from 15° to 243°. These tests, therefore, provide a rather wide coverage of the dimensional parameters for curved pipe with straight pipe tangents.

Analysis of the test results by Markl led to the equation:

$$i_f S N^{.2} = 245{,}000 \tag{7.31}$$

where

$i_f$ = fatigue stress intensification factor

= 1.0 for a "typical" girth butt weld in straight pipe

$S$ = nominal stress = M/Z, psi

$N$ = cycles-to-failure (crack through wall).

The fatigue stress intensification factor* was found to be quite close to one-half of the theoretical stress index; i.e.:

* For simplicity, the USAS Code uses Equations (7.32) for both in-plane and out-of-plane bending.

$$i_f \cong \frac{i}{2} \cong \frac{0.9}{h^{2/3}} \text{ for in-plane bending} \tag{7.32}$$

$$\cong \frac{0.75}{h^{2/3}} \text{ for out-of-plane bending} \tag{7.33}$$

It should be noted that $i_f$ is based on the fatigue strength of a typical girth butt weld in straight pipe taken as unity; whereas, as compared to polished bar fatigue tests at $10^5$ cycles-to-failure, the fatigue stress intensification factor of such welds is about two. The ratio of $i_f/i \cong 0.5$ rather than unity is due primarily to the girth-butt-weld basis for $i_f$.

The fatigue failures usually occurred at the locations indicated by the theory. For in-plane bending, the maximum stress is circumferential at $\varphi \cong 0$ (Figure 7.1); the fatigue cracks were longitudinal and occurred at about $\varphi = 0$. The theory indicates that stresses on the inside are greater than on outside surface; the fatigue tests indicate that cracks started on the inside surface. For out-of-plane bending, the highest stress is circumferential at $\varphi \cong \pm 45$, $\pm 135$ degrees and fatigue failures generally were at these locations.

The above comparisons are valid for arc angles of 90° and larger. Markl ran fatigue tests to explore the effect of arc angle. He found that for practical purposes the $i_f$-factors for arc angle less than 90° could be approximated by a linear interpolation between unity for $\alpha = 0$ to $i_f = 0.9/h^{2/3}$ for $\alpha = 90°$. The linear interpolation does not hold for small values of $\alpha$; apparently the effect of two closely spaced girth butt welds creates an intensification which overshadows the curvature effect, and gives an intensification greater than caused by an isolated girth butt weld. These results were obtained from tests on 4" nominal size elbows with $t \cong .25"$, $R = 6"$. They should not be extrapolated carelessly to other dimensional parameters. The tests do, however, give an indication of the significance of "end effects" in curved pipe analysis.

Lane[7.20] ran cyclic bending fatigue tests on 6" nominal size carbon steel elbows with R = 9", $t \cong 0.3"$, $\alpha$ = 90 degrees. In-plane bending loads were applied, both with zero internal pressure and with 1500 psi static internal pressure. The fatigue stress intensification factor, as defined by Equation (7.31), obtainable from Lane's data agrees quite closely with the factor obtained by Markl. Lane's data for tests with 1500 psi internal pressure were essentially the same as with zero internal pressure. Theoretically[7.48], the internal pressure would reduce stresses due to bending by about 8 percent. However, the mean stress due to the static internal pressure would about off-set the alternating stress reduction.

Two sets of test data are available for materials other than carbon steel. Markl[7.58] gives results of in-plane bending tests on austenitic stainless steel welding elbows. Rekate[7.59] gives results of in-plane bending tests on 70 - 30 copper-nickel alloy welding elbows. Considering the constant on the right-hand side of Equation (7.31) as a material property, the tests may be compared as follows:

| | |
|---|---|
| Carbon steel: | $i_f SN^{.2} = 245{,}000$ |
| Austenitic Stainless steel: | $i_f SN^{.2} = 281{,}000$ |
| 70-30 copper nickel: | $i_f SN^{.2} = 290{,}000$ |

It appears, therefore, that the constant in Equation (7.31) is not very sensitive to the material. It should be remarked that these tests cover a cycles-to-failure between about 100 and $2 \times 10^6$ cycles.

#### 7.413 Limit Bending Loads

Gross[7.17] and Gross and Ford[7.54] give test data on in-plane moments which cause "collapse" or very large displacements. Tests were run on curved pipe with $R/r = 3$, h from 0.09 to 0.35. The collapse moment, for an in-plane moment closing the bend, ranged from about 1.5 to 2.0 times the calculated yield moment $M_{yc}$, where

$$M_{yc} = \frac{ZS_y}{i} \tag{7.34}$$

where

$Z$ = section modulus of curved pipe cross section

$S_y$ = yield strength of curved pipe material

$i$ = stress index.

One test was reported by Gross and Ford[7.54] on an elbow with $R/r = 2.83$, $tR/r^2 = 2.72$, tested with an internal pressure of 1500 psi. The internal pressure increased the collapse load significantly, the ratio of collapse moment to $M_{yc}$ being 3.0 or larger, as compared to an analogous ratio of about 1.5 for a similar unpressurized curved pipe.

It should be emphasized that these tests were for a moment that closed the bend. Because of the kind of deformations involved, a higher limit bending moment might be expected for a moment that opens the bend.

### 7.42 Test Data, Miters

#### 7.421 Elastic Characteristics

Gross and Ford[7.54] ran in-plane bending tests on a 5-segment miter bend, with $\beta = 11.25°$, $s = 5.32''$, $r = 6.02''$, $t = .133''$. The combination of $\beta$

and s is such that the miter bend was similar to a curved pipe with $\alpha = 90°$, $R/r = 2.2$, and $tR/r^2 = .049$. The flexibility factor of this miter bend, related to the equivalent curved pipe, was found to be 37.8; the theoretical flexibility factor for the equivalent curved pipe is $1.65/h = 33.6$. Stresses were determined at the center section of the miter bend (remote from the juncture welds). The maximum measured stress index was about 10.0; the theoretical stress index for the equivalent curved pipe is $1.89/h^{2/3} = 14.2$. The maximum measured stress was in the hoop direction on the inside surface at $\varphi \cong 0$.

Lane & Rose(7.23) tested 3-segment and 4-segment miter bends with $\beta = 22\text{-}1/2°$ (3-segment) or $\beta = 15°$ (4-segment), s = 14.9" or 9.65", $r \cong 6.18$, $t \cong .37''$. The combinations of $\beta$ and s are such that the miter bends were similar to a curved pipe with $\alpha = 90°$, $R/r = 2.91$, and $tR/r^2 \cong .173$. The flexibility factors found for these miters (in-plane bending) ranged from 7.45 to 8.70, as compared to the theoretical flexibility factor (for the equivalent curved pipe) of about 9.5. A 3-segment miter was tested with 1,000 psi internal pressure; resulting in about 12% decrease in flexibility as compared to the theoretical decrease (for the equivalent curved pipe) of about 15%. The hoop stresses at sections midway between the juncture welds were found to be similar in distribution but smaller than calculated for the equivalent curved pipe. Close to the welds, the hoop stresses are higher than between welds; the measured values of $(\sigma_{\varphi b})_{max}$ were roughly 87% (3-segment) and 123% (4-segment) of the calculated value of $(\sigma_{\varphi b})_{max}$ in the equivalent curved pipe (i-factor = 6.1). At least one 4-segment miter was tested with out-of-plane bending, giving stresses again roughly comparable to those expected from theory for the equivalent curved pipe. No information is given on the flexibility factor for out-of-plane bending.

Kitching[7.60] ran in-plane bending tests on a 5-segment miter with $\beta = 22\text{-}1/2°$, s = 16.6", r = 6.15", t = 0.197". The combination of $\beta$ and s is such that the miter bend was similar to a curved pipe with $\alpha = 180°$, R/r = 3.3, and h = 0.104. The flexibility factor (related to the equivalent curved pipe) was found to be about 14.4 as compared to the theoretical value of 15.85 for the equivalent curved pipe. The maximum hoop stress was found to occur on the inside surface at $\varphi = 0$, at the juncture weld; the i-factor was 11.5 as compared to the theoretical value of 8.55 for the equivalent curved pipe. However, a slightly higher longitudinal stress was found at the juncture weld, around 12.5 times the nominal bending stress, M/Z.

Jones and Kitching[7.61] ran in-plane and out-of-plane bending tests on a two-segment miter with $\beta = 45°$, r = 4.214", t = 0.222". For in-plane bending, the mitered juncture was found to contribute significantly to the flexibility of the test assembly. Expressed in terms of an equivalent curved pipe with R/r = 3.0, the flexibility factor of the single miter is 3.59; the equivalent elbow (with r = 4.214", t = 0.222", R = 12.7", h = .158) is 10.4. The highest measured stress was longitudinal on the outside surface, 0.5" from the juncture* at $\theta = 60°$. This stress ratio (to M/Z) was 6.7; the equivalent elbow with R/r = 3 would have a calculated stress index of 6.5. The maximum measure hoop stress ratio was 5.9; on the inside surface, 0.5" from the juncture* at $\theta = 75°$. For out-of-plane bending, the flexibility of the miter was found to be negligible. On the miter segment subjected to out-of-plane bending, the maximum measured stresses were about 90% of those for in-plane bending. However, on the

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* Closest measurement to juncture.

miter segment subjected to torsion**, the maximum measured principal stress is about 10% higher than the maximum measured principal stress for in-plane bending.

### 7.422 Cyclic Bending Fatigue Tests

Markl[7.41] gives results of a series of in-plane and out-of-plane bending fatigue tests on miter bends. These tests form the basis of Equations (7.27) through (7.30) shown herein. Three specific remarks by the author should be noted:

(1) In contrast with curved pipe, the stress intensification factor for out-of-plane bending is generally higher than for in-plane bending.

(2) The correlation obtained is adequate for proportions represented by the 4" std. wt. specimens, but a strong influence of t/r ratio is present which is not accounted for by the correlations equations.

(3) The test specimens did not lend themselves to more than a rough determination of flexibility factor.

The last two remarks indicate that the equations for miters given in the USAS Piping Code (Equations 7.27 through 7.30 herein) may be significantly in error; particularly for small values of t/r and for the out-of-plane flexibility factor.

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** In this miter, there is an abrupt change at the juncture from out-of-plane bending to torsion with respect to the pipe axes which intersect at 90° to each other at the juncture.

Macfarlane[7.24] gives results of in-plane and out-of-plane bending fatigue tests on miter bends. His results are compared with Markl's correlation equations (Equations 7.28, 7.29, and 7.30 herein) in the following tabulation.

Table 7.1. Fatigue Stress Intensification Factor ($i_f$)

| Miter Bend | Equation (7.31) | | | | Eq. (7.29) |
|---|---|---|---|---|---|
| | In-Plane Test | | Out-of-Plane Test | | |
| | $N=10^5$ | $N=5 \times 10^3$ | $N=10^5$ | $N=5 \times 10^3$ | |
| 3-segment, 9" radius | 3.5 | 3.4 | 2.2 | 2.0 | 3.2 |
| 3-segment, 12" radius | 3.5 | 3.4 | 2.2 | 2.0 | 3.2 |
| 4-segment, 9" radius | 3.5 | 3.4 | 2.2 | 2.0 | 2.7 |
| 3-segment, 5-1/2" radius | 4.7 | 4.2 | 2.2 | 2.0 | 3.2 |

Macfarlane's test results for in-plane bending were higher than predicted by Equation (7.29); lower for out-of-plane bending. Macfarlane's results are opposite to those reported by Markl for in-plane versus out-of-plane bending. Macfarlane's test models had a t/r of 0.087; Markl's, a t/r of 0.11; a relatively small difference.

Presumably all failures in these tests occurred in the juncture welds; hence, the quality of the welding is significant. Markl's models were welded from the outside only, Macfarlane's from both outside and inside.

### 7.423 Limit Bending Loads

Lane and Rose[7.23] tested 3-segment and 4-segment miters with $r = 6.18"$, $t \cong 0.37"$. The combinations of $\beta$ and s were such that the miters were similar to a curved pipe with $R = 18"$, $\alpha = 90°$. For in-plane bending moments <u>closing</u> the bend, a 3-segment miter showed general yield at 424,000 in-lb, and collapsed at 842,000 in-lb; a 4-segment miter yielded at 370,000 in-lb, and collapsed at 935,000 in-lb. For in-plane bending moment <u>opening</u> the bend, a 4-segment miter yielded at 356,000 in-lb; a load of 2,340,000 in-lb did not produce collapse.* A 4-segment miter was subjected to an out-of-plane moment of 1,330,000 in-lb (limit of test apparatus) and this moment produced no pronounced permanent set in the bend.

For comparison, the calculated yield moment of the equivalent curved pipe by Equation (7.34) is about 300,000 in-lb.

## 7.5 Summary

Theories for the elastic stresses and displacements of curved pipe are generally adequately confirmed by test data except for "end effects". A suitable theory including end effects needs to be developed. No theory exists for behavior of curved pipe in the plastic region; the behavior there is significant in the burst pressure and collapse moment. Test data indicate that the burst pressure of curved pipe made of ductile material is not much lower than the burst pressure of straight pipe; however, adaquate coverage of dimensional parameters is not available. The in-plane collapse moment for curved pipe appears to be about two times the calculated (included the i-factor) yield moment.

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* The test apparatus was such as to give increasing moment arms for loads tending to close the bend; decreasing moment arms for loads tending to open the bend. This may partially explain the large difference in collapse loads; however, the type of deformation (See Fig. 7.3) would tend to give smaller collapse loads for "closing the bend" than for "opening the bend".

Theories for miters are less well developed; they may be reasonably applicable for small values of the miter angle $\beta$. For multi-segment miters similar in proportions to curved pipe with $R/r = 3$, test data indicate a similarity in stresses and deformations for bending loads. Further study of the available theory and data on miters is needed.

## References


(7.1) Lorenz, H., "Theörie der Rohrenfedermanometer", V.D.I., Vol 54, pp 1865-1867 (1910).

(7.2) Timoshenko, S., Strength of Materials, Part II, Advanced Theory and Problems, 2nd Edition, D. Van Nostrand Co., New York, N. Y., p 160.

(7.3) Clark, Gilroy, & Reissner, "Stresses and Deformations of Toroidal Shells of Elliptical Cross Section", J. of Appl. Mechs., Vol 74, pp 37-48 (1952).

(7.4) Kalnins, A., "Analysis of Curved Thin-Walled Shells of Revolution", AIAA/ASME 8th Structures, Structural Dynamics and Materials Conference, Palms Springs, California (March, 1967).

(7.5) National Electric Manufacturers Association, Pub. No. SM-20-1958, "Mechanical Drive Steam Turbines", 155 E. 45th Street, New York 17, N. Y.

(7.6) Weil, Brock, & Cooper, "Stresses in a Pipe Bent Into a Circular Arc", J. Eng'g. for Industry, pp 449-459 (November, 1961).

(7.7) Van der Neut, A., "Bending at the Oblique End Section of Cylindrical Shells", Proc. Symp. on the Theory of Thin Elastic Shells, Editor W. T. Koiter, (North-Holland Publishing Company, Amsterdam, p 247, 1960).

(7.8) Murthy, M. V. V., "Stresses in a Mitred Pipe Joint Under Uniform Internal Pressure", Int. J. Mech. Sci., Vol 6, pp 361-370 (1964).

(7.9) Green and Emmerson, "Stresses in a Pipe with a Discontinuous Bend", J. Mech. Phys. Solids, Vol 9, pp 91-104 (1961).

(7.10) Appleyard, W., "Bending Moment, Tension and Shear at Mitred Junction of Two Cylindrical Pressure Vessels", R. A. E. Tech. Note No. Aero 1906.

(7.11) Mackenzie and Beattie, "The Right-Angled Corner in Large-Diameter Pressure Ducting", Proc. Symp. on Pressure Vessel Research Toward Better Design, Inst. Mech. Engrs., London (1962).

(7.12) Kornecki, A., "The State of Stress and Strain in a Thin-Walled Tube Rigidly Fixed in an Oblique Cross Section" (in Polish), Rozprawy Inzynierskie 65, p 117 (1957).

(7.13) Estrin, M. J., "Analysis of a Cylindrical Shell Clamped Along an Oblique Contour", (in Russian), Izv. Akad. Nauk SSSR, Otd. Teckh. Navk 2, p 151 (1959).

References (Continued)

(7.14) Corum, J. M., "A Theoretical and Experimental Investigation of the Stresses in a Circular Cylindrical Shell with an Oblique Edge", Nuclear Engineering and Design, Vol 3, pp 256-280 (1956).

(7.15) Owen and Emmerson, "Elastic Stresses in Single Mitred Bends", J. Mech. Eng'g. Science, Vol 5, No. 4, pp 303-324.

(7.16) de Leiris and Barthelemy, 1946 Bulletin de l'Association Technique Maritime et Aeronautique, Memorandum No. 867.

(7.17) Gross, N., "Experiments on Short-radius Pipe Bends", Proc. Inst. Mech. Engrs., Vol 1B, p 465 (1952-1953).

(7.18) Zeno, D. R., Discussion of Reference 7.17.

(7.19) Rodabaugh, Melnick, and Atterbury, "Experimental Investigation of Flexibility and Stress Intensification Factors of Copper Nickel Elbows", Battelle Memorial Institute Report to the U. S. Naval Marine Engineering Laboratory (March, 1966).

(7.20) Lane, P. H. R., "Fatigue Tests on Seamless Mild-Steel Pipe Bends", International Conf. on Fatigue of Metals, Session 9, Paper 3, Instn. of Mech. Engrs., London (1956).

(7.21) "Criteria of Section III of the ASME Boiler and Pressure Vessel Code for Nuclear Vessels", Published by the ASME, 345 E. 47th St., New York, N. Y. 10017.

(7.22) Blair, J. S., Discussion of Reference 7.17.

(7.23) Lane and Rose, "Experiments on Fabricated Pipe Bends", British Welding Journal (June, 1961).

(7.24) Macfarlane, D. S., "Fatigue Strength of Gusseted Pipe Bends", British Welding Journal (December, 1962).

(7.25) Bantlin, A., "Formänderung und Beanspruchung federnder Ausgleichrohre", Z., V. D. I., Vol 54, pp 43-49 (1910).

(7.26) Karman, Th., "Uber die Formanderung dunnwandiger Rohre, insbesonders federnder Ausgleichrohre", Z., V. D. I., Vol 55, pp 1889-1895 (1911).

(7.27) Lorenz, H., "Die Biegung Krummer Rohre", Z. Physik, Vol 13, pp 768-774 (1912).

(7.28) Marbec, M., "Flexibilite des tubes", Bulletin de l'Association Technique Maritime, Vol 22, pp 441-557 (1911).

(7.29) Wahl, A. M., "Stresses and Reactions in Expansion Pipe Bends", Trans. ASME, Vol 49-50, Paper FSP-50-49 (1928).

References (Continued)

(7.30) Hovgaard, W., "The Elastic Deformation of Pipe Bends", J. of Math. and Physics, Vol 6, No. 2, pp 69-118 (1926).

(b) "Deformation of Plane Pipes", J. of Math. and Physics, Vol 7, No. 3, pp 198-238 (1928).

(c) "Further Research on Pipe Bends", J. of Math. and Physics, Vol 7, No. 4, pp 239-299 (1928).

(d) "Tests on High Pressure Pipe Bends", J. of Math. and Physics, Vol 8, No. 4, pp 293-344 (1929).

(e) "Deflections and Stresses in Pipe Bends", Proc. of the World Engineering Congress, Tokyo, Vol 3-4, pp 1-30 (1929).

(f) "Bending of Curved Pipe", Proc. of the 3rd Intl.Congress for Applied Mechanics, Stockholm, Vol II, pp 331-341 (1930).

(g) "Stresses in Three Dimensional Pipe Bends", Trans. ASME, Vol 57, FSP 57-12 (1935).

(h) Further Studies of Three Dimensional Pipe Bends", Trans. ASME, Vol 59, FSP 59-13 (1937).

(7.31) Cope and Wirt, "Load-Deflection Relations for Large Plain, Corrugated and Creased Pipe Bends", Trans. ASME, Vol 54, FSP-54-12 (1932).

(7.32) Vigness, I., "Elastic Properties of Curved Tubes", Trans. ASME, Vol 65, p 105-120 (1943).

(7.33) Shipman, W. H., "Design of Steam Piping to Care for Expansion", Trans. ASME, Vol 51, pp 415-446 (1929).

(7.34) Jenks, H. E., Discussion of Reference 7.33.

(7.35) Karl, H., "Biegung gekrümmeter dünwandiger Rohre", Z. Angew, Math. Mech., Vol 23, pp 331-345 (1943).

(7.36) Beskin, L., "Bending of Curved-Thin Tubes", J. of Appl. Mechanics, Trans. ASME, Vol 67, pp A-1-7 (1945).

(7.37) Clark and Reissner, "Bending of Curved Tubes", Advances in Applied Mechanics, Academic Press, New York, N. Y., Vol 2, pp 93-133 (1951).

(7.38) Symonds and Pardue, "Characteristics of Short Radius Tube Bends, Second Partial Report (Theoretical)", Naval Research Laboratory Report No. 0-2761 (February 18, 1946).

(7.39) Pardue and Vigness, "Properties of Thin-Walled Curved Tubes of Short-Bend Radius", Trans. ASME, Vol 73, pp 77-87 (1951).

References (Continued)


(7.40) Markl, A. R. C., "Fatigue Tests of Welding Elbows and Comparable Double-Metric Bends", Trans. ASME, Vol 69, pp 869-879 (1947).

(7.41) Markl, A. R. C., "Fatigue Tests of Piping Components", Trans. ASME, Vol 74, pp 287-303 (1952).

(7.42) Rodabaugh, E. C., Private Communication to S. E. Moore (September, 1967).

(7.43) Turner and Ford, "Examination of the Theories for Calculating the Stresses in Pipe Bends Subjected to In-Plane Bending", Proc. Inst. Mech. Engrs., Vol 171, pp 513-525 (1957).

(7.44) Smith, R. T., "Theoretical Analysis of the Stresses in Pipe Bends Subjected to Out-of-Plane Bending", J. Mech. Eng'g. Science, Vol 9, No. 2 (1967).

(7.45) Jones, N., "In-Plane Bending of a Short-Radius Curved Pipe Bend", J. of Eng'g. for Industry, Trans. ASME (May, 1967).

(7.46) Barthélemy, J., "Etude de la Deformation et des Tensions Internes des Tuyaux a Ligne Moyenne Plane, Soumis a des efforts exterieurs et a une pression interne", Bulletin de l'Association Technique Maritime (1947).

(7.47) Kafka and Dunn, "Stiffness of Curved Circular Tubes with Internal Pressure", J. of Appl. Mechanics, Trans. ASME, Vol 78, pp 247-254 (1956).

(7.48) Rodabaugh and George, "Effect of Internal Pressure on Flexibility and Stress-Intensification Factors of Curved Pipe or Welding Elbows", Trans. ASME, Vol 79, pp 939-948 (1957).

(7.49) Kitching, R. "Miter Bends Subjected to In-Plane Bending Moments", Int. J. Mech. Sci., Vol 7 pp 551-575 (1965).

(7.50) Jones and Kitching, "A Theoretical Study of In-Plane Bending of a Single Unreinforced Mitered-Bend", J. of Strain Analysis, Vol 1, No. 3 (1966).

(7.51) Wahl, Bowley, and Back, "Stresses in Turbine Pipe Bends", Mechanical Engineering, Vol 51, No. 11, pp 823-828 (1929).

(7.52) Mayrose, H. E., Discussion of 7.30(h), Trans. ASME, Vol 58, pp 395-397 (1936).

(7.53) Vigness, I., "Elastic Properties of Curved Tubes, Trans. ASME, Vol 65, pp 105-120 (1943).

(7.54) Gross and Ford, "The Flexibility of Short-Radius Pipe-Bends", Proc. Instn. Mech. Eng'rs., Series B, Vol 1, pp 480-491 (1952-1953).

References (Continued)

(7.55) Vissat and Del Buono, "In-Plane Bending Properties of Welding Elbows", Trans. ASME, Vol 77, pp 161-175 (1955).

(7.56) Smith and Ford, "Experiments on Pipelines and Pipebends Subjected to Three-Dimensional Loading", J. Mech. Eng'g. Science, Vol 9, No. 2, pp 124-137 (1967).

(7.57) Findlay and Spence, "In-Plane Bending of a Large 90 Degree Smooth Bend", J. of Strain Analysis, Vol 1, No. 4, pp 290-300 (1966).

(7.58) Markl, A. R. C., Descussion of Stewart and Schreitz, "Thermal Shock and Other Comparison Tests of Austenitic and Ferritic Steels for Main Steam Piping--A Summary Report", Trans. ASME (August, 1953).

(7.59) Rekate, H. L., "Fatigue Characteristics of 70-30 Cu-Ni Pipe and Fittings for Submarine Sea-Water Systems", U. S. Navy Marine Engineering Laboratory (Annapolis, Maryland), R and D Phase Report No. 98/65 (1965).

(7.60) Kitching, R., "In-Plane Bending of a 180° Mitered Pipe Bend", Int'l. J. Mech. Science, Vol 7, pp 721-736 (1965).

(7.61) Jones and Kitching, "An Experimental Investigation of a Right-Angled Single Unreinforced Mitered-Bend Subjected to Various Bending Moments", J. of Strain Analysis, Vol 1, No. 3 (1966).

(7.62) Rodabaugh, Duffy and Atterbury, "The Internal Pressure Capacity of Butt Welding Elbows", Battelle (Columbus, Ohio) Report to the American Gas Association, Sept. 1969.

(7.63) Feltz and Phillips, "Smooth Bends on Steel Mains", Project GII-3, Michigan Consolidated Gas Company.

CHAPTER 8

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# 8. BRANCH CONNECTIONS

The term "branch connections" as used herein is intended to cover both standard tees, crosses, laterals, and wyes as well as shop or field fabricated branch connections. Nomenclature is shown in Figures 8.1 and 8.2.

Standard branch connections are defined as those manufactured to meet:

ASA B16.9, "Wrought Steel Buttwelding Fittings"

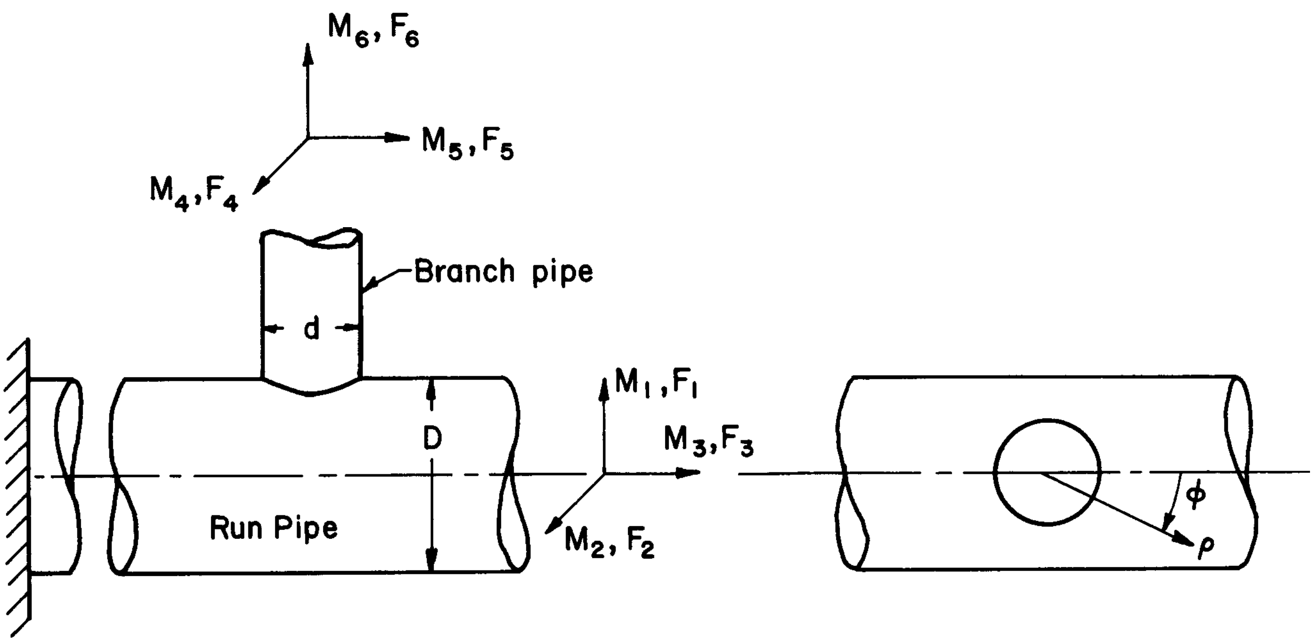
ASA B16.11, "Forged Steel Fittings, Socket-Welded and Threaded"

ASA B16.5, "Steel Pipe Flanges and Flanged Fittings".

Branch connections for critical service piping systems are usually standard fittings, to the extent that standard fittings are available in suitable size. B16.9 reducing fittings have minimum branch sizes about equal to one-half of the run size; for smaller branch sizes a fabricated branch connection is used. For example, a 12" or 10" branch in a 12" line would usually be a standard tee; a branch smaller than 5" would usually be fabricated.

Fabricated branch connections are usually reinforced by addition of metal around the opening. Such additional metal may consist of pads or saddles, heavy-wall couplings, heavy-wall branch pipe, or manufactured nozzle reinforcements such as "Weldolets".

The large majority of branch connections in piping systems are "tees"; i.e., a pipe branch with axis normal to the run pipe surface. Reinforcing is ordinarily placed so that the flow-area of the run and branch pipes is not reduced. There are, however, many other types of branch connections used in piping systems. For example; laterals, wyes, crosses or hillside connections. In addition, branch connections are occasionally made in closures; those closures may be formed heads or flat plates.




D = run pipe mean diameter

T = run pipe wall thickness

d = branch pipe mean diameter

t = branch pipe wall thickness

$\emptyset$, $\rho$ = co-ordinates and stress directions

P = internal pressure

$M_i$, $F_i$ = moment or force vector, i = 1 to 6

For "straight" tee d = D

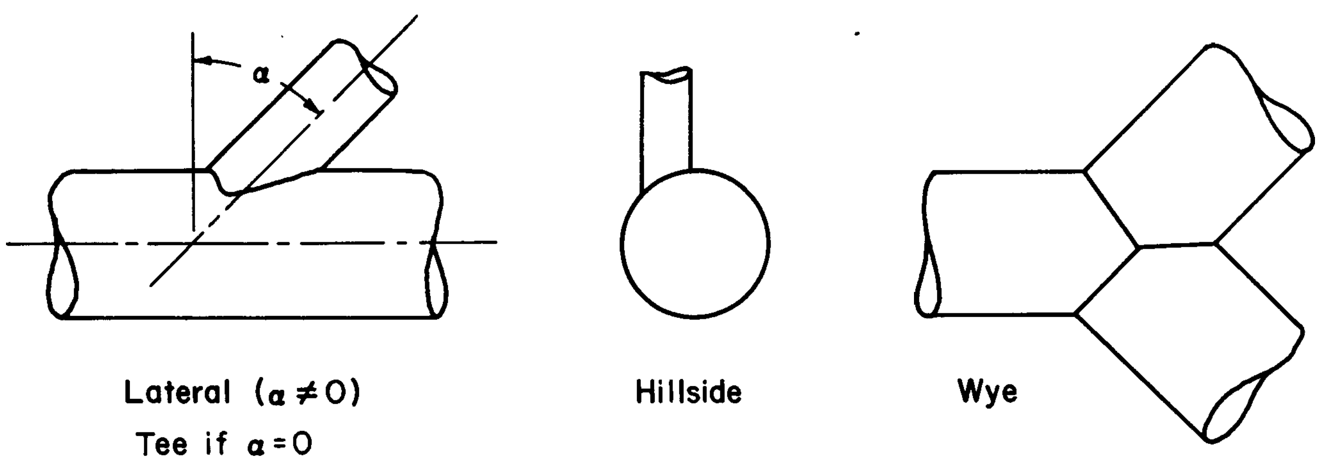
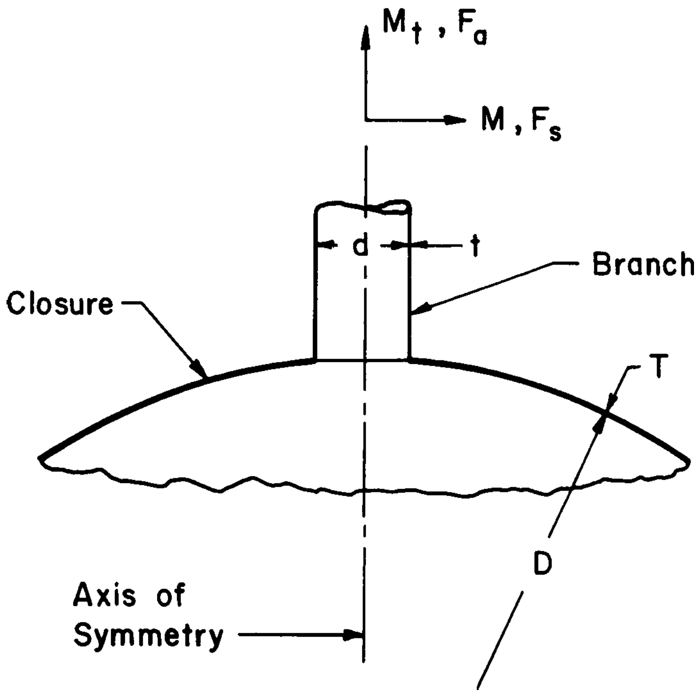



FIGURE 8.1: NOMENCLATURE ILLUSTRATION, CYLINDER-TO-CYLINDER CONNECTIONS




Radial Connection

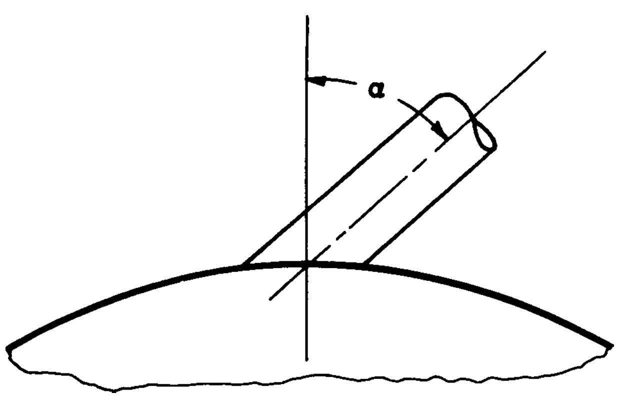
(When D $\longrightarrow \infty$ , the closure becomes a flat plate)

$M_t$ = torsional moment on branch

$M$ = bending moment on branch

$F_a$ = axial force on branch

$F_s$ = shear force on branch




Non - Radial Connection

FIGURE 8.2: NOMENCLATURE ILLUSTRATION, CYLINDER-TO-CLOSURE CONNECTIONS

Branch connections are included in the still broader category of openings in pressure vessels, including aircraft structures and submarine hulls. Much of the theory and test data cited herein was developed in connection with pressure vessel technology. A related technological area is that of connections between tubular members used for structures.

There are several summary papers or reports on the theory and test data pertinent to branch connections. These are tabulated below:

| Author(s) | Reference No. | Contents |
|---|---|---|
| Waters, E. O. | 8.1 | Historical background and summary of test data available up to 1955. |
| Mershon, J. L. | 8.2 | A critical review and comparison of test data accumulated in the PVRC* program plus other significant data available at that time (1962). |
| Langer, B. F. | 8.3 | Summary of data and theory on external loadings (1964). |
| Mershon, J. L. | 8.4 | Summary and comparisons of test data on openings and branch connections with internal pressure loading (1964). |
| Mershon, J. L. | 8.5 | Evaluation of test data obtained in PVRC program at University of Illinois and Westinghouse. |
| Rodabaugh, et al | 8.6 | Phase Report 2: Branch connections in spherical shells, comparisons of theories and comparisons of analysis with test data. |

* PVRC is the Pressure Vessel Research Committee of the Welding Research Council.

| Author(s) | Reference No. | Contents |
|---|---|---|
| Rodabaugh, et al | 8.7 | Phase Report 3: Flexibility factors of branch connections in spherical shells. |
| | 8.8 | Phase Report 5: Branch connections in cylindrical shells, comparisons of theories and comparisons of analysis with test data. |
| | 8.9 | Phase Report 6: Flexibility factors of branch connections in cylindrical shells. |

## 8.1 Internal Pressure Loading, Theory

### 8.11 Branches in Pipe, Theory

Until a few years ago, analytical estimates of stresses at small branches or small openings ($d/D \ll 1$) were often obtained by reducing the problem to that of an opening or nozzle in a flat plate with edge loads. Papers by Beskin[(8.10)] and Waters[(8.11)] are examples of this kind of approximation. A further step consisted of the solution of a cylindrical shell with a circular opening. Papers by Lourye[(8.12)], Withum[(8.13)], Eringen, et al[(8.14)], Lekkerkerker[(8.15)], Savin[(8.16)], and Van Dyke[(8.17)], give solutions to this problem. The next step consisted of the solution of two normally intersecting cylindrical shells. Solutions to this problem are given by Reidelbach[(8.18)] and Eringen, et al[(8.19)].

The theory developed in Reference (8.19) has been programmed for a computer. The theoretical results obtained are compared with test data in Reference (8.8). Eringen's[(8.19)] analysis is limited to relatively small branch connections having the following parameters:

(a) $d/D < \sim 1/3$

(b) $(d/D) \sqrt{D/T} < \sim 1.1$

(c) $D/T$ and $d/t > \sim 10.$

(d) Branch pipe and run pipe with uniform wall thickness

(e) Branch pipe axis normal to the run pipe surface, and branch pipe extending to run pipe surface (no inward protrusion)

(f) An isolated branch connection; i.e., no other branch connection or other source of stress discontinuity in the neighborhood.

Whereas Eringen's analysis is limited to relatively small branch connections, Lind[8.20] has developed a semiempirical method for estimating the value of $\sigma_\varphi$ at $\varphi = 0$, $\rho = d/2$ (usually, the maximum stress) for branch connections in pipe for d/D up to 1.00. Lind's analysis is subject to limitations (d), (e), and (f) listed above for Eringen's analysis. Comparisons of Lind's method with test data and with Eringen's analysis are given in Reference (8.8).

Bijlaard, Dohrman, and Wang[8.21] give a theoretical development intended to be applicable to straight tees and includes certain elements of thick-shell theory. A proposed method for the numerical solutions is given; however, at this time numerical results from the method have not been published.

Tabakman[8.22] gives a theoretical development and computer program listing applicable to normally-intersecting cylindrical shells. Shallow-snell theory is used for the run cylinder, therefore d/D is limited to about one-third. The theory is developed only for "open-ends" of the cylinders. Numerical results are given, but because of the open-end boundaries, no comparisons can be made with available test data or theories.

As applied to either standard or fabricated pipe line branch connections, the theories for the intersection of two uniform-wall cylindrical shells discussed above are rather limited in direct application for the following reasons. Most pipe-line branch connections include local reinforcing close to the branch. Tees (and crosses) made to ASA B16.9 are usually provided with fairly large transition radii between the branch and the run portions, and with heavy end reinforcements.

Fabricated branch connections are usually reinforced to meet pressure vessel or piping rules. These rules require that the area cut out by the opening must be replaced within a restricted region around the opening. In a crude sense, this rule may be considered as being the result of a limit pressure analysis. Provided the material is sufficiently ductile, the area replacement rule should insure that the pressure causing gross yielding or bursting of the branch connection is not much less than that pressure required to yield or burst the unperforated run pipe. This kind of analysis is applicable to many structures not included in the theories; e.g., laterals, hillside branches, closely-spaced branches, openings other than circular.

Computer programs using finite elements to model complex structural shapes have been under development for several years. These kinds of analyses are, in principle, applicable to such complex shapes as USAS B16.9 tees. Insofar as the writer is aware, at present such programs* have not yet been developed to the stage where they can be used with confidence to predict accurately the stresses in an ASA B16.9 tee.

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* Some examples: ELAS[8.150], FORMAT II[8.23], SAMIS[8.24], CSMTRX[8.25], PAPA[8.26].

Some analytical guidance is available with respect to "limit pressures" of branch connections in cylinders. Hodge(8.27) gives a theory for a ring-reinforced opening in a cylindrical shell. Coon, Gill, and Kitching(8.28) give the lower bound to the limit pressure of a cylindrical shell with a circular opening. Cloud and Rodabaugh(8.29) give an approximate analysis for a branch pipe in a cylindrical shell.

## 8.12 Branches in Closures, Theory

If a branch is located in a closure so that in the vicinity of the branch connection the radius of curvature of the shell is constant, then axi-symmetric analytical methods can be applied. A number of computer programs (References 8.30 - 8.33) have been developed for specific application to nozzles in spherical shells; general shell-of-revolution programs (References 8.34 - 8.38) are also available; the specific programs have an advantage over the general programs in that in-put data are simpler and computer running time is less. Finite element programs (8.39, 8.40) are also available. They should be more accurate, particularly for thick-wall shells, however, the input data and computer time are several orders of magnitude more time-consuming than for the shell programs. Finally, there is a point-matching program (8.41) available for axisymmetric-structures. This, like the finite-element programs, should be more accurate for thick-wall shells but at increased input data and computer time cost. Some comparisons of the results from several of the programs listed are given in Reference (8.6).

The limit pressure of nozzles in spherical shells can be calculated by a number of different approaches. Upper and/or lower bound analyses are given in References (8.42) through (8.46). An exact (with a certain yield criteria) analysis of the limit pressure is given by Gerdeen[8.47]; this being a general shell of revolution program applicable to nozzles in spheres as a special case. The computer program FEELAP[8.40] (finite element, elastic-plastic) gives stresses and displacements in both the elastic regime and plastic regime and includes nozzles in spheres as a special case of axi-symmetric structures.

As indicated by the above, the theory for branch connections, in closures where they are axisymmetric structures, is relatively well advanced. However, where the branch connection is not normal to a formed head surface or is in a region of variable radius of curvative (e.g., the knuckle of a flanged-and-dished head), the axisymmetric theories are not applicable. Johnson[8.48] gives an analysis for a non-radial nozzle in a spherical shell. The angle $\alpha$ must be fairly small; less than 20° for R/T of 5, less than 9° for R/T of 30. Corum[8.49] gives an analysis for a cylindrical shell with an oblique edge that may be developable to a basis for non-radial connections.

## 8.2 Internal Pressure Loading, Test Data

Table 8.1 gives a summary of available, published test data on branch connections and openings in piping or pressure vessels. Table 8.1 includes references giving test data on external loads as well as internal pressure loading. With respect to internal pressure loading, four types of results are available:

(Text continued on p. 22)

TABLE 8.1: TEST DATA ON BRANCH CONNECTIONS

Sheet 1 of 12

(1) Structure

These columns indicate the general type of test specimen. An "X" under "Cyl." indicates a branch connection in a cylindrical run pipe. An "X" under "Closure" indicates a branch connection in a spherical shell, ellipsoidal head or a flat plate. The entry under "$\alpha$" indicates whether the nozzles were radial ($\alpha$ = o) or non-radial ($\alpha \neq$ o). See Figures 8.1 and 8.2 for definition of $\alpha$.

(2) Loads

These two columns indicate what test loads were applied. An "X" under "P" indicates internal pressure loading was applied. Entries under "M" indicate what external loads were applied; as defined in Figures 8.1 and 8.2.

(3) Measurements

These columns indicate what information is presented.

"X" under S indicates stresses are given, either from strain gages or photoelastic measurements.

"X" under $P_y$ indicates that internal pressure corresponding to yielding or limit pressure is given.

"X" under $P_u$ indicates that internal burst pressure is given.

Entries in column "N" indicate fatigue test were run. $N_p$ indicates cycles of internal pressure loading to obtain fatigue failure are given; $N_m$ is analogous, for external loads.

"X" under K indicates displacements are given such that a flexibility factor can be calculated.

TABLE 8.1: TEST DATA ON BRANCH CONNECTIONS

Sheet 2 of 12

| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | $\alpha$ | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.50 | Atterbury, et al | X | -- | 0 | X | $M_4$, $M_5$ | X | -- | -- | $N_p$ | -- |
| 8.51 | Atterbury, et al | X | -- | 0 | X | -- | X | -- | -- | -- | -- |
| 8.52 | Atterbury, et al | X | -- | 0 | X | -- | X | -- | -- | -- | -- |
| 8.53 | Barkow & Huseby | X | -- | 0 | X | -- | X | X | X | -- | -- |
| 8.54 | Berman & Pai | X | -- | (1) | X | -- | X | -- | -- | -- | -- |
| 8.55 | Berman & Pai | X | -- | (1) | X | -- | X | -- | -- | -- | -- |
| 8.56 | Berman & Pai | X | -- | (1) | X | -- | -- | -- | -- | $N_p$ | -- |
| 8.57 | Bernsohn, et al | X | -- | 0 | -- | -- | -- | -- | -- | $N_m$ (2) | -- |

(1) Includes one "hillside" branch connection.

(2) Bending fatigue tests, austenitic steel materials at 900 F and 1050 F.

TABLE 8.1: (Continued)

Sheet 3 of 12

| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | $\alpha$ | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.58 | Blair, J.S. | X | -- | Various | X | $M_4$ | (1) | X | X | $N_m$ | (2) |
| 8.59 | Clare & Gill | X | -- | 0 | X | -- | -- | X | -- | -- | -- |
| 8.60 | Cloud, R. L. | X | -- | 0 | X | -- | -- | X | -- | -- | -- |
| 8.61 | Cottam & Gill | X | -- | 0 | X | -- | -- | X | X | -- | -- |
| 8.62 | Cranch, E. T. | X | -- | 0 | X | (3) | X | -- | -- | -- | X |
| 8.63 | Dally, J. W. | -- | X | 0 | — | $F_a$ & M | X | -- | -- | -- | X |

(1) Gives proportional limit for moment loading.

(2) Possibly contains enough information to determine flexibility factor.

(3) Loads $F_1$, $M_4$ and $M_5$.

TABLE 8.1: (Continued)

Sheet 4 of 12

| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | $\alpha$ | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.64 | Dinno & Gill | -- | X | 0 | X | -- | -- | X | -- | -- | -- |
| 8.65 | Dubuc & Welter | X | -- | 0 | X | -- | -- | -- | -- | $N_p$ | -- |
| 8.66 | Ellyin, F. | -- | X | 0 | X | -- | -- | X | -- | -- | -- |
| 8.67 | Faupel & Harris | X | -- | (1) | X | -- | X | -- | -- | -- | -- |
| 8.68 | Fessler & Lewin | X | -- | 0 | X | -- | X | -- | -- | -- | -- |
| 8.69 | Fessler & Lewin | X | -- | 0 | X | -- | X | -- | -- | -- | -- |

(1) Holes--no branch attached.

TABLE 8.1: (Continued)

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| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | α | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.70 | Everett & McCutchan | X | -- | 0 | X | -- | X | -- | X | -- | -- |
| 8.71 | Gross, Nicol | X | -- | 0 | X | -- | X | -- | -- | -- | -- |
| 8.72 | Graalfs, H. E. | (1) | -- | (1) | X | -- | X | X | X | -- | -- |
| 8.73 | Greenwald, D. K. | X | -- | 0 | X | -- | -- | -- | X | -- | -- |
| 8.74 | Greenstreet, et al | -- | X | (2) | X | -- | X | -- | -- | -- | -- |
| 8.75 | Hardenbergh, et al | X | -- | 0 | X | (3) | X | -- | -- | -- | -- |

(1) 10", 90° elbow with 6" back branch connection.

(2) Cluster of nozzles in spherical shell plus several isolated nozzles.

(3) Loadings: $F_6$, $M_4$, $M_5$, $M_6$.

TABLE 8.1: (Continued)

Sheet 6 of 12

| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | $\alpha$ | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.76 | Hardenbergh & Zamrick | X | -- | 0 | X | (1) | X | -- | -- | -- | -- |
| 8.77 | Heirman & Stockman | X | -- | Various | X | -- | X | -- | -- | $N_p$ | -- |
| 8.78 | Hiltscher & Florin | -- | X | Various | X | -- | X | -- | -- | -- | -- |
| 8.79 | Hiltscher & Florin | -- | (2) | $52^o$ | (2) | -- | X | -- | -- | -- | -- |
| 8.80 | Horseman, R. W. | -- | X | Various | X | -- | X | -- | -- | -- | -- |
| 8.81 | Kaufman, W. J. | X | -- | 0 | X | -- | X | -- | -- | -- | -- |
| 8.82 | Kitching & Duffield | -- | X | 0 | X | $F_a$ | X | -- | -- | -- | -- |

(1) Loadings: $F_6$, $M_4$, $M_5$.

(2) Oblique nozzle in a plane plate in tension.

TABLE 8.1: (Continued)

Sheet 7 of 12

| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | $\alpha$ | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.83 | Kitching & Olsen | -- | X | 0 | X | M | X | -- | -- | -- | -- |
| 8.84 | Kitching & Jones | -- | X | 0 | X | $F_a$, M | X | -- | -- | -- | -- |
| 8.85 | Lane, P.H.R. | X | -- | 0 | X | (1) | X | -- | -- | -- | X |
| 8.86 | Lane, P.H.R. | X | -- | 0 | X | (1) | X | -- | -- | -- | X |
| 8.87 | Lane & Quartermaine | X | -- | 0 | X | (1) | X | -- | -- | -- | -- |
| 8.88 | Lane, P.H.R. | X | -- | 0 | X | (1) | X | X | X | $N_p$ | X |
| 8.89 | Lane, P.H.R. | X | -- | 0 | X | -- | -- | -- | -- | $N_p$ | -- |
| 8.90 | Lane & Rose | X | -- | 0 | X | (2) | X | -- | -- | $N_p$ | -- |
| 8.91 | Lane & Rose | X | -- | 0 | X | -- | -- | -- | -- | $N_p$ | -- |
| 8.92 | Lane & Rose | X | -- | 0 | X | -- | X | -- | -- | -- | -- |
| 8.93 | LeCocq, J. | X | -- | 0 | -- | $F_6$, $M_4$, $M_5$ | X | -- | -- | -- | -- |

(1) $F_6$, $M_4$, $M_6$

(2) $F_6$, $M_4$, $M_5$, $M_6$

TABLE 8.1: (Continued)

Sheet 8 of 12

| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | $\alpha$ | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.94 | Leven, M. M. | X | X | 0 | X | M | X | -- | -- | -- | -- |
| 8.95 | Leven, M. M. | -- | X | $45^o$ | X | -- | X | -- | -- | -- | -- |
| 8.96 | Leven, M. M. | X | -- | 0 | X | -- | X | -- | -- | -- | -- |
| 8.97 | Lind, et al | X | -- | 0 | X | -- | X | X | -- | -- | -- |
| 8.98 | Lind & Palusamy | -- | X | 0 | X | $F_s$, M | -- | (1) | -- | -- | -- |
| 8.99 | MacKenzie & Spence | -- | X | Various | X | -- | X | -- | -- | -- | -- |
| 8.100 | Mantle & Proctor | -- | X | $50^o$ | X | -- | X | -- | -- | -- | -- |
| 8.101 | Markl, A.R.C. | X | -- | 0 | -- | Various | -- | -- | -- | $N_m$ | -- |
| 8.102 | Markl, et al | X | -- | 0 | X | -- | -- | -- | -- | $N_p$ | -- |
| 8.103 | Maxwell & Holland | -- | X | 0 | X | M,$F_a$ $F_s$ | X | -- | -- | -- | -- |
| 8.104 | Maxwell & Holland | -- | X | 0 | X | M, $F_a$, $F_s$ | X | -- | -- | -- | -- |

(1) Limit load under combined pressure and external loads.

TABLE 8.1: (Continued)

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| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | $\alpha$ | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.105 | Maxwell, et al | -- | X | 0 | X | M , $M_t$, $F_a$, $F_s$ | X | -- | -- | -- | -- |
| 8.106 | Maxwell & Holland | -- | X | 0 | X | M , $M_t$, $F_a$, $F_s$ | X | -- | -- | -- | -- |
| 8.107 | McClure, et al | X | -- | 0 | X | $M_4$, $M_5$ | X | X | X | -- | X |
| 8.108 | McClure, et al | X | -- | 0 | X | $M_4$, $M_5$ | X | -- | X | $N_p$ | X |
| 8.109 | Mehringer & Cooper | X | -- | 0 | X | $F_6$, $M_4$, $M_5$ | X | -- | -- | -- | X |
| 8.110 | Mills, et al | X | -- | 0 | X | $M_4$, $M_5$ | -- | -- | -- | $N_m$ | X |
| 8.111 | O'Toole, Rodabaugh & George | X | -- | 0 | X | -- | -- | -- | -- | $N_p$ | -- |
| 8.112 | Pickett & Gregory | X | X | (1) | X | X | X | -- | -- | $N_p$, $N_m$ | X |

(1) Includes a nonradial nozzle on a spherical end closure.

TABLE 8.1: (Continued)

Sheet 10 of 12

| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | $\alpha$ | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.113 | Pickett & Gregory | X | X | (1) | X | -- | X | -- | -- | $N_p$ | -- |
| 8.114 | Riley, W. F. | X | X | 0 | X | (2) | X | -- | -- | -- | -- |
| 8.115 | Rodabaugh & George | X | -- | Various | X | Various | -- | X | X | $N_p$, $N_m$ | -- |
| 8.116 | Rodabaugh, E. C. | X | -- | 0 | -- | $M_4$, $M_5$ | -- | -- | -- | $N_m$ | X |
| 8.117 | Rose, R. T. | X | -- | 0 | X | -- | X | X | X | -- | -- |
| 8.118 | Rose, R. T. | -- | X | 45° | X | -- | X | -- | -- | -- | -- |
| 8.119 | Schoessow & Kooistra | X | -- | 0 | -- | $F_6$, $M_4$, $M_5$ | X | -- | -- | -- | -- |
| 8.120 | Schoessow & Brooks | X | X | 0 | X | -- | X | -- | -- | -- | -- |
| 8.121 | Seabloom, E. R. | X | -- | 0 | X | -- | -- | -- | X | -- | -- |

(1) Includes a nonradial nozzle on a spherical closure.

(2) For connection in closure (sphere): $F_a$ and M. For connection in cylinder, $F_6$,$M_4$ and $M_5$ and combined internal pressure with $M_5$.

TABLE 8.1: (Continued)

Sheet 11 of 12

| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | $\alpha$ | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.122 | Siebel & Schwaigerer | X | -- | Various | X | -- | X | X | X | -- | -- |
| 8.123 | Soete, et al | X | -- | 0 | X | -- | X | -- | -- | $N_p$ | -- |
| 8.124 | Stepanek, S. | X | -- | 0 | X | -- | X | -- | -- | -- | -- |
| 8.125 | Stockman, G. | X | -- | 0 | X | -- | X | -- | -- | $N_p$ | -- |
| 8.126 | Stone & Hochschild | X | -- | 0 | X | -- | X | -- | -- | -- | -- |
| 8.127 | Taylor & Waters | X | X | 0 | X | -- | X | -- | -- | -- | -- |
| 8.128 | Taylor & Lind | X | X | 0 | X | -- | X | -- | -- | -- | -- |
| 8.129 | Taylor, T. E. | X | -- | 0 | X | -- | -- | -- | -- | $N_p$ | -- |
| 8.130 | Taylor, T. E. | -- | X | (1) | X | -- | X | -- | -- | $N_p$ | -- |
| 8.131 | Townley, et al | -- | X | (2) | X | -- | X | X | X | -- | -- |

(1) Opening in spherical shell. Fatigue failure at opening.

(2) Cluster of nozzles in a spherical shell. Various angles.

TABLE 8.1: (Continued)

Sheet 12 of 12

| Ref. No. | Author(s) | Structure | | | Loads | | Measurements | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cyl. | Closure | α | P | M | S | $P_y$ | $P_u$ | N | K |
| 8.132 | Watzke, J. T. | X | -- | 0 | X | $M_4$, $M_5$ | X | -- | -- | -- | X |
| 8.133 | Wellinger, et al | X | -- | 0 | X | -- | X | X | -- | -- | -- |
| 8.134 | Wellinger & Krageloh | X | -- | 0 | X | -- | X | X | -- | -- | -- |
| 8.135 | Wellinger, et al | X | -- | 0 | X | -- | X | X | X | -- | -- |
| 8.136 | Wells, Lane & Rose | X | -- | 0 | X | -- | X | -- | -- | -- | -- |
| 8.137 | Welters & Dubuc | X | -- | 0 | X | -- | -- | -- | -- | $N_p$ | -- |
| 8.138 | Welters & Dubuc | X | -- | 0 | X | -- | X | -- | -- | $N_p$ | -- |
| 8.139 | Williams & Huler | X | -- | (1) | X | -- | X | X | -- | -- | -- |
| 8.140 | Winkler, et al | X | -- | 0 | X | -- | -- | X | -- | -- | -- |
| 8.141 | Wollering & Vazquez | X | -- | 0 | X | $M_4$, $M_5$ | X | - | X | -- | -- |
| 8.151 | Zick, Crossett & Lankford | X | -- | 0 | X | -- | -- | - | X | -- | -- |

(1) Unreinforced openings in a cylindrical pressure vessel.

(1) Static pressure, measured strains (converted to stresses)

(2) Static pressure, yielding determined (including limit-pressure data)

(3) Static pressure, burst pressure determined

(4) Cyclic pressure, cycles to produce fatigue failure determined.

These four types of test results are discussed in the following.

## 8.21 Static Pressure, Measured Stresses

Prior to the general use of bonded, electrical resistance strain gages (around 1945 in pressure vessel and piping tests), strains were measured by means of mechanical strain gages such as the Berry or Tuckerman types. For accessible areas in which strain gradients are small, such gages can give reasonably good results. Test data obtained from such gages are given in References (8.70) and (8.127).

About 1950 three major programs were started in which stresses due to internal pressure loading were determined.

### 8.211 Pressure Vessel Research Committee (PVRC), with Navy and USAEC Funds

This work was started in 1951. Test data on stresses due to internal pressure are given in:

| Reference No. | Contents |
|---|---|
| 8.62 | Cornell University (Cranch) tests in a cylindrical shell with two radial branch connections. (Other attachments included; tests run for correlation with Bijlaard's(8.142) analysis.) |

| Reference No. | Contents |
|---|---|
| 8.75, 8.76 | Penn State (Hardenbergh) tests on cylindrical steel models, somewhat representative of pipeline branch connections. |
| 8.114 | IIT (Riley) tests on a thin-wall (D/T $\cong$ 230) cylindrical and spherical steel models |
| 8.128 | University of Illinois (Taylor and Lind) photoelastic test models, principally representative of nozzles in pressure vessels; mostly spherical |
| 8.94 | Westinghouse Research Laboratories (Leven) photoelastic test models, principally representative of nozzles in pressure vessels, mostly spherical |
| 8.95 | Westinghouse Research Laboratories (Leven) photoelastic test models of oblique (45°) nozzles in spheres |
| 8.96 | Westinghouse Research Laboratories (Leven) photoelastic test models of a thin-wall (D/T = 100) cylinder-to-cylinder intersection |
| 8.112, 8.113 | Southwest Research Institute (Pickett and Grigory) tests on steel models with nozzles representative of pressure vessel nozzles in both cylinders and spherical heads |
| 8.103, 8.104, 8.105, 8.106 | University of Tennessee (Maxwell and Holland) tests on aluminum or steel spherical shells with radial nozzles, protruding and flush |

| Reference | Contents |
|---|---|
| 8.74 | Oak Ridge National Laboratory (Greenstreet) tests on a pressure vessel spherical head with clustered nozzles. |

8.212 American Gas Association (A.G.A.)

This work was carried out at Battelle Memorial Institute, Columbus Ohio. Because the results of this program have not been published in engineering journals, an abstract of the program is given in APPENDIX A of this Chapter.

8.213 British Welding Research Association

This program was aimed primarily at pipeline branch connections as contrasted to pressure vessel nozzles. It includes tests on (probably) the equivalent of an ASA B16.9 tee; the only known, published data on what is probably the most used branch connection in pipelines. Results of most of this work are given in various reports by Wells, Lane, and Rose (References 8.85 through 8.92; 8.117, 8.118, and 8.136).

In addition to the three major programs listed above, many other contributions have been made as indicated by the references in Table 8.1.

In evaluating this data, the following points might be noted.

(1) Essentially no data exist on stresses due to internal pressure for standard tees (USAS B16.9, B16.11, B16.5).

(2) In many references cited, the <u>maximum</u> stress may not have been measured.

(3) For most test models with $\alpha = 0$, the maximum stress was found on the inside surface, $\sigma_\varphi$ at $\varphi = 0$, $\rho = d/2$ (See Figure 8.1 and 8.2). This stress is largely of a membrane nature. For flush nozzles, the maximum stress usually is on the inside corner; for inwardly protruding nozzles the maximum stress is at about the midwall of the shell. For small nozzles ($d/D < 0.5$) reinforced by "area-replacement", this membrane-type stress will probably not exceed 3 or 4 times S, where S is the nominal stress due to internal pressure in the un-perforated shell. However, for pad or saddle reinforced connections in thin-wall cylinders, the peak stress at the edge of the reinforcing can be quite high (See p. 48 of APPENDIX A).

(4) For laterals in cylinders with large $\alpha$, the stress at the acute inside corner is probably significantly higher than for a corresponding branch connection with $\alpha = 0$.

(5) Relatively little data are available on "closely spaced" nozzles or branch connections.

## 8.22 Static Pressure, Yielding or Limit Pressure

As indicated by the check marks on Table 8.1, data are available on a fairly extensive range of branch connections in both spherical and cylindrical shells. One set of data (Reference 8.97) is available for two ASA B16.9 tees.

The German design procedure, AD-Merkblatt-B9, is based on a pressure which produces a permanent strain at the branch connection of 0.2%. Test data to establish this type of limit are given by Wellinger, et al, references 8.133, 8.134, and 8.135. Rose[8.117] also gives data for permanent strain limit and discusses its significance. Comparisons of AD-Merkblatt-B9 with limit-pressure theory are given in References 8.146 and 8.147.

In evaluating this data for branch connections in piping, the question arises as to both the definition of "yield pressure" or "limit pressure" and their significance. The problem of definition arises because, even for a material with a sharp yield point, the displacements or strains in a branch connection depart only gradually from the elastic behavior. One must arbitrarily choose some displacement that defines the yield or limit pressure. The problem of significance arises because the yield or limit pressure does not indicate the ultimate pressure capacity of the branch connection; the burst pressure may be higher than the limit pressure by a factor of up to 3 or more.

## 8.23 Static Pressure, Burst Pressure Determined

In the references cited in Table 8.1, one will find relatively little data on burst pressures; and none on ASA B16.9 tees. Such data as do exist indicate that for branch connections in cylinders with $\alpha = 0$, reinforced by "area-replacement" and if made of reasonably ductile material (including welds), the burst pressure will be essentially the same as the unperforated shell. For unreinforced tees and laterals, in which the branch and run pipe are of the same schedule number, an empirical equation(8.115) for estimating the burst pressure is:

$$\frac{P_b}{P_{bn}} = 1 - \frac{d}{D} (1 - 0.7 \sin^{1.5} \alpha) \tag{8.1}$$

where $P_b$ = burst pressure of branch connection

$P_{bn}$ = calculated burst pressure of unperforated run pipe

$d$ = branch diameter

D = run diameter

$\alpha$ = lateral angle, see Figure 8.1 ($0 \leq \alpha \leq 60°$).

The test data were obtained on carbon steel such as A106 Gr B for diameter-to-thickness ratios of about 50 or less. The equation should not be used beyond the indicated limits.

Tees conforming to ASA B16.9 and B16.11 are required by these standards to be capable of sustaining a pressure equal to the calculated burst pressure of the pipe of the designated schedule number and material. This requirement is discussed in Chapter 7, pp 17-20. Some burst test data on B16.11 components are given in Reference 8.73. No published data on burst pressures of B16.9 tees are available; however, the writer is aware of the existence of considerable amount of such data by one particular manufacturer--all showing burst pressures higher than required by ASA B16.9.

With regard to the relationship between yield pressure and burst pressure, one notes that for straight pipe this ratio is about the same as the ratio of yield strength to ultimate strength of the material. For B16.9 tees, the ratio of yield strength to ultimate strength is probably an upper bound to the ratio of yield pressure to burst pressure. That is, for A-106-B, the ratio of yield strength to ultimate strength is typically around 0.55. The ratio of yield pressure to burst pressure of B16.9 tees probably would not be more than 0.55 and might be (depending on how yield pressure is defined and the nominal dimensions of tee) as low as 1/3.

## 8.24 Cyclic Pressure, Cycles to Produce Fatigue Failure

Cyclic pressure fatigue tests are useful in that they indicate locations of high stress concentration. Usually there is good correlation between the location of failures produced in fatigue tests and the location of service failures. Most cyclic pressure fatigue tests were run with the upper pressure limit well above the maximum pressure expected in service, hence there is seldom a direct correlation between the test data and service experience or predicted service life.

The majority of the available test data on branch connections was developed at five organizations. These are:

Ecole Polytechnique, Montreal, Canada

Tube Turns/Battelle

British Welding Research Association

University of Ghent, Belgium

Southwest Research Institute.

The available data from these five organizations will be discussed in the following five sections.

### 8.241 Ecole Polytechnique

In 1951 PVRC initiated a research program at Ecole Polytechnique, Montreal, Canada. Results obtained from the program are given by Dubuc and Welter[8.65] (1956), Welter and Dubuc[8.137] (1957) and Welter and Dubuc[8.138] (1962). The first test specimens consisted of cylinders 12" I.D. x 3/4" wall, ~ 36" long, with closures. Cylinders were fabricated from rolled and welded A-201-A or A-302-B steel plate. Branch connections (nozzles) consisted of 1.25" I.D. x 0.375" wall pipe. A total of 12 vessels with nozzles were fatigue-tested, each of which contained at least two branch connections.

In addition to the data on nozzles, References 8.65, 8.137, and 8.138 contain data on fatigue strength of other details, e.g., flat plate closures (which had to be abandoned because they were so poor), elliptical heads, girth welds to the heads, longitudinal welds in the cylinders, notches in the surface of the cylinder, plug weld repairs, branch connections in a hemispherical head, insert-patch welds, etc. These extraneous test failures are not well documented in the References, however, in some ways they may be more significant than the data obtained on the nozzle failures. For example, it appears from the test results that patching a hole in a pressure vessel, either by plug welding or by an insert patch-plate, may create a much weaker point in the vessel than a nozzle.

### 8.242 Tube Turns/Battelle

At about the same time as the Ecole Polytechnique tests were started, Tube Turns (Division of Chemetron) initiated a series of cyclic pressure tests to determine the relative merit of various types of branch connections used in gas transmission lines. Later (1955), Battelle Memorial Institute (Columbus) sent to Tube Turns a series of seven reinforced branch connections which Battelle had fabricated and determined stresses using SR-4 strain gages with static internal pressure loading. These were subjected to cyclic pressure loading at the Tube Turns cyclic pressure test facilities. Results of these tests are partially contained in a paper by Markl, George, and Rodabaugh[8.102]; the entire test series results are given in a later paper by Rodabaugh and George[8.115]. The results of the Battelle/Tube Turns tests are also contained in Reference 8.108.

These tests involved thinner-wall run pipe and larger branches than were used in the Ecole Polytechnique tests. The run pipe D/T was in the range of 68 to 77; branch-to-diameter ratio, d/D from 0.19 to 0.58. The test series included some 50 test specimens. The types of specimens of general interest were:

Straight pipe with a longitudinal weld, 22" diameter, 5/16" wall

USAS B16.9 tee, 22" x 12" x 10", 5/16" nominal wall

Saddle reinforced branch connections

Pad reinforced branch connections

Drawn outlets

Unreinforced branch connections.

Test specimen closures consisted of USAS B16.9 caps; these are ellipsoidal shaped with an axis ratio of 2:1. No failures were encountered in these caps nor in the girth welds thereto.

### 8.243 British Welding Research Association (BWRA)

The BWRA tests were run on 20" I.D. x 1" wall run pipe with 6.875" I.D. x 0.375" wall branch pipe. The first set of results is given by Lane[8.89] on different weld details and flush vs. inwardly protruding branches. The second set of test results is given by Lane and Rose[8.90] on flush and inwardly protruding nozzles and on various pad reinforcements — outside pad only, inside pad only or a combination of outside and inside pad. These tests include about 60 test specimens. S-N curves are shown in Reference (8.90).

The nominal stress range (PD/2T) to produce failure in $10^5$ cycles was not very sensitive to reinforcing or weld details. The nominal stress range at $10^5$ cycles was between 16,000 and 22,000 psi for all variants. The pseudo-stress range* for carbon steel material at $10^5$ cycles is about 100,000 psi, implying a fatigue stress intensification factor of 6.2 to 4.5 for the various types of branch connections tested.

Cyclic pressure test results on a "welding tee" are given by Lane and Rose(8.89). The tee is identified as an 8" Schedule 80; presumably it is equivalent to a USAS B16.9 tee of that nominal size and schedule. The tee failed (in the crotch) at 590,000 cycles. Part of the cycles were at 70 to 1500 psi and part at 70 to 2000 psi. Assuming all cycles at 70 to 2000 psi, the nominal stress range (PD/2T) was 15,700 psi. The pseudo-stress range for carbon steel material at N = 590,000 cycles (including mean stress effect) is about 60,000 psi, indicating a fatigue stress intensification factor of about 3.8.

## 8.244 University of Ghent

One test series was made on 22.05" I.D. x 0.781" wall run pipe. Branch pipes were 8.89" I.D. x 0.47" wall. Tests were run on:

Straight pipe -- i.e., no nozzles

Branch reinforced by a pad

Branch reinforced by locally increased thickness of the branch pipe

Branch reinforced as above, plus an inward protuberance of the nozzle.

---

* The pseudo-stress range is given by the product $E\epsilon$ where E is the modulus-of-elasticity, $\epsilon$ is the strain range in a strain controlled fatigue test on the material. The data for carbon steel material is taken from Figure 9 of Reference 8.149. Additional comparisons of this type are shown in Table A8.3 of APPENDIX A.

Another test series was made on 9.41" I.D. x 0.547" wall run pipe. Branch pipes were 1.97" I.D. x 0.276", 0.393", or 0.511" wall.

Results of both of the above test series are given by Soete, et al.[8.123].

Another test series was run on "inclined branch connections"; the results are given by Heirman and Stockman[8.77]. The test specimens were intended to be about half-scale in comparison to the test specimens of Reference 8.123. The dimensions of the run pipe were 11.2" I.D. x 0.413" wall, and of the branch pipes 4.33" I.D. x 0.67" wall.

The types of test specimens were

$\alpha = 0$; i.e., an unreinforced tee

$\alpha = 30°$, an unreinforced lateral

$\alpha = 60°$, an unreinforced lateral.

Unreinforced, hillside branch; side of nozzle tangent to side of pipe, branch axis offset about 1.0 d.

Unreinforced, hillside branch; branch axis offset about 0.5 d.

### 8.245 Southwest Research Institute (SWRI)

The PVRC initiated cyclic pressure tests at the Southwest Research Institute in 1958. The test work was guided by PVRC and jointly sponsored by PVRC and the USAEC. Test results are given in a series of progress reports written during the period October, 1959, to the present time. Pickett and Grigory[8.113] give a summary of cyclic pressure tests on full-size vessels. In addition, a few cyclic pressure fatigue tests were made on "half-scale" pressure vessels.

The test series on full-size vessels included 8 vessels; all with a number of nozzles. The cylindrical shell of the vessels was 36" I.D. x 2" wall thickness. Vessels of A-201-B, A-302-B, T1 and 2-1/4 Cr - 1 Mo materials were tested. A variety of nozzle designs, both in the cylindrical shell and in the hemispherical end closures, were included.

## 8.3 External Loads, Theory

The set of external loads considered are shown in Figures 8.1 and 8.2. The loads are considered as being resultants of uniformly distributed shear stresses (for force loads) or linearly varying normal stresses (for moment loads) in the attached pipes. For force loads, the distance along the pipe at which the force is applied is highly significant. In principle, the force load can be considered as producing a moment and a shear load at some convenient reference point; e.g. the center lines intersection. By comparing the stress field produced by a pure moment with that produced by a force, that part of the stress due to shear can be isolated. To the extent that superposition holds, the stresses due to any combination of the external loads can be obtained if stresses due to each of the individual loads are known.

For external loads, the displacements of the branch connection may be significant because the flexibility of the connection enters into the calculation of forces in the piping system; where such forces arise from thermal expansion of the piping or from movements of equipment attached to the piping. The flexibility of piping components is given in the USAS piping codes as "flexibility factors". No useable factor is given for branch connections; a common assumption is that the run and branch pipe extend to the center lines intersection, and that the intersection is rigid. This assumption is probably conservative for static loading; it may not be conservative for dynamic loading.

## 8.31 Branches in Pipe, Theory

At the present time, there are no theoretical methods available for calculation of stresses or displacements due to any of the external loads shown in Figure 8.1. A theory for $M_5$ has been developed by Dr. A. C. Eringen (General Technology Corporation) and that theory is being programmed for a computer at Oak Ridge National Laboratory. The theory is based on the intersection of two uniform-wall cylindrical shells, with d/D limited to about 1/3. When the programming is completed, adequate theoretical guidance should be available for out-of-plane loading on the branch of uniform-wall tees with $d/D < 1/3$.

For external loads applied to the branch and for connections with d/D limited to about 1/3, the theory developed by Bijlaard[8.142-8.144] along with empirical modifications thereto by Wichman, et. al.[8.145] gives some guidance. This theory is for distributed loads over a small, rectangular area of a cylinder. The resultants of the distributed loads can be proportioned to give $M_4$, $M_5$, $F_4$, $F_5$, or $F_6$. To the extent that the branch-pipe stiffness is equivalent to the stiffnes of the material removed from the run pipe by the branch, this theory might be expected to give some indication of stresses and displacements in the run pipe. It cannot, of course, give any stresses in the branch pipe.

The computer program for out-of-plane bending, based on Eringen's theory, will not be applicable to B16.9 tees because the d/D-ratios are greater than 1/3 and also because the branch-run intersection is locally reinforced and includes significant transition radii. Similarly, even for $d/D < 1/3$ but with local reinforcement, the theory will not be directly applicable. Finite-element computer programs may eventually provide a theoretical solution. The problem of non-radial nozzles in cylindrical shells with external loads would presumably also be amenable to finite-element or finite-difference computer programs.

A shell-type solution for laterals and/or hillside connections with external loadings would seem to be within the state-of-the-art; however, no suitable computer program has been developed insofar as the writers are aware. Bijlaard's work may be of some significance in this area.

Adequate theories for limit external loads (analogous to limit pressures) have not been developed, nor are elastic-plastic analyses available -- for either a tee or non-radial branch connection.

## 8.32 Branches in Closures, Theory

As for internal pressure, if the branch is located in a closure so that geometric symmetry exists, available analytical methods can be applied. An additional complication of non-symmetry of loading exists, however, this can also be handled at least in the elastic regime. It is not known whether computer programs have been developed for elastic-plastic or limit analysis with non-symmetric loads. The references cited under the discussion of the theory for internal pressure loading are at least partially applicable to external loads. Some comparisons of computer programs results (References 8.31, 8.33, and 8.34) with each other and with test data are given in Reference 8.6.

## 8.4 External Loads, Test Data

Table 8.1 includes references giving test data on external loads. Four types of results are available:

(1) Static Loads, measured strains (converted to stresses)

(2) Static Loads, measured displacements (convertible to flexibility factors)

(3) Static Loads, gross yielding or limit load determined

(4) Cyclic Loads, cycles to produce fatigue failure.

These four types of test results are discussed in the following four sections.

### 8.41 Static External Loads, Measured Stresses

The earliest known data on stresses due to external loads were published by Schoessow and Kooistra[8.119] in 1945; these being on relatively small branches in a cylindrical shell. Insofar as the three major programs discussed under internal pressure loading (Section 8.21), the following summary may be made.

(1) PVRC Program

Ref. No.

8.62 Cornell University (Cranch) tests on thin-wall cylinder with various branch connections and attachments. Tests aimed at correlation with Bijlaard theory.

8.63 Cornell University (Dally) tests on spherical shells with various radial nozzles. (Internal pressure loading not included.)

8.75, 8.76 Penn State (Hardenbergh). Part of the models used for internal pressure were also used for external load tests; loads being applied to the nozzle.

8.114 IIT (Riley). Models were subjected to thrust and moment loads applied to the branch.

8.128 University of Illinois (Taylor & Lind). No external load tests

8.94 Westinghouse (Leven). Includes tests on five nozzles in spherical shells with moment load on nozzle.

8.103/ 8.106 University of Tennessee (Maxwell & Holland). Moment and force loads applied to nozzles in spherical shells.

8.74 ORNL (Greenstreet). Moments and forces applied to nozzles in spherical shell.

(2) American Gas Association

Includes external loads on branches, see APPENDIX A.

(3) British Welding Research Association

Test data obtained by the BWRA includes some stresses due to external loads applied to branches; including data on the probable equivalent of an ASA B16.9 tee. Results of most of this work are given in various reports by Lane and Rose (References 8.85 through 8.88 and 8.90).

In evaluating data on stresses due to external loads, the following points might be noted:

(1) Essentially no data exist for standard tees (USAS B16.9, B16.11, B16.5).

(2) Data exist only for external loads applied to the branches. None of the references appear to attach any significance to the reaction loads and, in fact, a careful study of the photographs or illustrations is necessary to determine what reactions were used. For small d/D-ratios, the reaction load details may not be significant; for large d/D the reaction loads probably are significant.

(3) For models with fillet welds, the maximum stress is usually associated with the toe of the fillet weld. Strain gage results do not show this stress, however its significance is shown by cyclic bending fatigue tests.

## 8.42 Static External Loads, Measured Displacements

As indicated by the check marks in the "K" column of Table 8.1, some data exist from which flexibility factors for branch connections can be deduced. Most of this data is evaluated in References 8.7 and 8.9. The conclusion reached in these references was that, for small nozzles in thin-wall shells attached to relatively stiff piping systems, ignoring the flexibility of the branch connection could lead to overestimates of external loads on the nozzle by a factor of ten or greater.

## 8.43 Static External Loads, Gross Yielding or Limit Load Determined

Test data in this area appear to be practically nonexistent. One example is shown in Reference 8.115 in which a static load corresponding to a nominal bending stress in the branch pipe of 73,000 psi was applied* resulting in large (15° rotation) but not unbounded displacement of the branch pipe.

## 8.44 Cyclic External Loads, Cycles to Produce Fatigue Failure

Stress intensification factors given in ASA B31.1-1955 are based on cyclic fatigue tests by Markl[8.101] and his generalization of the test results. As indicated in Table 8.1, some additional test data of this type has become available since publication of Reference 8.101.

Markl's tests were run almost entirely on 4" nominal size straight tees. Stress intensification factors for reducing tees were only vaguely defined in ASA B31.1-1955. ASA B31 Code Case 53 (July, 1963)** gives more specific rules for stress intensification factors for reducing tees. Evaluation of most of the data indicated in Table 8.1, along with comparisons of later (than Markl's) data with B31.1 and Code Case 53, is contained in Reference 8.8.

Some correlations between stress intensification factors, as defined by Markl, and maximum measured stresses can be made. As discussed in Reference 8.8, these correlations indicate that maximum measured stresses are approximately double the stress indicated by Markl's stress intensification factors.

## 8.5 Combination of Pressure & Moment Loads

If a complete stress field for a given branch connection were available for pressure loading and for each external load, and if superposition were applicable, then the stress field due to any combination of loads could be obtained.

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* Yield strength of material was about 40,000 psi.

** This Code Case is now incorporated in USAS B31.1.0-1967, Power Piping.

There is no indication from either test data or theory that superposition of stresses (or small displacements) due to external loadings is not accurate. At present, knowledge of complete stress fields for any loadings is quite limited; however, a better knowledge of maximum stresses due to the individual loadings does exist. As discussed in Reference 8.6, there may be some conservatism involved in assuming superposition for combined pressure and external loads. Accordingly, a conservative approach is to assume that maximum stresses due to various loads coincide in location and direction.

## 8.6 Summary

### 8.61 Theories

The status of theory for elastic stresses and displacements of branch connections may briefly be summarized as follows:

| | |
|---|---|
| (1) Branches in closures with $\alpha = 0$ (Geometric symmetry about branch centerline) | Theory is adequate for both pressure and external load and for both uniform wall and local reinforcing |
| (2) Branches in cylinders with $\alpha = 0$ (shell theories) | |
| (a) Pressure, uniform wall shells | Eringen theory for $(d/D)\sqrt{D/T}$ to about 1.1 |
| (b) External load, $M_5$ uniform wall shells | Computer program being written based on Eringen theory |
| (c) Other external loads on branch, $d/D < \sim 1/3$ | Bijlaard theory, for lack of anything more applicable |

(3) Finite-element or finite difference computer programs for asymmetric structures apparently have not yet been developed to the point where they can be used for branch connections such as a B16.9 tee.

## 8.62 Test Data

It is obvious from Table 8.1 that many significant contributions have been made to establish the load-carrying characteristics of branch connections and that our knowledge of such characteristics has advanced greatly in the past 15 years. From the standpoint of pipeline branch connections, however, there are large gaps in available information.

Almost all branch connections in critical-service pipelines in nominal sizes 4" and larger and $d/D > 0.5$ are made with ASA B16.9 tees. Very little test data exist for such tees; this constitutes probably the most significant gap in available test data.

Steel tees with socket-welded or threaded ends (B16.11) are used in small size pipelines (4" and smaller). Available test data are restricted to some burst tests. These tests, and examination of the dimensions of such tees, indicate that failure due to rated pressure loadings is highly unlikely* and that failure due to external loads is most likely to occur at the juncture between fitting and pipe; at the threads or fillet weld between fitting and pipe. Data on these kinds of joints have been obtained from fatigue tests by Markl[8.101].

* Assuming absence of defects; a manufacturing and inspection problem rather than a dimensional design problem.

Flanged fittings (B16.5) are probably never used in present-day critical piping systems and, because of economic considerations, are seldom used in any piping. The significant aspects of B16.5 are more related to the flanges (See Chapter 12) and valve bodies (See Chapter 11). Accordingly, there is no apparent need for test data on B16.5 fittings.

With regard to small d/D branch connections, available data are particularly inadequate with respect to bending loads applied to the branch.

# APPENDIX A

## CHAPTER 8

## BATTELLE-COLUMBUS TESTS ON BRANCH CONNECTIONS

The American Gas Association sponsored a series of tests at Battelle during the period 1952 through 1962. These results can be classified under four groups:

(1) Unreinforced Branch Connections, Static Loads

(2) Reinforced Branch Connections, Static Loads

(3) Reinforced Branch Connections, Cyclic Pressure

(4) Reinforced and Drawn Outlet Branch Connections, Cyclic Moments

In the following, a description of the test specimens and test loads included in these four groups of tests are abstracted from the A.G.A. or Battelle reports.

1. Unreinforced Branch Connections, Static Loads

Dimensions and identification numbers of the eight test specimens are shown in Table 8A.1. All test specimens were made from commercially available carbon steel pipe. Fillet welds were kept small so as to minimize reinforcing by that weld.

The first series of tests were run on the three test specimens with 24" O.D. x 312" wall run pipe; specimen numbers U1, U2 and U3 of Table 8A.1. Strain gages (13/16" gage length) were placed on the outside surface only. Each test specimen had roughly 50 strain rosettes (three gage). These gages were all placed in one quadrant of the test specimens, along $\phi = 0, 25°, 45°, 60°$, and $90°$. Loadings, in this first test series, consisted of:

(a) Internal pressure

(b) In-plane moment ($M_4$ of Figure 8.1)

(c) Out-of-plane moment ($M_5$ of Figure 8.1)

Results of this first series of tests are covered in detail in Reference 8.148 (Sept. 30, 1953).

The second series of tests were made on specimen numbers U4 through U8 of Table 8A.1. Strain gages (1/4" gage length) were placed on both inside and outside surfaces along $\phi = 0$ and $\phi = 90°$. Loading, in this second test series, consisted of internal pressure only.

Results of this second test series are covered in detail in Reference 8.51 (Feb. 19, 1960).

TABLE 8A.1: DIMENSIONS, DIMENSIONAL RATIOS AND MODEL IDENTIFICATION, UNREINFORCED TEST SPECIMENS

DIMENSIONS (inches)

| Run | Pipe | Branch Pipes and Model Numbers | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| O.D. | T | O.D. | t | No. | O.D. | t | No. | O.D. | t | No. |
| 24 | 0.312 | 4.5 | 0.237 | U1 | 12.75 | 0.250 | U2 | 24.00 | 0.312 | U3 |
| 24 | 0.281 | 6.625 | 0.250 | U4 | 12.75 | 0.250 | U5 | 18.00 | 0.250 | U6 |
| 24 | 0.375 | — | — | — | 12.75 | 0.375 | U7 | — | — | — |
| 24 | 0.687 | — | — | — | 12.75 | 0.625 | U8 | — | — | — |

DIMENSIONAL RATIOS

| $\frac{D}{T}$ | $\frac{d}{D}$ | $\frac{s}{S}$ | No. | $\frac{d}{D}$ | $\frac{s}{S}$ | No. | $\frac{d}{D}$ | $\frac{s}{S}$ | No. |
|---|---|---|---|---|---|---|---|---|---|
| 76 | .18 | .24 | U1 | .53 | .66 | U2 | 1.00 | 1.00 | U3 |
| 84 | .27 | .30 | U4 | .53 | .60 | U5 | .75 | .84 | U6 |
| 63 | — | — | — | .52 | .52 | U7 | — | — | — |
| 34 | — | — | — | .52 | .57 | U8 | — | — | — |

$s/S = (d/t)/(D/T)$

## 2. Reinforced Branch Connections, Static Loads

Dimensions and identification numbers of the ten test specimens are shown in Table 8A.2. All test specimens were made from commercially available carbon steel pipe and carbon steel reinforcements.

The first series of tests were run on test specimens with 24" O.D. x 0.312" wall run pipe; specimen numbers R1 through R7 of Table 8A.2. Strain gage rosettes (13/16" gage length) were placed on:

(a) Outside surface of pipes and reinforcement

(b) Outside surface of pipes, under reinforcement

(c) Inside surface of pipes.

Loadings, in this first test series, consisted of:

(a) Internal pressure

(b) In-plane moment ($M_4$ of Figure 8.1)

(c) Out-of-plane bending ($M_5$ of Figure 8.1)

Results of this first series of tests on reinforced connections is covered in Reference 8.107 (March 30, 1956). These seven reinforced test specimens were later sent to Tube Turns for cyclic pressure testing.

The second series of tests were run of test specimens with 24" O.D. x 0.281" wall run pipe; specimen numbers R8, R9, and R10 of Table 8A.2. Strain gages (1/4" gage length) were placed on the outside and inside surfaces of the pipes along $\phi = 0$ and $\phi = 90°$. Loadings, in this second test series, consisted of internal pressure only.

Results of this second series of tests on reinforced connections is covered in Reference 8.52 (Jan. 30, 1961).

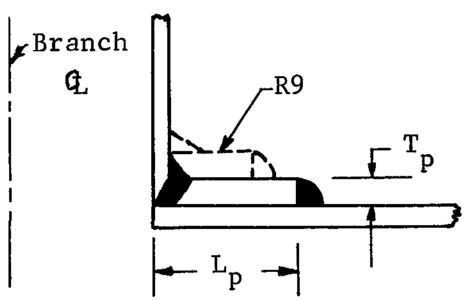
TABLE 8A.2: DIMENSIONS, DIMENSIONAL RATIOS AND MODEL IDENTIFICATION, REINFORCED TEST SPECIMENS

PIPE DIMENSIONS (inches)

| Run Pipe | | Branch Pipes and Model Numbers | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| O.D. | T | O.D. | t | Reinf. | No. | O.D. | t | Reinf. | No. | O.D. | t | Reinf. | No. |
| 24 | 0.312 | 4.50 | 0.237 | Pad | R1 | 8.625 | 0.250 | Pad | R2 | 12.75 | 0.250 | Pad | R3 |
| ↓ | ↓ | " | " | Saddle | R4 | " | " | Saddle | R5 | ↓ | ↓ | Saddle | R6 |
| | | -- | -- | -- | -- | -- | -- | -- | -- | | | Sleeve | R7 |
| 24 | 0.281 | 6.625 | 0.250 | Double Pad | R8 | 12.75 | 0.250 | Pad | R9 | 18.00 | 0.250 | Sleeve & Pad | R10 |

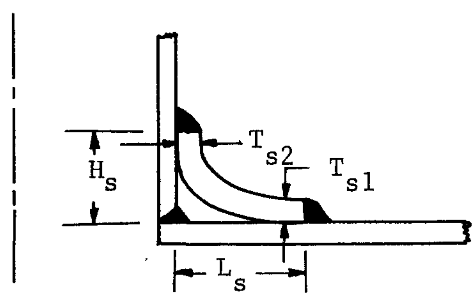
REINFORCEMENT DIMENSIONS (inches)

Pads



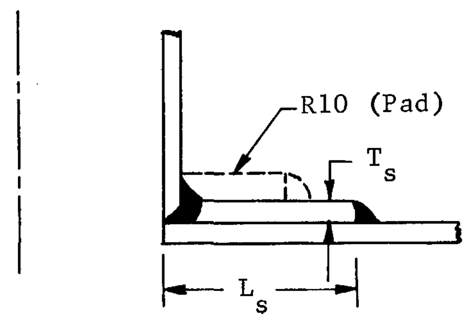

| No. | $T_p$ | $L_p$ | |
|---|---|---|---|
| R1 | 3/8 | 1-7/8" | R9 |
| R2 | 3/8 | 3-13/16" | Top Pad |
| R3 | 3/8 | 6- 1/8" | $T_p$ = .281 |
| R8 | .281 | 4.5" | $L_p$ = 2.91 |
| R9 | .281 | 5.4" | |

Saddles




| No. | $T_{s1}$ | $T_{s2}$ | $L_s$ | $H_s$ |
|---|---|---|---|---|
| R4 | 11/32 | 13/32 | 2-13/16 | 2 |
| R5 | 1/2 | 7/16 | 4- 9/16 | 3 |
| R6 | 7/16 | 1/2 | 5- 3/4 | 3-1/2 |

Sleeve




| No. | $T_s$ | $L_s$ |
|---|---|---|
| R7 | 3/8 | 6-1/8 |
| R10 | .281 | 5.42 |

R10, pad
$T_p$ = .281, $L_p$ = 3.0

3. Reinforced Branch Connections, Cyclic Pressure

The first series of tests were run on test specimens R1 through R7 of Table 8A.2. The pressure was cycled from 900 to 1550 psi, corresponding to roughly 55 to 95 percent of the calculated yield pressure of the unperforated run pipe. Results of this test series are given in Reference 8.108 (April, 1967).

The second test series was carried out as a cooperative effort by the T. D. Williamson Co., the American Gas Association and Battelle. Eight branch connections were tested; all except one were 16"x16"x8" branch nominal size. The run pipe was 16"O.D. x 0.312" for all eight specimens. One butt-welding straight tee and two saddle reinforced connections were tested, the remaining five test specimens had some type of complete encirclement reinforcement. The pressure was cycled from 1000 to 1800 psi, corresponding to roughly 50 to 90 percent of the calculated minimum yield pressure of the unperforated run pipe. Detailed test specimen descriptions and results are given in Reference 8.50 (December 30, 1958).

A "peak stress intensification factor", $i_p$, can be derived from the cyclic pressure tests by the procedure* shown in the footnotes to Table 8A.3. These $i_p$-factors may appear to be quite high, in view of the fact that all of the test specimens would meet the usual code (except ASME Section III) rules for reinforcement of openings. However, the strain gage test results seem to imply the same magnitudes of peak

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* Application of the procedure may be debatable because the primary stress was greater than one-third the yield strength and the secondary stress range, by implication, was greater than twice the yield strength. However, no evidence of racheting appears in the data given in the references.

TABLE 8A.3: FATIGUE STRESS INTENSIFICATION FACTORS DERIVED FROM CYCLIC PRESSURE TESTS

| Spec. No. | $\frac{D}{T}$ | $\frac{d}{D}$ | Reinforcing | $\Delta P$, psi | $N_p$ | $S_N$, psi | $S_v$, psi | $i_p$ |
|---|---|---|---|---|---|---|---|---|
| R1 | 76 | .18 | Pad | 650 | 47,500 | 62,000 | 12,400 | 5.0 |
| R2 | | .35 | Pad | | 31,200 | 70,000 | | 5.6 |
| R3 | | .53 | Pad | | 12,700 | 100,000 | | 8.1 |
| R4 | | .18 | Saddle | | 34,500 | 68,000 | | 5.5 |
| R5 | | .35 | Saddle | | 22,200 | 80,000 | | 6.5 |
| R6 | | .53 | Saddle | | 22,200 | 80,000 | | 6.5 |
| R7 | | .53 | Sleeve | | 10,000 | 110,000 | | 8.9 |
| 1 | 50 | 1.00 | Tee* | 800 | >302,000* | --- | 10,000 | --- |
| 2 | | | CER** | | 37,700 | 68,000 | | 6.8 |
| 3 | | | CER | | 8,300 | 120,000 | | 12.0 |
| 4 | | | CER | | 27,100 | 75,000 | | 7.5 |
| 5 | | | CER | | 8,000 | 120,000 | | 12.0 |
| 6 | | | CER | | 64,800 | 58,000 | | 5.8 |
| 7 | | | Saddle | | 7,000 | 140,000 | | 14.0 |
| 8 | | .53 | Saddle | | 67,000 | 58,000 | | 5.8 |

$\Delta P$ = pressure range in cyclic pressure test
$N_p$ = cycles-to-failure (leakage) in cyclic pressure test
$S_v$ = variable nominal stress amplitude in cyclic pressure test, $S_v = [\Delta PD/2T] \times 0.5$
$S_N$ = stress amplitude for carbon steel material for $N_p$ cycles, from Fig. 9 of Ref. 8.149.
$i_p$ = fatigue-effective stress intensification factor, $i_p = S_N/S_v$

* Butt welding tee, fatigue failure in attached pipe
** CER ≡ complete encirclement reinforcement, see Reference 8.50 for details

stresses. That is, for pad or saddle reinforced connections, if one extrapolates the stresses to the toe of the fillet weld at $\phi = 90°$, and then multiplies that value by a factor of two for the local stress at the toe of the fillet weld**, the resulting estimated peak stress is in the same "ball park" as those $i_p$-factors shown in Table 8A.3.

There are two significant points indicated by this analysis:

(1) In thin-wall run pipe, fully reinforced to ASME Section VIII or piping code rules, peak stresses in the range of 5 to 15 times the nominal stress in the run pipe may exist.

(2) In thin-wall run pipe, pad or saddle reinforced branch connections, maximum peak stresses are more likely to occur at $\phi = 90°$ at the pipe-reinforcement juncture, rather than at the inside corner at $\phi = 0$.

A similar comparison can be made for at least one unreinforced connection. The unreinforced connection specimen number U2 (D/T = 76, d/D = 0.53, s/S = 0.66) had a maximum measured stress ratio ($\sigma_{max}/S$) of 2.27. From the shape of the stress vs $\rho$ , one might judge that the stress extrapolated to the toe of the intersection weld would be about double the measured stress, and further multiplying by a factor of 2 for the notch at the weld leads to an estimated peak stress of 4 x 2.27 = 9.1. Cyclic pressure tests given in Reference 8.115 on three roughly comparable unreinforced connections (D/T=68, d/D=.47,

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** With one exception, all fatigue failures (in pad or saddle reinforced connections) occurred at $\phi = 90°$ in the run pipe at the toe of the fillet weld between run pipe and reinforcement. In the one exception(Specimen No. R3), the failure occurred in the intersection weld at $\phi = 15°$; apparently starting from a ripple on the inside surface of the weld at that point.

s/S = 0.31) produced fatigue failures in 4700, 5300, and 17,700 cycles for the three specimens. The value of $S_v$ (see Table 8A.3) was about 11,000 psi. Using an average life $N_p$ = 9000, the corresponding value of $S_n$ is about 120,000 psi. The value of $i_p$ is then 120,000/11,000 = 10.9 as compared to the estimated peak stress from strain gage data of 9.1.

Fatigue failures of the three unreinforced specimens all occurred at the intersection weld, one at Ø = 0, one at Ø = 90°, and the third had simultaneous failures at Ø = 0 and Ø = 90°. This also is in agreement with the strain gage results in that maximum stresses were about the same for all values of Ø.

4. Reinforced and Drawn Outlet Branch Connections, Cyclic Moments

Reinforced test specimens were made from 16" O.D. x 0.500" wall run pipe, 6.625" O.D. x 0.280" wall branch pipes. Reinforcing consisted of a pad with $T_p$ = 0.500", $L_p$ = 3" (See Table 8A.2 for definition of $T_p$ and $L_p$) or of a commercial 16" x 6" saddle. Drawn outlets were of two types: (1) drawn from 16" x 0.500" wall pipe; (2) drawn from 16" x 1.00" wall pipe. All material was carbon steel.

Loadings consisted of:

(1) In-plane moment ($M_4$ of Figure 8.1)

(2) Out-of-plane moment ($M_5$ of Figure 8.1)

(3) Static internal pressure combined with (1) or (2) above.

A significant number of each of the four types of specimens were tested in order to develop S-N curves. Detailed results are given in Reference 8.110 (May, 1962). The results are discussed in relationship to piping code stress intensification factors in Reference 8.8.

# 8. References


8.1 Waters, E. O., "Reinforcement of Openings in Pressure Vessels", Welding Journal Research Supplement, 1958.

8.2 Mershon, J. L. "PVRC Research on Reinforcement of Openings in Pressure Vessels", Welding Research Council Bulletin No. 77 (May, 1962).

8.3 Langer, B. F., "PVRC Interpretive Report of Pressure Vessel Research, Section 1, Design Considerations, Chapter 1.5, External Loading" Welding Research Council Bulletin No. 95 (April, 1964).

8.4 Mershon, J. L., Reference 8.3, Chapter 1.6, "Reinforcement of Openings Under Internal Pressure".

8.5 Mershon, J. L. "Preliminary Evaluation of PVRC Photoelastic Test Data on Reinforced Openings in Pressure Vessels", Welding Research Council Bulletin No. 113.

8.6 Rodabaugh, Witt & Cloud, "Stresses at Nozzles in Spherical Shells Loaded with Pressure, Moment or Thrust", Phase Report No. 2, Battelle-Columbus to USAEC, July 15, 1966.

8.7 Rodabaugh and Atterbury, "Flexibility of Nozzles in Spherical Shells" Phase Report No. 3, Battelle-Columbus to USAEC, June 28, 1966.

8.8 Rodabaugh and Atterbury, "Stresses at Nozzles in Cylindrical Shells Loaded with Pressure, Moment or Thrust", Phase Report No. 5, Battelle-Columbus to USAEC, Dec. 22, 1967.

8.9 Rodabaugh and Atterbury, "Flexibility of Nozzles in Cylindrical Shells", Phase Report No. 6, Battelle-Columbus to USAEC, Dec. 22, 1967.

8.10 Beskin, L., "Strengthening of Circular Holes in Plates Under Edge Loads", ASME J. of App. Mechs., 66, p. A-140 (1944).

8.11 Waters, E. O., "Theoretical Stresses Near a Cylindrical Opening in a Flat Plate with a Cylindrical Outlet", Welding Research Council Bulletin No. 51 (June, 1959)

8.12 Lourye, A. I., "Concentration of Stress in the Vicinity of an Aperture in the Surface of a Circular Cylinder", ASTIA AD250308 (Nov., 1960).

8.13 Withum, D., "The Cylindrical Shell with a Circular Hole under Torsion", Ingr.-Arch., 26, 435-446 (1958).

8.14 Eringen, Naghdi and Thiel, "State of Stress in a Circular Cylindrical Shell with a Circular Hole", Welding Research Council Bulletin No. 102 (Jan. 1962).

8.15 Lekkerkerker, J. G., "Stress Concentration Around Circular Holes in Cylindrical Shells", Proc. App. Mech. Conference, Munich, Germany, 1964.

8.16 Savin, G. N., "Concentration of Stresses Around Curvilinear Holes in Plates and Shells", Proc. App. Mech. Conference, Munich, Germany, 1964.

8.17 Van Dyke, Peter, "Stresses About a Circular Hole in a Cylindrical Shell", AIAA Journal, 3 (9) (Sept., 1965).

8.18 Reidelbach, W., "The State of Stress at the Perpendicular Intersection of Two Right Circular Tubes", Ingenieur-Archiv., 30 (5) 293-246 Translated by M. W. Stamisic, General Technology Corp., Tech. Note No. 3-1 (April, 1962).

8.19 Eringen, Naghdi, Mahmood, Thiel and Ariman, "Stress Concentrations in Two Normally Intersecting Cylindrical Shells Subject to Internal Pressure", General Technology Corp., Tech. Report No. 3-9 (Jan., 1967).

8.20 Lind, N. C., "Approximate Stress Concentration Analysis for Pressurized Branch Pipe Connections", ASME Paper No. 67-WA/PVP-7.

8.21 Bijlaard, Dohrman, and Wang, "Stresses in Junction of Nozzle to Cylindrical Pressure Vessel for Equal Diameter of Vessel and Nozzle", Nuclear Engineering and Design, 5, 349-365 (1967).

8.22 Tabakman, H. D., "A Numerical Solution of the Stresses at the Intersection of Two Cylindrical Shells", Aero-jet General Corp., Azuza, Calif., Report No. 3222, May, 1966.

8.23 FORMAT II, See "FORMAT-II-Second Version of Fortran Matrix Abstraction Technique", by J. Pickard, et. al., Douglas Aircraft Co., AFFDL-TR-66-207, Volumes I, II, III and IV.

8.24 SAMIS, See "Summary of the Functions and Capabilities of the Structural Analysis and Matrix Interpretive System Computer Program", by T. E. Lang, NASA Technical Report 32-1075, April, 1967.

8.25 CSMTRX, See "Space-Vehicle Stabilized-Platform Gimbel-System Weight-Reduction Study, Phase 1, Design of Ring Gimbels, Battelle Report to NASA, Dec. 28, 1962.

8.26 PAPA, See "PAPA Structural Analysis of Plates and Shells Using Trapezoidal and Triangular Plates Elements", by E. L. North, General Electric Co., Report GEAP-5471, March, 1967.

8.27 Hodge, P. G., "Full-Strength Reinforcement of a Cut-out in a Cylindrical Shell", J. of App. Mech., Dec. 1964.

8.28 Coon, Gill and Kitching, "A Lower Bound to the Limit Pressure of a Cylindrical Pressure Vessel with an Unreinforced Hole", Int. J. Mech. Engr. Science, February, 1967.

8.29 Cloud and Rodabaugh, "Approximate Analysis of the Plastic Limit Pressure of Nozzles in Cylindrical Shells", ASME Paper No. 67-WA/PVP-4.

8.30 Waters, E. O., "Stresses Near a Cylindrical Outlet in a Spherical Vessel", Welding Research Council Bulletin No. 96 (May, 1964).

8.31 Bijlaard, P. P., "Stresses in Spherical Vessels from External Moments Acting on a Pipe", Welding Research Council Bulletin No. 49, April, 1959.

8.32 Penny and Leckie, "Solutions for the Stresses at Nozzles in Pressure Vessels", Welding Research Council Bulletin No. 90, September, 1963.

8.33 Moore and Witt, "CERL-II, A Computer Program for Analyzing Hemisphere-Nozzle Shells of Revolution with Axisymmetric and Unsymmetric Loadings, Oak Ridge National Laboratory, ORNL-3817, Oct. 1965.

8.34 MOLSA, See "Analysis of Shells of Revolution Subjected to Symmetrical and Nonsymmetrical Loads, by A. Kalnins, ASME J. of App. Mech., Sept. 1964.

8.35 SEAL-SHELL-2, See "A Computer Program for the Stress Analysis of a Thick Shell of Revolution with Axisymmetric Pressure, Temperatures and Distributed Loads", by C. M. Friedrich, Bettis Atomic Power Lab., Report WAPD-TM-398.

8.36 USA, See "Gas-Cooled Reactor Program Semiannual Progress Report for Period Ending March 31, 1965", by Trauger and Whitman, Oak Ridge National Lab., USAEC Report ORNL-3807.

8.37 NONLIN, See "On Nonlinear Analysis of Elastic Shells of Revolution by A. Kalnins and J. F. Lestingi, J. App. Mech., Trans. ASME, Series E, Vol. 34, No. 1, pp 59-64, (March, 1967).

8.38 SHOREF, See "On Free and Forced Vibration of Rotationally Symmetric Layered Shells", by A. Kalnins, J. App. Mech., Vol. 32, pp 941-943 (1965).

8.39 AXISOL, See "Structural Analysis of Axisymmetric Solids", by E. L. Wilson, AIAA Journal, Vol. 3, pp 2269-2274, (1965).

8.40 FEELAP, See "Elastic-Plastic Analysis of Two-Dimensional Stress Systems by the Finite Element Method", by P. V. Marcal and I. P. King, Int. J. Mech. Sci., Vol. 9, pp 143-155 (1967).

8.41 BASIC, See "A Study of Rotary-Shaft-Sealing Concepts for Pressurized Water-Reactor Applications, by Grieser, et.al., Battelle Memorial Institute, Report No. BMI-1676, June 30, 1964.

8.42 Cloud, R. L., "The Limit Pressure of Radial Nozzles in Spherical Shells", Nuclear Structural Engineering, 1, 403-413 (1965).

8.43 Dinno and Gill, "The Limit Analysis of a Pressure Vessel Consisting of the Junction of a Cylindrical and Spherical Shell", Int. J. Mech. Science, Vol. 7, 21-42 (1965).

8.44 Dinno and Gill, "Limit Pressure for a Protruding Cylindrical Nozzle in a Spherical Pressure Vessel", J. Mech. Eng. Sci., Vol. 7, No. 3 (1965).

8.45 Gill, S. S., "The Limit Pressure for a Flush Cylindrical Nozzles in a Spherical Pressure Vessel", Int. J. Mech. Science, Vol 6, 105-115 (1964).

8.46 Lind, N. C., "Plastic Analysis of Radial Outlets from Spherical Pressure Vessels", J. of Eng. for Ind., May, 1964.

8.47 Gerdeen, J. C., "Theoretical Analysis of the Plastic Collapse of Thin Shell Structures", Battelle Report, December, 1966.

8.48 Johnson, D. E., "Stresses in a Spherical Shell with a Non-radial Nozzle", ASME J. of App. Mech., Nov. 1966.

8.49 Corum, J. M., "A Theoretical and Experimental Investigation of the Stresses in a Circular Cylindrical Shell with an Oblique Edge", Nuclear Engineering and Design, Vol 3, 256-280 (1966).

8.50 Atterbury, Beall, McClure, Ver Nooy and Battisto, "Experimental Stress Analysis of Several Full-Opening Reinforced Branch Connections, Battelle Report to A.G.A., Dec. 30, 1958.

8.51 Atterbury, McClure, Roos, and Grover, "Branch Connections - Data for Design", Battelle Report to A.G.A., Feb. 19, 1960.

8.52 Atterbury, Vagins and McClure, "Branch Connections-Development of Rules for Design", Battelle Report to A.G.A., Jan. 30, 1961.

8.53 Barkow and Huseby, "Welded Tee Connections", Welding Research Council Bulletin No. 22, May, 1955.

8.54 Berman and Pai, "An Experimental Investigation of Stresses in an HY-80 Marine Boiler Drum", Welding Research Supplement, July, 1962.

8.55 Berman and Pai, "An Experimental Investigation of Stresses in an HY-80 Marine Boiler Drum with Added Attachments", Welding Research Supplement, January, 1963.

8.56 Berman and Pai, "Internal Pressure Cyclic Fatigue Test of an HY-80 Marine Boiler Drum", Welding Research Supplement, January, 1964.

8.57 Bernsohn, Burgreen and Cummins, "Experimental Evaluation of Austenitic Stainless Steel Piping for Sodium-Deuterium Reactor", Nuclear Development Corp. Report NDA 84-11, Aug. 29, 1958.

8.58 Blair, J. S., "Reinforcement of Branch Pieces", Engineering, July 5, 1946; Sept. 6, 1946; Nov. 29, 1946; Dec. 6, 1946; Dec. 13, 1956; Dec. 20, 1946; Dec. 27, 1946.

8.59 Clare and Gill, "Effect of Vessel Diameter/Thickness Ratio on the Behavior Beyond the Elastic Limit of Flush Nozzles in Cylindrical Pressure Vessels: Experimental Investigation", J. Mech. Eng. Sci., Vol 8, No. 4 (1966).

8.60 Cloud, R. L., "The Limit Pressure of Radial Nozzles in Spherical Shells", Nuclear Structural Engineering, Vol. 1 pp. 403-413 (1965).

8.61 Cottam and Gill, "Experimental Investigation of the Behavior Beyond the Elastic Limit of Flush Nozzles in Cylindrical Pressure Vessels", J. Mech. Eng. Sci., Vol 8, No. 3 (1966).

8.62 Cranch, E. T., "An Experimental Investigation of Stresses in the Neighborhood of Attachments to a Cylindrical Shell", Welding Research Council Bulletin No. 60, May, 1969.

8.63 Dally, J. W., "An Experimental Investigation of Stresses Produced in Spherical Vessels by External Loads Transferred by a Nozzle", Welding Research Council Bulletin No. 84, Jan. 1963.

8.64 Dinno and Gill, "Experimental Investigation into the Plastic Behavior of Flush Nozzles in Spherical Pressure Vessels", Int. J. Mech. Sci., Vol 7, p. 817 (1965).

8.65 Dubuc and Welters, "Investigation of Static and Fatigue Resistance of Model Pressure Vessels", Welding Research Supplement, July, 1956.

8.66 Ellyin, F., "An Experimental Study of Plastic Deformation of Cylinder/Sphere Intersecting Shells", University of Sherbrooke (Sherbrooke, Quebec, Canada), Technical Report No. FE-2-67, Sept. 1967.

8.67 Faupel and Harris, "Stress Concentration in Heavy-Walled Cylindrical Pressure Vessels, Effect of Elliptic and Circular Side Holes", Industrial and Engineering Chemistry, Vol 49, p 1979, Dec. 1957.

8.68 Fessler and Lewin, "Stress in Branched Pipes Under Internal Pressure", Proc. Instn. Mech. Engrs., Vol 176, No. 29 (1962).

8.69 Fessler and Lewin, "Stress Distribution in a Tee Junction of Thick Pipes", British Journal of Applied Physics, Vol 7, Feb. 1956.

8.70 Everett and McCutchan, "Investigation of Stress Conditions in a Full-Size Branch Connection", Trans. ASME, Vol 60, FSP-60-12 (1933).

8.71 Gross, N., "Researches on Welded Pressure Vessels and Pipelines", British Welding Journal, April, 1954.

8.72 Graalfs, H. E., "Stress Analysis on 90° Elbow with Branch Connection", Northern Natural Gas Co., Private Communication, Jan 20, 1965.

8.73 Greenwald, D. K., "Burst Tests of Some 1" and 2" Socket-Welded Pipe Fittings", Private Communication, April 28, 1967.

8.74 Greenstreet, Holland, La Verne, Maxwell, Shobe and Witt, "Experimental Stress Analysis of EGCR Pressure Vessel", Oak Ridge National Laboratory, ORNL-3157, Nov. 28, 1961.

8.75 Hardenbergh, Zamrik and Edmondson, "Experimental Investigation of Stresses in Nozzles in Cylindrical Pressure Vessels", Welding Research Council Bulletin 89, July, 1963.

8.76 Hardenbergh and Zamrik, "Effects of External Loadings on Large Outlets in a Cylindrical Pressure Vessel", Welding Research Council Bulletin No. 96, May, 1964.

8.77 Heirman and Stockman, "Essais sur Tubulures Inclinees (Tests on Inclined Nozzles)", Revue de la Soudure, No. 1, 1964, Translation in Welding Research Abroad, Dec., 1964.

8.78 Hiltscher and Florin, "The Stresses at Oblique Pipe Connections to Spherical Pressure Vessels", Konstruktion, 15, H. 11, 444-449, Translation in Welding Research Abroad, April, 1964.

8.79 Hiltscher and Florin, "Stress Distribution at an Oblique Nozzle in a Plane Plate in Tension", Swedish State Power Board, Report L-106, Sept., 1960.

8.80 Horseman, R. W., "Stresses in Oblique Nozzles on Pressure Testing of Reactor Pressure Vessels", Instn. Mech. Engrs., Pressure Vessel Research, 1961.

8.81 Kaufman, W. J., "Configuration of Nozzles in High Pressure Steam Drums of Low Alloy Steel, Welding Research Abroad, Oct. 1964.

8.82 Kitching and Duffield, "Stresses Due to Axial Loads and Internal Pressure on Forged Nozzles in a Spherical Pressure Vessel", Int. J. Mech. Sci., Vol 6, pp 77-103 (1964).

8.83 Kitching and Jones, "Effect of Bending Moments on Nozzles with Forged Transition, Pieces", Proc. Instn. Mech. Engrs., Vol 178 (Pt. 3), p 211 (1963-1964).

8.84 Kitching and Olsen, "Further Experiments with Forged Nozzles in Pressure Vessels", Proc. Instn. Mech. Engrs., Vol 179, Pt. 1, No. 29 (1964-1965).

8.85 Lane, P.H.R., "Stresses in a Welded Branch Connection; Part 1: Ratio Between Branch and Barrel five to six. British Welding Research Association Report FE.16/19/54 and FE. 16/20/54, March, 1954.

8.86 Lane, P.H.R., "Stresses in a Welded Branch Connection; Part II: Ratio Between Branch and Barrell One to One, British Welding Research Association Report FE.16/21/54, March, 1954.

8.87 Lane and Quartermaine, "Stresses in Unreinforced Branch Connections", British Welding Research Association Report FE.12/39/55 and FE. 16/37/55, April, 1955.

8.88 Lane, P.H.R., "Tests on Pipe Branch Connections", British Welding Research Association, Report No. FE.16/45/57, March, 1957.

8.89 Lane, P.H.R., "Pulsating Pressure Fatigue Tests on Pressure Vessel Branch Connections", British Welding Journal, Vol. 5, No. 7, July, 1958.

8.90 Lane and Rose, "Design of Welded Pipe Fittings", British Welding Research Association, Abington Hall, Abington, Cambridge, Published Nov. 1959.

8.91 Lane and Rose, "Comparative Performance of Pressure Vessel Nozzles Under Pulsating Pressure", Instn. Mech. Engrs., Pressure Vessel Research, 1961.

8.92 Lane and Rose, "Stress Analysis of Nozzles in Cylindrical Pressure Vessels", Instn. Mech. Engrs., Pressure Vessel Research, 1961.

8.93 Le Cocq, J., "Experimental Analysis of Stresses, Inlet Nozzles under External Loads", S.E.N.A. Repport A.DC. 352, Dec. 20, 1962.

8.94 Leven, M. M., "Photoelastic Determination of the Stresses in Reinforced Openings in Pressure Vessels", Welding Research Council Bulletin No. 113, April, 1966.

8.95 Leven, M. M., "Photoelastic Determination of Stresses at Oblique Openings in Spherical Pressure Vessels", Westinghouse Research Laboratories, Report 67-9D7-PHOTO-R2, Nov. 22, 1967.

8.96 Leven, M. M., "Photoelastic Determination of Stresses at an Opening in a Thin-Walled Cylindrical Pressure Vessel", Westinghouse Research Laboratories, Report 67-9D7-PHOTO-R1, Aug. 24, 1967.

8.97 Lind, Sherbourne, Ellyin and Dainora, "Plastic Tests of Two Branch Pipe Connections", U. of Waterloo Draft Report, May, 1967.

8.98 Lind and Palusamy, "Experimental Investigation of Intersecting Shells Under a Small External Force Disturbance", U. of Waterloo, Sept., 1967.

8.99 Mackenzie and Spence, "Oblique Nozzle Array in Spherical Shell", Instn. Mech. Engrs., Pressure Vessel Research, 1961.

8.100 Mantle and Proctor, "Stress Analysis of a Series of Single Oblique Nozzles", Instn. Mech. Engrs., Pressure Vessel Research, 1961.

8.101 Markl, A.R.C., "Fatigue Tests of Piping Components", ASME Trans., Vol. 74, No. 3 (1952).

8.102 Markl, George and Rodabaugh, "Pressure-Pulsation Tests of Branch Connections to Large-Diameter Pipe", American Gas Association Conference, May, 1955, Pittsburgh, Pa.

8.103 Maxwell and Holland, "Experimental Determination of Stresses in the Vicinity of a Pipe Attached to a Spherical Shell", U. of Tennessee, Engineering Experiment Station, Feb., 1957.

8.104 Maxwell and Holland, "Experimental Determination of Stresses in a Spherical Shell with Attached Pipe", U. of Tennessee, Engineering Experiment Station, April, 1959.

8.105 Maxwell, Holland and Cofer, "Experimental Stress Analysis of the Attachment Region of Hemispherical Shells with Radially Attached Nozzles", U. of Tennessee Engineering Experiment Station, June, 1965.

8.106 Maxwell and Holland, Experimental Stress Analysis of the Attachment Region of Hemispherical Shells with Radially Attached Nozzles, Part 2", U. of Tennessee, Engineering Experiment Station, April, 1967.

8.107 McClure, Sweeney and Grover, "Investigation of Stresses in Pipeline Branch Connections", Battelle-Columbus Report to A.G.A., March 30, 1956.

8.108 McClure, Abraham, Sweeney and Grover, "Branch Connections", American Gas Association PAR Report, April, 1957.

8.109 Mehringer and Cooper, "Experimental Determinations of Stresses in the Vicinity of Pipe Appendages to a Cylindrical Shell", S.E.S.A. Proceedings, Vol XIV, No. 2 (1957).

8.110 Mills, Atterbury and McClure, "Study of Effects of Cyclic Bending Loads on Performance of Branch Connections", Battelle Memorial Institute Report to the American Gas Association, May, 1962.

8.111 O'Toole, Rodabaugh and George, "Pressure Pulsation Tests of Reinforced Branch Connections", Tube Turns Report to Battelle Memorial Institute, 3/6/56.

8.112 Lemcoe, Pickett, Grigory, et.al., A series of Progress Reports, Tests on Pressure Vessels sponsored by the USAEC at Southwest Research Institute, 1959 to current date.

8.113 Pickett and Grigory, "Studies of the Fatigue Strength of Pressure Vessels, Part I. Cyclic Pressure Tests of Full Size Pressure Vessels", Welding Research Council Bulletin No. 135, November, 1968.

8.114 Riley, W. F., "Experimental Determination of Stress Distributions in Thin-Walled Cylindrical and Spherical Pressure Vessels with Circular Nozzles", Welding Research Council Bulletin No. 108, Sept. 1965.

8.115 Rodabaugh and George, "Welded Pipeline Branch Connections", ASCE Transactions, Vol. 125, Part I, 1960.

8.116 Rodabaugh E. C., "Cyclic Bending Tests of a Half-Scale Model of an 8"x 24" Saddle Reinforced Branch Connection", Tube Turns Report No. 8.011, August 25, 1953.

8.117 Rose, R. T., "Stress Analysis on Nozzles in Thin Walled Cylindrical Pressure Vessels", B.W.R.A. Report D3/22/63, March, 1964.

8.118 Rose, R. T., "Stress Analysis of a Single Oblique Nozzle", Instn. Mech. Engrs., Pressure Vessel Research, 1961.

8.119 Schoessow and Kooistra, "Stresses in a Cylindrical Shell Due to Nozzle or Pipe Connection", Trans. ASME, Vol. 67 (1945).

8.120 Schoessow and Brooks, "Analysis of Experimental Data Regarding Certain Design Features of Pressure Vessels", Trans. ASME, July, 1950.

8.121 Seabloom, E. R., "Bursting Pressure Tests of Welded Pipe Headers Show Need for Nozzle Reinforcements", Valve World, July-August, 1941.

8.122 Siebel and Schwaigerer, "Investigations into the Strength of Contoured Branch Pipe Connections", V.R.B. Report No. 1, 2nd ed., January, 1954.

8.123 Soete, Hebrant, Dechaene and Heirman, "The Reinforcement of Openings in Pressure Vessels", Revue de la Soudure, Vol 17, No. 1 (1961) Translation in Welding Research Abroad, March, 1962.

8.124 Stepanek, S., "Stress Concentrations in the Nozzle Ring of a Pressure Vessel", Nuclear Structural Engineering, Vol 2, pp 211-223 (1965).

8.125 Stockman, G., "Dimensioning the Reinforcement of Openings", Welding Research Abroad, June-July, 1966.

8.126 Stone and Hochschild, "The Effect of Nozzle Spacing on the Pressure Stresses at the Intersection of Cylindrical Nozzles and Shells", ASME Paper No. 66-WA/PVP-7.

8.127 Taylor and Waters, "The Effect of Openings in Pressure Vessels", ASME Transactions, 1934.

8.128 Taylor and Lind, "Photoelastic Study of the Stresses Near Openings in Pressure Vessels", Welding Research Council Bulletin No. 113, April, 1966.

8.129 Taylor, T. E., "Effect of Test Pressure on the Fatigue Performance of Mild Steel Cylindrical Pressure Vessels Containing Nozzles", British Welding Journal, March, 1967.

8.130 Taylor, T. E., "Fatigue Testing of an Al-Mg-Mn Alloy Pressure Vessel", British Welding Journal, February, 1967.

8.131 Townley, Proctor and Gadd, "Tests Beyond Yield on a Spherical Vessel Containing a Series of Oblique Nozzles, Applied Mechanics Convention, Newcastle upon Tyne, April, 1964, Instn. Mech. Engrs.

8.132 Watzke, J. T., "Stresses in a 6" x 24" Insert Type Branch Connection", El Paso Natural Gas Co., June 17, 1957.

8.133 Wellinger, Krageloh and Beckman, "Internal Pressure Experiments on Thick-Walled Branch Connections", Report from State Materials Testing Institute, Technische Hochschule Stuttgart (Date unknown, probably 1960).

8.134 Wellinger and Krageloh, "Stresses in Boiler Drums", Welding Research Abroad, October, 1966.

8.135 Wellinger, Schoch and Krageloh, "Elongation Measurement in Tests on a Dismantled Boiler Drum", Mtt. der VGB, Vol 107, pp 91-98 (April, 1967).

8.136 Wells, Lane and Rose, "Stress Analysis of Nozzles in Cylindrical Pressure Vessels", Instn. Mech. Engrs., Pressure Vessel Research, 1961.

8.137 Welters and Dubuc, "Fatigue Resistance of Simulated Nozzles in Model Pressure Vessels", Welding Research Supplement, June, 1957.

8.138 Welters and Dubuc, "Fatigue Resistance of Simulated Nozzles in Model Pressure Vessels of T-1 Steel", Welding Research Supplement, August, 1962.

8.139 Williams and Huler, "Unreinforced Openings in a Pressure Vessel", Welding Research Council Bulletin No. 51, June, 1959.

8.140 Winkler, Lowenberg and Pickett, "Experimental Investigation of Plastic Collapse Pressure for Pressure Vessel Models", Southwest Research Institute Report to Bettis Atomic Power Laboratory, SWRI project No. 03-1618, Sept. 15, 1965.

8.141 Wollering and Vazquez, "Design of Manifold Fittings for Special Temperature and Pressure Conditions", ASME Paper No. 56-PET-34.

8.142 Bijlaard, P. P., "Stresses from Local Loadings in Cylindrical Pressure Vessels", Trans. ASME, August, 1955.

8.143 Bijlaard, P. P., "Stresses from Radial Loads and External Moments in Cylindrical Shells Under Local Loadings", Welding Research Supplement, December, 1955.

8.144 Bijlaard, P. P., "Additional Data on Stresses in Cylindrical Shells Under Local Loadings", Welding Research Council Bulletin No. 50, May, 1959.

8.145 Wichman, Mershon and Hopper, "Local Stresses in Spherical and Cylindrical Shells Due to External Loadings", Welding Research Council Bulletin No. 107, August, 1965.

8.146 Cloud and Rodabaugh, "Proposed Reinforcement Design Procedure for Radial Nozzles In Spherical Shells with Internal Pressure, Phase Report No. 1, Battelle-Columbus to USAEC, March 31, 1966.

8.147 Rodabaugh and Cloud, "Proposed Reinforcement Design Procedure for Radial Nozzles in Cylindrical Shells with Internal Pressure", Phase Report No. 4, Battelle-Columbus to USAEC, Dec. 22, 1967.

8.148 Jackson, et.al., "Stresses in Unreinforced Branch Connections", Battelle-Columbus Report to American Gas Association, Sept. 30, 1953.

8.149 "Criteria of Section III of the ASME Boiler and Pressure Vessel Code for Nuclear Vessels", Published by ASME, 345 E. 47th St., New York, N. Y. 10017

8.150 ELAS, See "ELAS--A General-Purpose Computer Program for the Equilibrium Problems of Linear Structures", Jet Propulsion Laboratory Technical Report 32-1240, February 1968.

8.151 Zick, Crossett, and Lankford, "Destructive Tests of 9 Percent Nickel-Steel Vessels at -320 F", ASME Paper No. 62-WA-273.

CHAPTER 9

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# 9. REDUCERS

Typical types of reducers are shown in Figure 9.1. Butt-welding-end concentric and eccentric reducers are covered by USAS B16.9[(9.1)]. This standard includes reducers in large-end sizes from 3/4" to 24". Large-end sizes 26, 30, 34 and 36" are covered by MSS SP-48[(9.2)]. The small-end diameter is listed in these standards down to about one-half* of the large-end diameter. These standards give the overall length, minimum wall thickness throughout the body, diameters at the ends and a hydrostatic test requirement. The hydrostatic test requires that the reducer be capable of withstanding an internal pressure equal to the computed bursting pressure of the pipe with which it is designated to be used. These standards do not specify the shape of the reducers, except at the ends.

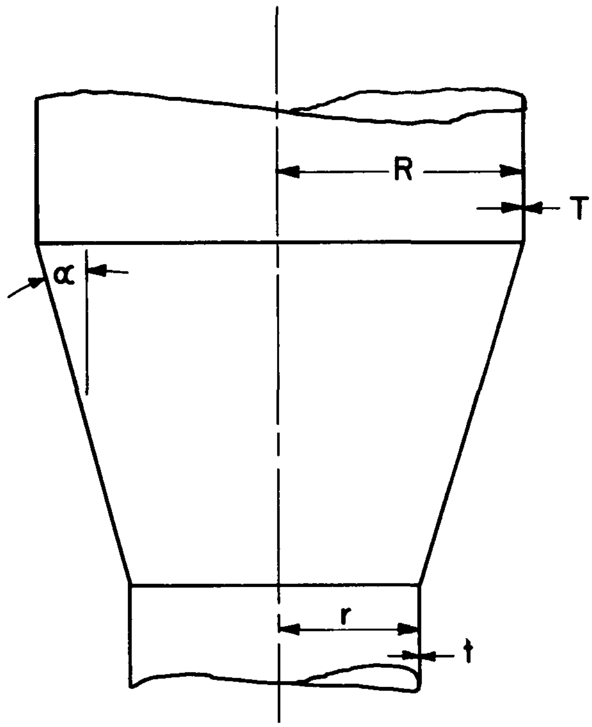
The design of concentric reducers is covered in the ASME Unfired Pressure Vessels Code[(9.3)], Par UG-36(e).

Reducers are also used with socket-weld ends or threaded ends, the later including threaded bushings. Design problems in these fittings are principally concerned with the pipe-to-fitting joint (see Chapters 6 and 13). Reducing flanges (see Chapter 12) are also used. These kinds of reducers are not discussed in this chapter.

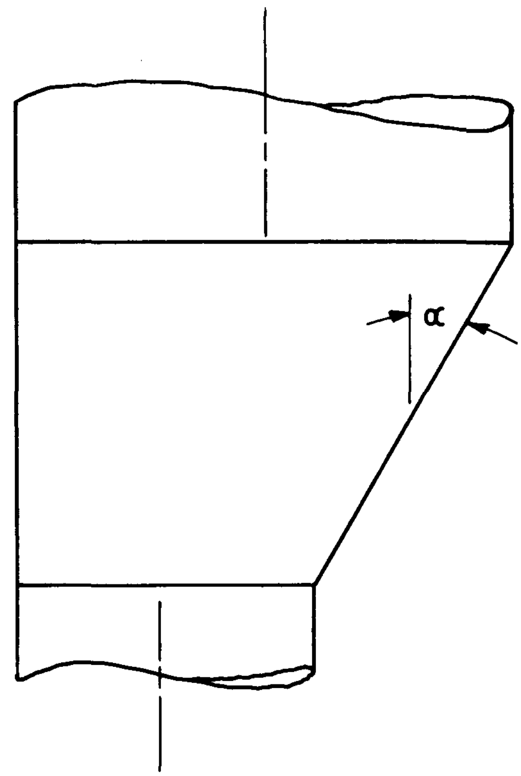
## 9.1 Manufacture of Reducers

Conical reducers of the types shown in Figure 9.1 (a) and (b) can be made by rolling a plate into a conical section. This results in a

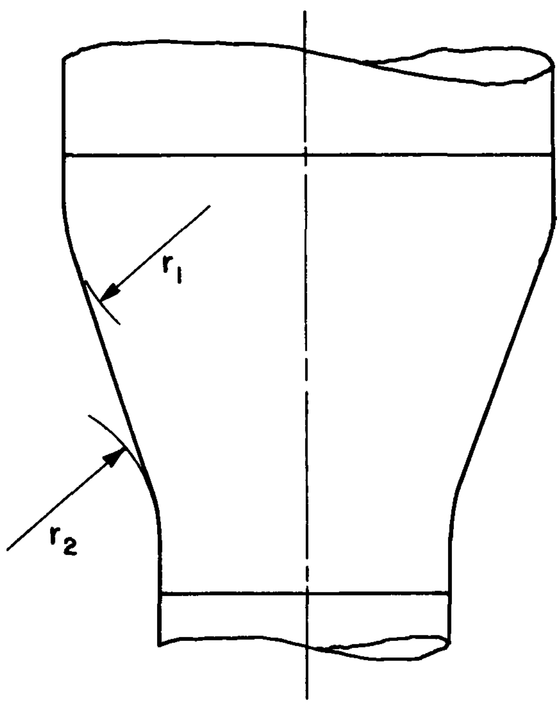
* When the small-end pipe diameter is considerably less than one-half of the large end diameter, the design is usually considered as a branch connection in a head. Branch connections are discussed in Chapter 8.



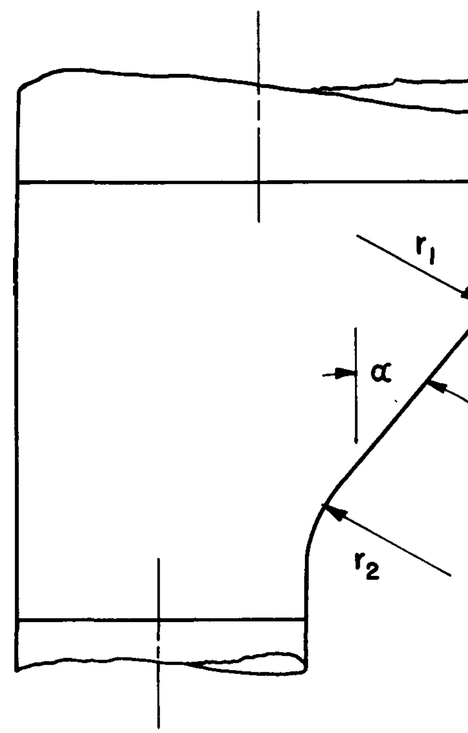

(a) Concentric, Conical




(b) Eccentric, Conical




(c) Concentric, with Transition Sections (Knuckles) and Tangents




(d) Eccentric, with Transition Sections (Knuckles) and Tangents

FIGURE 9.1 TYPES OF REDUCERS

reducer of constant wall thickness. While such reducers are permissible under B16.9 and SP-48, and are sometimes so furnished, the common sizes of reducers are not so constructed. Most butt-welding reducers are made by one of two processes.

(1) For smaller diameters and/or heavier walls

The reducer is machined out of bar stock. A typical cross section of a machined reducer is shown in Figure 9.2.

(2) For larger diameters and/or thinner walls

A die is made which has internal contours about the same as the desired external contour of the finished reducer. A length of pipe, of the large-end nominal size and wall thickness, is then heated and pushed into the die as shown in Figure 9.3. A "pull-ball" with external diameter equal to the internal diameter of the small end of the reducer, is then pulled through the pipe. The formed reducer, with wall thicknesses as illustrated in Figure 9.3, is then removed from the die, cut to the required length and welding bevels are machined on the ends.

There is no definite break-over point between the two manufacturing processes. However, the 1" (large-end) size would normally be made by process (1); the 4" (large-end) size and larger sizes would normally be made by process (2). For Grade B carbon steel, schedules 40 and 80, the 2" (large-end) size and larger sizes would normally be made by process (2).

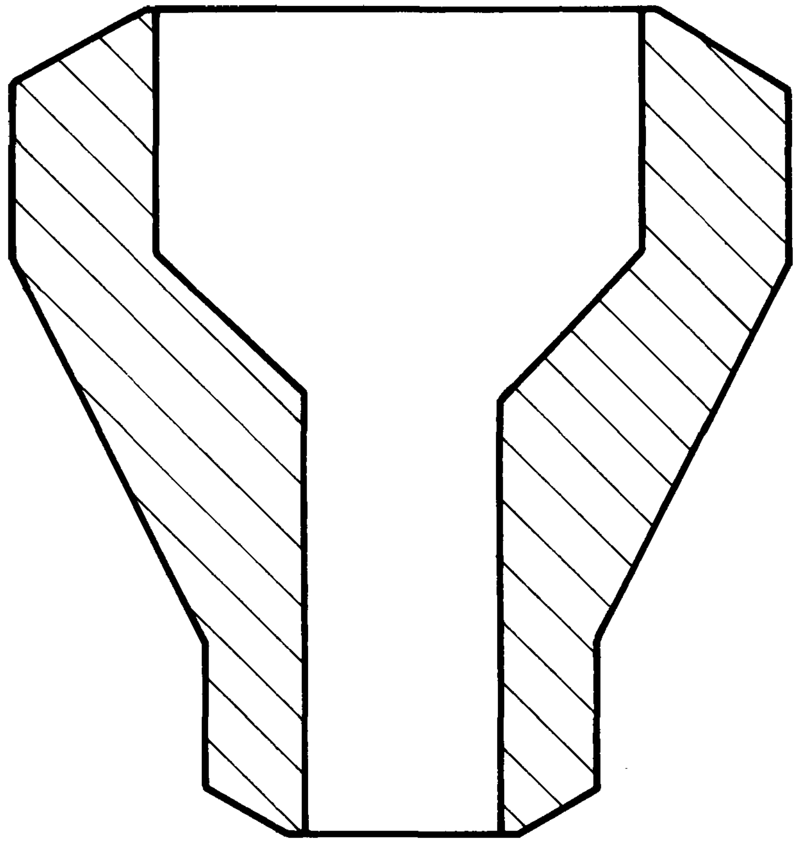


FIGURE 9.2 CROSS SECTION OF A TYPICAL MACHINED CONCENTRIC REDUCER

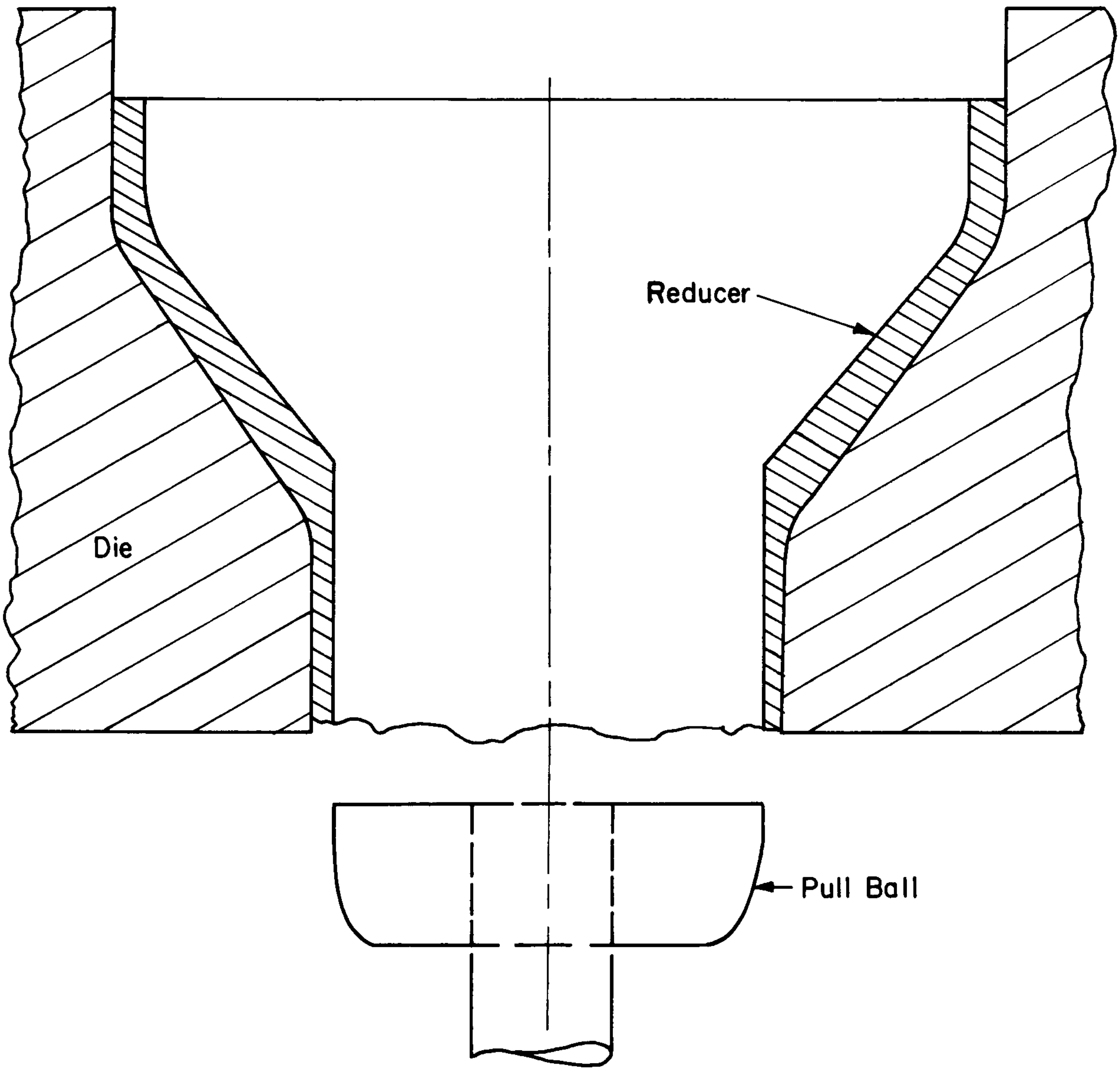



FIGURE 9.3 A METHOD OF MANUFACTURING CONCENTRIC REDUCERS, METHOD (2) OF TEXT

## 9.2 Internal Pressure, Theory

### 9.22 Concentric Reducers

Concentric reducers are axisymmetric structures, accordingly the elastic stresses can be calculated by using axisymmetric-shell or axisymmetric-body computer programs. Table 9.1 summarizes results of a few calculations on concentric, uniform wall, conical reducers using the MOLSA[9.4] shell computer program. The stresses are shown as stress indices where the nominal stress is that due to pressure in the large-end pipe; i.e., S = PR/T.

In the ASME Code[9.3], the following equation is given for the thickness of a conical portion of a concentric reducer.

$$t = \frac{PD}{2 \cos \alpha \ (SE - 0.6P)} \tag{9.1}$$

where t = minimum wall thickness of conical section

P = internal pressure

D = inside diameter at point under consideration

$\alpha$ = cone angle (see Figure 9.1)

S = allowable stress

E = weld joint efficiency.

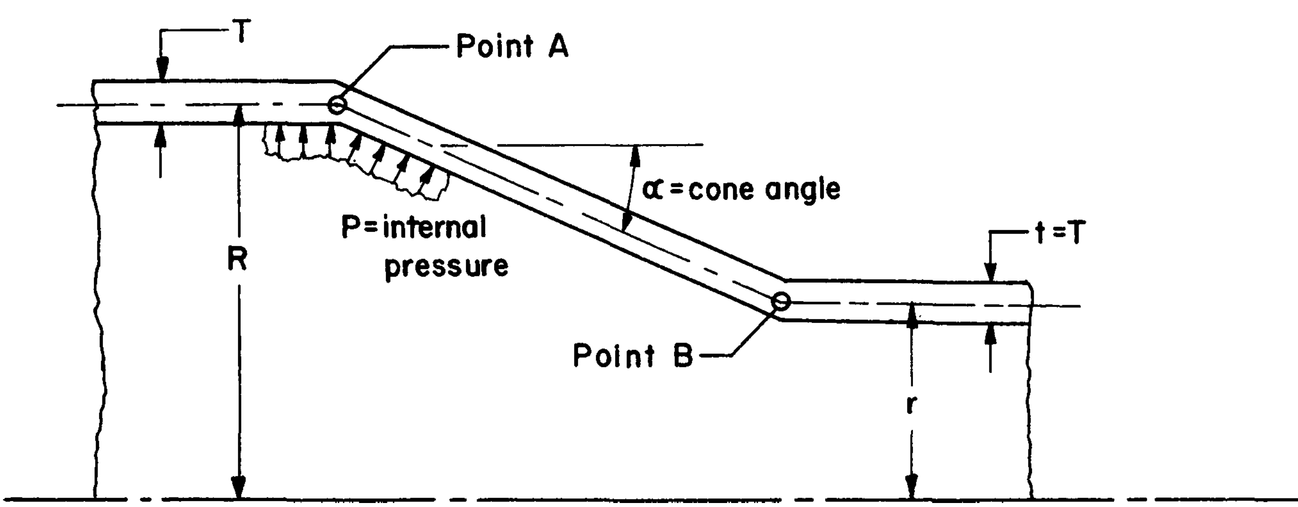
Equation (9.1) is based on the circumferential membrane stress in a conical shell, remote from end-effects. The reducer may be a simple conical shell section (Figure 9.1a) without a knuckle provided $\alpha$ is not

TABLE 9.1: CALCULATED STRESSES IN CONICAL, CONCENTRIC REDUCERS. r/R = 0.5, INTERNAL PRESSURE LOADING

| $\frac{T}{R}$ | Point | Stress (1) | $\sigma/S$ for $\alpha$ of 15° | 30° | 45° | 60° |
|---|---|---|---|---|---|---|
| 0.2 | A | $\sigma_{m\varphi}$ | 0.50 | 0.50 | 0.50 | 0.49 |
| | | $\sigma_{b\varphi}$ | 0.29 | 0.61 | 1.02 | 1.52 |
| | | $\sigma_{m\theta}$ | 0.80 | 0.60 | 0.43 | 0.32 |
| | | $\sigma_{b\theta}$ | 0.09 | 0.19 | 0.31 | 0.43 |
| | B | $\sigma_{m\varphi}$ | 0.25 | 0.25 | 0.25 | 0.25 |
| | | $\sigma_{b\varphi}$ | -0.07 | -0.15 | -0.20 | -1.17 |
| | | $\sigma_{m\theta}$ | 0.60 | 0.78 | 0.96 | 0.98 |
| | | $\sigma_{b\theta}$ | -0.03 | -0.04 | -0.16 | -0.35 |
| | | $\sigma_{max}$ | 0.79 | 1.11 | 1.52 | 2.01 |
| 0.025 | A | $\sigma_{m\varphi}$ | 0.50 | 0.50 | 0.50 | 0.51 |
| | | $\sigma_{b\varphi}$ | 0.96 | 2.00 | 3.27 | 5.08 |
| | | $\sigma_{m\theta}$ | 0.47 | -0.07 | -0.67 | -1.42 |
| | | $\sigma_{b\theta}$ | 0.29 | 0.60 | 0.97 | 1.50 |
| | B | $\sigma_{m\varphi}$ | 0.25 | 0.25 | 0.25 | 0.25 |
| | | $\sigma_{b\varphi}$ | -0.33 | -0.68 | -1.07 | -1.63 |
| | | $\sigma_{m\theta}$ | 0.71 | 0.96 | 1.32 | 2.12 |
| | | $\sigma_{b\theta}$ | -0.10 | -0.20 | -0.32 | -0.48 |
| | | $\sigma_{max}$ | 1.46 | 2.50 | 3.77 | 5.59 |

(1) $S = PR/T$
Subscripts: m = membrane, b = bending (+ for inside surface), $\varphi$ = axial, $\theta$ = hoop.

$\sigma_{max}$ = maximum surface stress.



greater* than 30°. Transition sections (knuckles) may consist of sections of toroidal, hemispherical or ellipsoidal shells. The thickness for such transition sections is determined by membrane stress equations for such shells, again ignoring end-effects and bending stresses. Finally, the ASME Code (9.3) recognizes the possibility of high stresses at the cone-to-cylinder juncture or at the transition sections and may require that a reinforcement ring be provided at either or both ends of the reducer.

The stress conditions at the large-end juncture of a reducer are comparable to those in conical or tori-conical heads, with internal pressure loading hence the theory of such heads is pertinent to the design of reducers. The presence of high stresses at the cone-cylinder juncture of heads has been known for many years. Boardmann[(9.5)], in 1944, analyzed a sharp intersection between a cone head and a cylindrical shell and proposed rules for compression ring reinforcements. The analogous problem in reducers is shown in Table 9.1 by the tabulated values of $\sigma_{m\theta}$ for the large end (Point A). As can be seen in Table 9.1, as $\alpha$ and R/T increases

---

* However, Par UA-5 (e) permits $\alpha > 30°$, provided a discontinuity stress analysis is made and that

$$\sigma_{mh} + \sigma_{dh} < 1.5\ SE \quad , \text{ and}$$

$$\sigma_{ml} + \sigma_{bl} < 4.0\ SE$$

where $\sigma_{mh}$ = membrane hoop stress

$\sigma_{dh}$ = average discontinuity hoop stress

$\sigma_{ml}$ = membrane longitudinal stress

$\sigma_{bl}$ = discontinuity longitudinal bending stress.

the value of $\sigma_{m\theta}$ decreases. For the worst case shown in Table 9.1, $\sigma_{m\theta}$ = -1.42S. An analogous situation exists at the small-end juncture as shown in Table 9.1 for $\sigma_{m\theta}$ at Point B. Here, as $\alpha$ and R/T increase, the value of $\sigma_{m\theta}$ increases; for the worst case of Table 9.1, $\sigma_{m\theta}$ = 2.12S. While the small end exhibits the larger value of $\sigma_{m\theta}$, the compressive stress at the large end may be of greater concern since it may cause local plastic buckling in the juncture region. For very thin shells, elastic buckling may be a problem.

For most typical concentric pipeline reducers, the high juncture stresses are not a problem because the value of $\alpha$ seldom exceeds 30° and T/R is seldom less than 0.025. One notes in Table 9.1 that, for $\alpha$ = 30°, T/R = 0.025, $\sigma_{max}$ = 2.50S. This is approaching the limit of 3 $S_m$ permitted in some codes for secondary bending stresses. Further, for conical reducers with a weld between cone and cylinder, this nominal stress of 2.50S coincides with a weld so that significantly higher peak stresses may occur due to weld irregularities. However, typical B16.9 reducers are usually furnished with a toroidal transition section and a tangent. The maximum stress is substantially reduced by the transition section. For example, for $\alpha$ = 30°, T/R = 0.025, a toroidal section with radius of 0.1R reduces the value of $\sigma_{max}$ from 2.50S to 1.26S.

The limit pressure load of concentric reducers can be calculated by such computer programs as CLPSHL(9.6). Elastic-plastic analysis may be made with such computer programs as FEELAP(9.7) or NONLEP(9.8). A recent paper by Gerdeen(9.9) discusses the status of plastic limit analysis of pressure vessels. The problem of collapse of heads due to internal pressure is closely related to the analogous problem in reducers. Papers by Shield and Drucker(9.10, 9.11) and Cloud(9.12) are pertinent in this respect.

### 9.22 Eccentric Reducers

Eccentric reducers pose a more difficult analytical problem. Presumably, accurate analysis of the stresses in such reducers can be obtained by use of finite-element computer programs which are not limited to axisymmetric structures (see Chapter 3). Until such programs are developed for practical use, a conservative approach would probably consist of assuming that the most eccentric profile existed everywhere. The axisymmetric computer programs could then be applied to that profile.

## 9.3 Moment Loading, Theory

### 9.31 Concentric Reducers

Elastic stresses for concentric reducers can be obtained by an axisymmetric shell program.* Table 9.2 gives the results of a few calculations using the MOLSA[9.4] program. The stresses are shown as stress indices where the nominal stress is that due to a bending moment applied to the small-end pipe; i.e., $S = M/\pi r^2 t$.

For moment loading, as might be expected, the high stresses occur at the small end of the reducer. As for pressure loading, for large $\alpha$ and R/T the membrane stresses in the hoop direction become significant. For $\alpha = 60°$, T/R = 0.025, $\sigma_{m\theta}$ at the small end is 3.57S. At the large end, $\sigma_{m\theta}$ =-1.59 S. The maximum stress occurs at point B (small end); it is an axial stress on the outside surface. As for pressure, the maximum stress is substantially reduced by a toroidal transition section. For example, for $\alpha = 30°$, T/R = 0.025, a toroidal section with radius of 0.1R reduces the value of $\sigma_{max}$ from 3.98 S to 1.86 S.

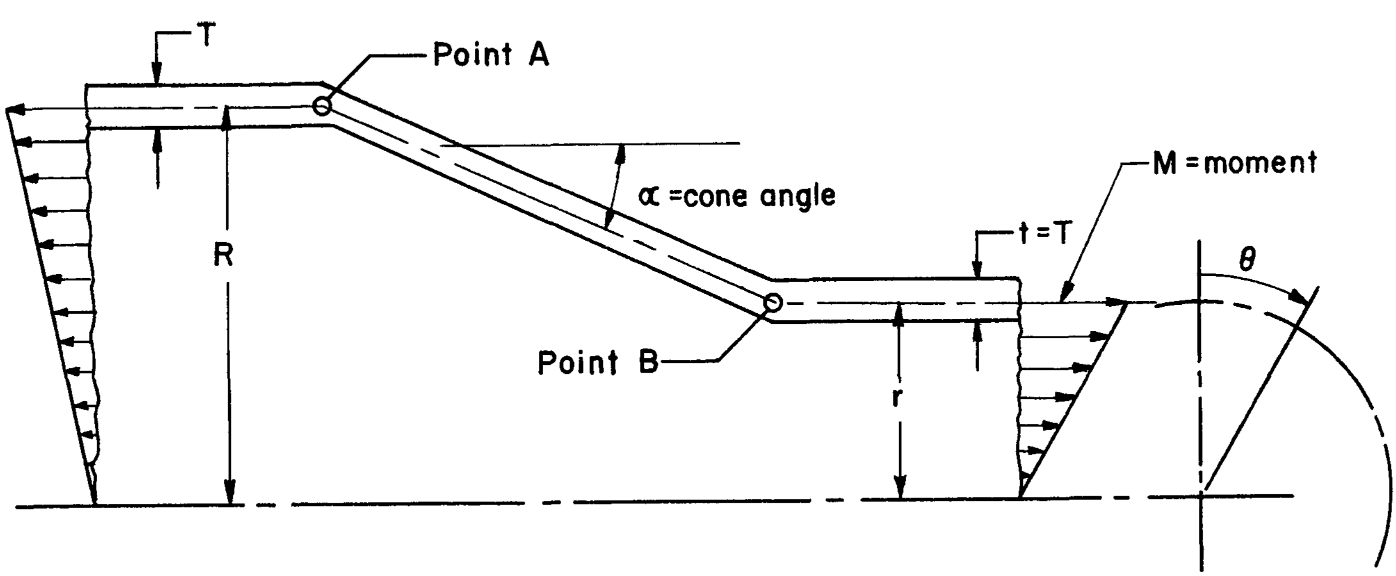
* MOLSA includes non-axisymmetric loadings in the form of Fourier series. The moment loading was modeled by using a boundary axial load proportional to cos $\theta$, where $\theta$ is shown in Table 9.2.

TABLE 9.2: CALCULATED STRESSES IN CONICAL, CONCENTRIC REDUCERS, r/R = 0.5, MOMENT LOADING

| $\frac{T}{R}$ | Point | Stress (1) | $\sigma/S$ for $\alpha$ of (2) 15° | 30° | 45° | 60° |
|---|---|---|---|---|---|---|
| 0.2 | A | $\sigma_{m\varphi}$ | 0.24 | 0.25 | 0.25 | 0.25 |
| | | $\sigma_{b\varphi}$ | 0.16 | 0.40 | 0.75 | 1.07 |
| | | $\sigma_{m\theta}$ | -0.08 | -0.18 | -0.24 | -0.18 |
| | | $\sigma_{b\theta}$ | 0.06 | 0.14 | 0.21 | 0.27 |
| | B | $\sigma_{m\varphi}$ | 0.98 | 0.97 | 0.97 | 0.97 |
| | | $\sigma_{b\varphi}$ | -0.49 | -1.07 | -1.71 | -2.37 |
| | | $\sigma_{m\theta}$ | 0.16 | 0.33 | 0.42 | 0.47 |
| | | $\sigma_{b\theta}$ | -0.18 | -0.42 | -0.70 | -0.97 |
| | | $\sigma_{max}$ | 1.47 | 2.04 | 2.68 | 3.34 |
| 0.025 | A | $\sigma_{m\varphi}$ | 0.25 | 0.25 | 0.26 | 0.28 |
| | | $\sigma_{b\varphi}$ | 0.50 | 1.04 | 1.21 | 2.78 |
| | | $\sigma_{m\theta}$ | -0.15 | -0.57 | -0.95 | -1.59 |
| | | $\sigma_{b\theta}$ | 0.27 | 0.32 | 0.53 | 0.85 |
| | B | $\sigma_{m\varphi}$ | 0.98 | 0.98 | 0.97 | 0.97 |
| | | $\sigma_{b\varphi}$ | -1.45 | -3.00 | -4.92 | -7.78 |
| | | $\sigma_{m\theta}$ | 0.72 | 1.48 | 2.37 | 3.57 |
| | | $\sigma_{b\theta}$ | -0.45 | -0.93 | -1.52 | -2.43 |
| | | $\sigma_{max}$ | 2.43 | 3.98 | -5.89 | 8.75 |

(1) $S = M/\pi r^2 t$.
Subscripts: m = membrane, b = bending (+ for inside surface), $\varphi$ = axial, $\theta$ = hoop.
$\sigma_{max}$ = maximum surface stress.

(2) Stresses shown at $\theta = 0$, stresses are proportional to $\cos \theta$.



## 9.32 Eccentric Reducers

As for pressure loading, the accurate analysis of eccentric reducers with moment loading is not presently feasible. Finite-element programs in development may provide the necessary analysis tool.

## 9.4 Test Data

The writer is not aware of any published test data on the performance characteristics of ASA B16.9 or MSS SP-48 reducers with internal pressure loading. Because of the method of manufacture of most B16.9 reducers (see Par 9.1), one would not expect any problem with static pressure loading. Reducers sold to B16.9 or SP-48 must be capable of withstanding a pressure equal to the calculated burst pressure of the mating pipe (presumably, the weaker of the large-end or small-end mating pipe). Manufacturers probably have run hydrostatic tests to assure that their reducers meet this requirement.

There are some published test data on heads with internal pressure loading. Kientzler and Borg[9.13] tested a cylindrical shell with two conical heads* with $\alpha = 45°$, T/R = 0.0058. One of the two cones was reinforced with a 2 x 2 x 3/8 angle at the cone-cylinder juncture. The other cone was essentially unreinforced at the cone-shell juncture. Strains were measured on both surfaces. Eventually, pressure was increased sufficiently to cause yielding of both the cylindrical shell and the cones. The test was stopped at 240 psi. At this pressure, the nominal

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* Actually, conical reducers. The cones terminated in man-ways at the small end.

stresses were:

In the cone, large end

$$S = \frac{PR}{t \cos \alpha} = \frac{240 \times 30}{0.175 \times 0.707} = 58{,}200 \text{ psi}$$

In the cylinder

$$S = \frac{PR}{T} = \frac{240 \times 30}{.120} = 60{,}000 \text{ psi}$$

The carbon steel material had a yield strength of 38,000 psi, tensile strength of 49,000 psi. At 240 psi, the tank had not failed either by rupture or by metal fracture. At the unreinforced end considerable deformation had taken place. The 45° juncture had assumed a curved surface. The junction with the reinforcing ring showed no deformation in the ring and slight bending in the cone. The cylinder bulged outward, starting to yield significantly at 170 psi.

Jones[9.14] tested cylindrical vessels with heads: (a) torispherical, (b) 2 to 1 ellipsoidal, (c) $\alpha$ = 45°, toriconical* and (d) $\alpha$ = 60°, toriconical*. The cylinder T/R was 0.0040. Average wall thickness of the heads were (a) 0.110, (b) 0.088, (c) 0.135, (d) 0.137. Plots of strain vs. pressure are shown. Yielding is not mentioned in the paper.

Markl[9.15] briefly discusses bending moment fatigue tests on 4 x 2 standard weight reducers (presumably concentric). He found a stress intensification factor of unity with respect to the fatigue strength of a typical butt weld in the 2" pipe. The failures (3 tests) all consisted of circumferential cracks at the edge of the attachment weld to the 2"

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* There were actually toriconical reducers with a man-way at the small end of the cone.

pipe or in the center of that weld. Accordingly, for this specific reducer ($\alpha \cong 25°$; T/R, large end, = 0.118; T/R, small end, = 0.165; transition section radii $\cong$ 0.75"; contour like Figure 9.3), the fatigue-effective stresses were less than those at the 2" pipe butt weld.

One notes in Table 9.2 that moment loading on a uniform wall, concentric reducer can produce significant stress intensification with respect to the nominal stress in the small-end pipe. For example, with $\alpha = 30°$, T/R = 0.025; $\sigma_{max} = 3.98$ S. Further, there is a weld at this point. Accordingly, the calculations indicate that a cyclic bending moment would produce a fatigue failure at the small end juncture. However, this table is not indicative of the stresses in typical B16.9 reducers such as illustrated in Figures 9.2 and 9.3

## 9.5 Summary

From a design standpoint, butt-welding end reducers may be divided into two classes.

(1) Typical B16.9 concentric reducers such as shows in Figures 9.2 and 9.3; with cone angles not greater than 30°, T/R not less than about 0.02, and with toroidal transition sections with radius not less than 0.1 of the large end radius.

Such reducers appear to be amply strong for their nominal pressure ratings and for cyclic moment loading.

(2) Conical reducers as shown in Figure 9.1 (a) with large $\alpha$ (e.g., $> 15°$), and small T/R.

Such reducers, based on exploratory calculations, may be subject to high stresses at the cone-to-cylinder junctures; either from pressure or moment loading. Plastic or elastic buckling may be a problem for extremes of $\alpha$ and T/R values. Further work is required to assign suitable stress indices for the B31.7 Code.

## 9. REFERENCES


9.1 Wrought Steel Butt-Welding Fittings, USAS B16.9, Published by the American Society of Mechanical Engineers, 345 E. 47th Street, New York, N. Y. 10017.

9.2 Steel Butt-Welding Fittings (26" and larger), MSS-SP-48, Published by the Manufactures Standardization Society of the Valve and Fittings Industry, 420 Lexington Avenue, New York, N. Y. 10017.

9.3 ASME Boiler and Pressure Vessel Code, Section VIII, Pressure Vessels, Division I, Published by the American Society of Mechanical Engineers 345 E. 47th Street, New York, N. Y. 10017.

9.4 MOLSA, See "Analysis of Shells of Revolution Subjected to Symmetrical and Nonsymmetrical Loads", by A. Kalnins, ASME J. of App. Mech., September, 1964.

9.5 Boardman, H. C., "Stresses at Junction of Cone and Cylinder in Tanks with Cone Bottoms or Ends", Water Tower, 1944 (Also published in ASME, Pressure Vessel and Piping Design, Collected Papers, 1927-1959).

9.6 CLPSHL, See "Theoretical Analysis of the Plastic Collapse of Thin Shell Structures", by J. C. Geerden, Battelle-Columbus, December, 1966.

9.7 FEELAP, See "Elastic-Plastic Analysis of Two-Dimensional Stress Systems by the Finite Element Method", by P. V. Marcal and I. P. King, Int. J. Mech. Science, Vol. 9, pp 143-155 (1967).

9.8 NONLEP, See "On Nonlinear Elastic-Plastic Analysis of Shells of Revolution", by J. C. Gerdeen, Battelle-Columbus Special Report (August, 1968).

9.9 Gerdeen, J. C., "Use of the Computer in the Plastic Limit Analysis of Pressure Vessels". To be presented at ASME Pressure Vessel and Petroleum Division Meeting, Dallas, September 26, 1968.

9.10 Shield, R. J. and Drucker, D. C., "Limit Strength of Thin-Walled Pressure Vessels with an ASME Standard Torispherical Head", Proceedings, Third U. S. National Congress of Applied Mechanics, ASME, 665-672 (1958).

9.11 Shield, R. J. and Drucker, D. C., "Design of Thin-Walled Torispherical and Toriconical Pressure Vessel Heads", J. App. Mechanics, Vol. 28, pp 292-297 (June, 1961).

9.12 Cloud, R. L., "Interpretive Report on Pressure Vessel Heads", Welding Research Council Bulletin No. 119 (January, 1967).

9.13 Kientzler, C. and Borg, S. F., "Observations of Strains Near Reinforced and Non-Reinforced Cone Cylinder Intersections", Welding Research Council Bulletin No. 46 (January, 1959).

9.14 Jones, E. O., "The Effects of Internal Pressure on Thin-Shell Pressure Vessel Heads", Welding Research Council Bulletin No. 69 (June, 1961).

9.15 Markl, A.R.C., "Fatigue Tests of Piping Components", Trans. ASME, Vol. 74, pp 287-303 (1952).

CHAPTER 10

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# 10. GIRTH TRANSITION JOINTS

The types of structures pertinent to this Chapter are:

(1) Joints involving a change in wall thickness as illustrated in USAS B31.7[10.1], Figure 1-727.3.1; included herein as Figure 10.1. This type of joint is encountered, for example, in welding pipe to butt-welding-end valves.

(2) Joints involving a fillet weld between pipe and threaded* or socket-welding valves, fittings or flanges, or between pipe and slip-on flanges.

(3) Butt-welded joints between pipes of equal thicknesses, but with an offset or misalignment of the mid-wall centerlines. This aspect is significant in relation to tolerances on pipe and piping components.

These types of joints are covered to some extent in Chapter 6, particularly from the standpoint of fatigue strength under cyclic bending loads. In this Chapter, these joints are discussed from a theoretical approach which gives some indication of stress levels with internal pressure or thermal gradient loadings, as well as for moment loadings. Additional pertinent test data (measured stresses) are also cited.

It might be noted that the problems involved in this Chapter include two of the 18 topics listed by the ASME Special Committee to Review Code Basis as research topics on which further information was needed; these are:

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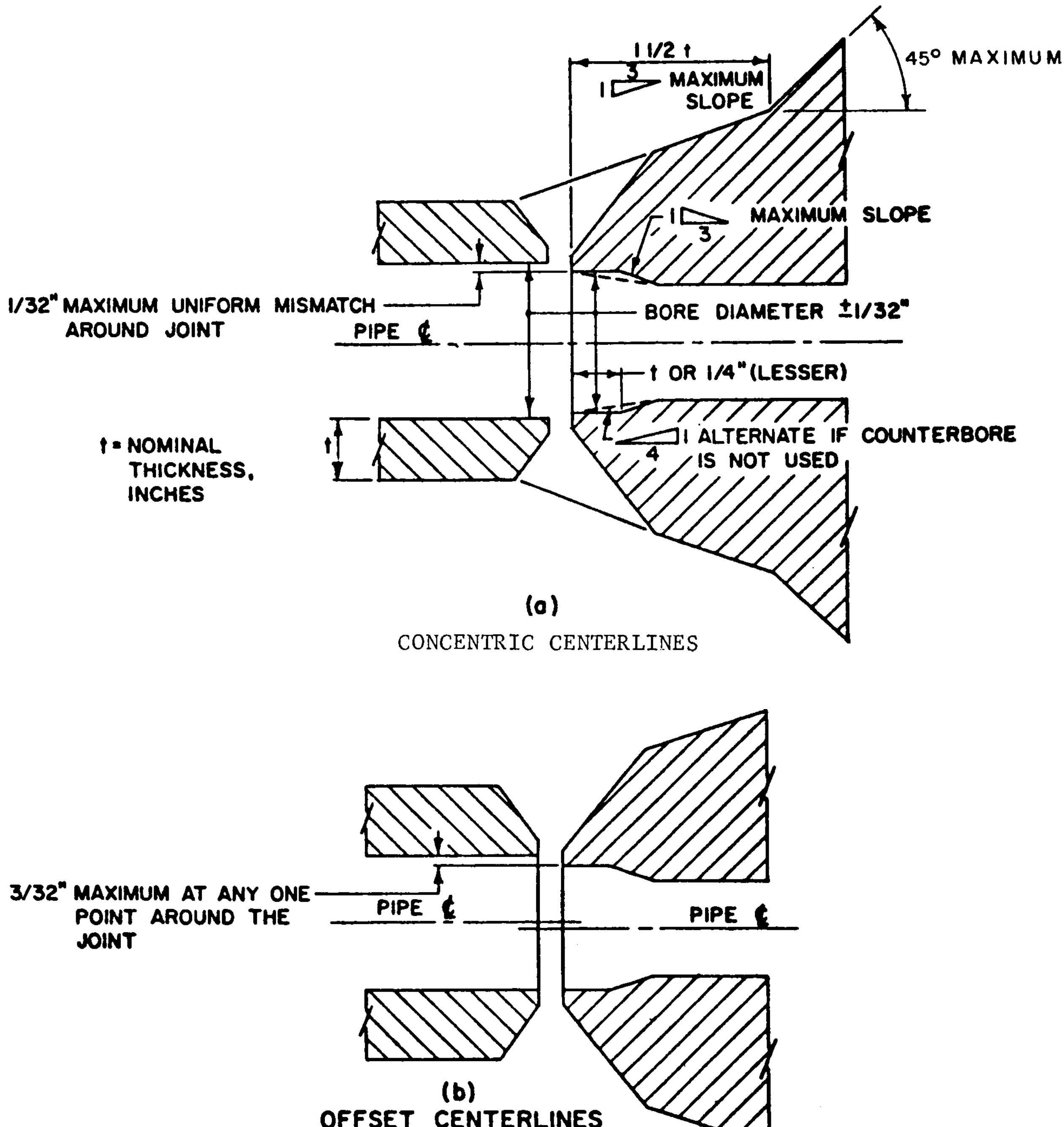
*A seal weld must be used on threaded joints in USAS B31.7.




**NOTE:**

THE COMBINED INTERNAL AND EXTERNAL TRANSITION OF THICKNESS SHALL NOT EXCEED AN INCLUDED ANGLE OF 30° AT ANY POINT WITHIN 1 1/2 T OF THE LAND.

FIGURE 10.1 TRANSITION JOINTS AS ILLUSTRATED BY FIGURE 1-727.3.1 OF USAS B31.7

| Topic No. | |
|---|---|
| 2 | Stress Concentration in Circumferential Fillets |
| 15 | Attachments and Fit-up |

ASME Topic No. 2 has been studied by the PVRC Subcommittee on Stresses in Ligaments; research work is now underway and will be referenced herein. ASME Topic No. 15 has been assigned to the PVRC Fabrication Division; however, no formal study has been started by the PVRC.

## 10.1 Theory

The structures considered herein are axisymmetric in geometry; hence, the relatively well-advanced theory for such structures can be used. In the analysis of stresses in such joints, it is convenient to consider separately

(a) primary and secondary stresses

(b) peak stresses.

The primary and secondary stresses may be calculated with reasonable accuracy* by shell theory. The peak stresses, at least for some geometries and loadings, can be calculated using axisymmetric finite element or finite difference computer programs; e.g., AXISOL[10.2] or DUZ-1[10.3].

### 10.11 Shell Theory (Primary and Secondary Stresses)

For the axisymmetric structures under consideration there are a number of general-purpose shell computer programs which can be used to calculate the primary and secondary stresses in such geometries. However, special-purpose computer programs are also available and, for some limiting cases, very simple equations can be developed. These programs and equations are convenient and economical for preparing graphs and "stress indices".

*Provided that the diameter-to-thickness ratio is greater than about 10.

Rodabaugh and Atterbury[10.4] developed a theory for internal pressure loading the "tapered transition joint" as shown in Figure 10.2. A design graph is given for the axial bending stress at the juncture of the thin-wall pipe to the taper. The study includes a step change in wall thickness ($h = 0$ in Figure 10.2a, b, and c) which is useful in setting an upper bound for the secondary stress at a weld between pipe and a relatively heavy fitting, flange, or valve. In this case, the axial bending stress at the juncture is given by:

$$\sigma_{ap} = \pm 1.54 \frac{C_5'}{P^*} \left(\frac{Pr}{t_1}\right) \tag{10.1}$$

where

$\sigma_{ap}$ = axial bending stress at juncture, pressure loading

$P$ = internal pressure

$r$ = mean cylinder radius

$t_1$ = wall thickness of thin cylinder

$t_2$ = wall thickness of thick cylinder

$$\frac{C_5'}{P^*} = \frac{\rho(\rho^2-1)(\rho-1)}{f(\rho)} + \frac{1 + 2\rho^{3/2} + \rho^2}{f(\rho)} a$$

$\rho = t_2/t_1$

$f(\rho) = 1 + 2\rho^{3/2} + 2\rho^2 + 2\rho^{5/2} + \rho^4$

$a = 0$ for joints per Figure 10.2(a)

$a = +0.972\ (\rho-1)$ for joints per Figure 10.2(b)

$a = -0.972\ (\rho-1)$ for joints per Figure 10.2(c)

$a = +1.944\ (m/t_1)$ for joints per Figure 10.2(d)

$a = -1.944\ (m/t_1)$ for joints per Figure 10.2(e).

(the a-values given are based on Poisson's ratio = 0.3)

In Equation (10.1) the + part of the ± sign refers to the inside surface. For transitions shown in Figure 10.2(d) and (e) the stresses are for the left-hand side of the juncture.

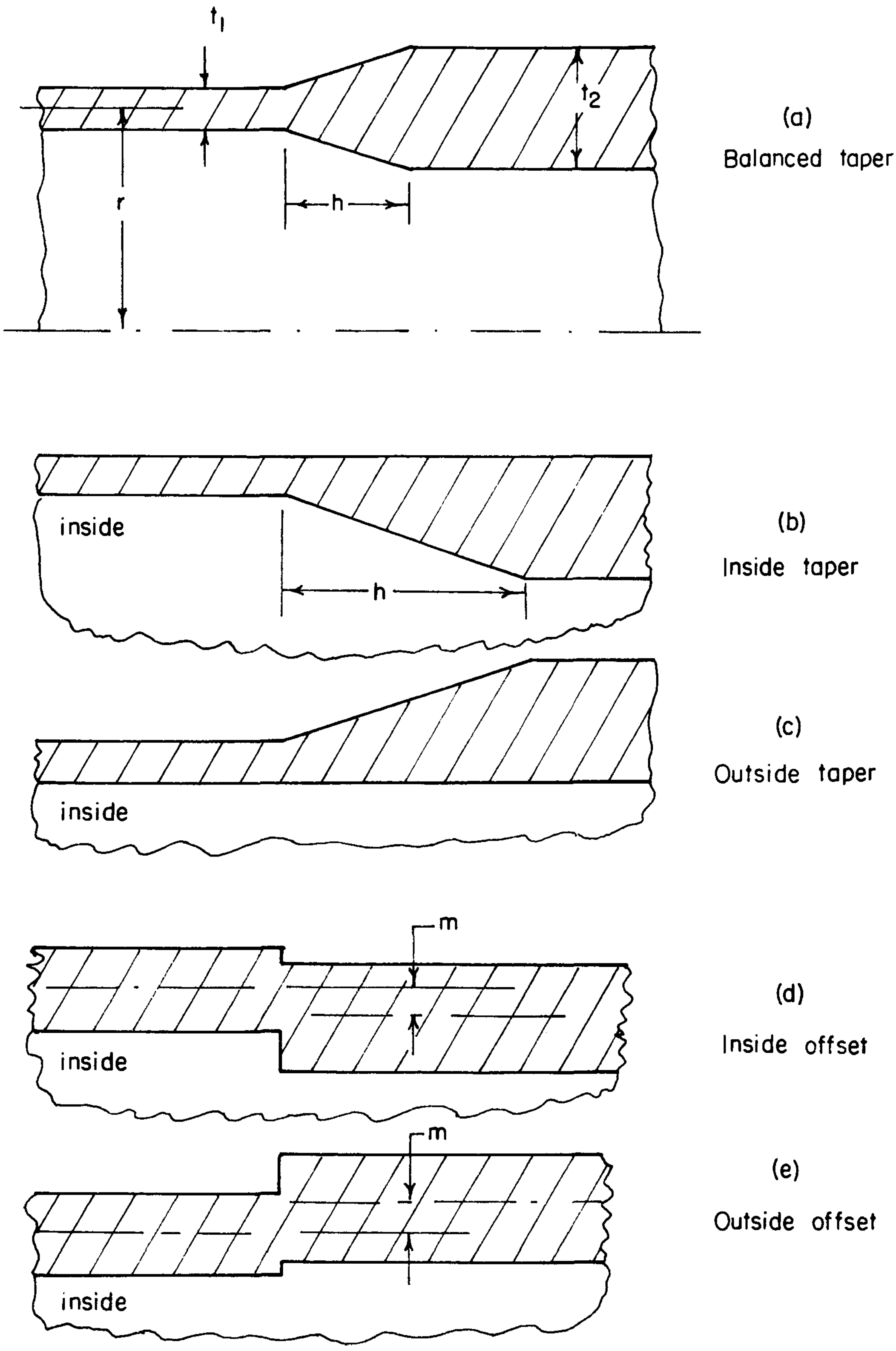



FIGURE 10.2 TRANSITION JOINTS INCLUDED IN THEORY BY RODABAUGH AND ATTERBURY(10.4)

Bizon[10.5] develops equations for the types of geometries shown in Figure 10.3. and gives extensive results in the form of graphs. Bizon's development includes nonlinear effects; however, for pipe with r/t of 100 or less and with internal pressure such that the nominal hoop stress (pr/t) is 30,000 psi or less, the nonlinear effects are negligible (less than 2 percent). For the type of joint shown in Figure 10.3(a), it can be shown that Bizon's results (omitting nonlinear terms) are the same as those obtained by Rodabaugh and Atterbury for their special case of an abrupt change in wall thickness.

The above theories can readily be extended to external moment loading on the pipe by assuming that the maximum force due to the applied external moment, $M_E$, exists all around the pipe circumference; i.e.,

$$N_1 = \frac{M_E t_1}{z_1} = \frac{M_E t_2}{z_2} \tag{10.2}$$

where $N_1$ is the membrane force (lb/in.)

$t_1$, $t_2$ = wall thickness

$z_1$, $z_2$ = section modulus.

For the step transitions (Figures 10.2(d) and (e)), the axial bending stress at the juncture is:

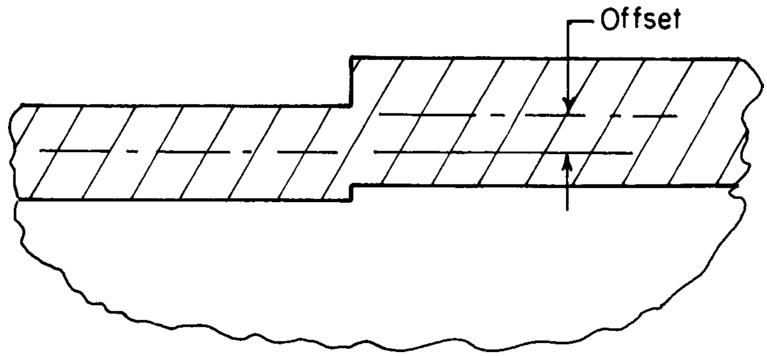
$$\sigma_{am} = \left[\frac{1 + 2\rho^{3/2} + \rho^2}{f(\rho)}\right] \left[\frac{6m}{t_1}\right] \left[\frac{M_E}{z_1}\right] \tag{10.3}$$

where $\sigma_{am}$ = axial bending stresses at juncture, moment loading

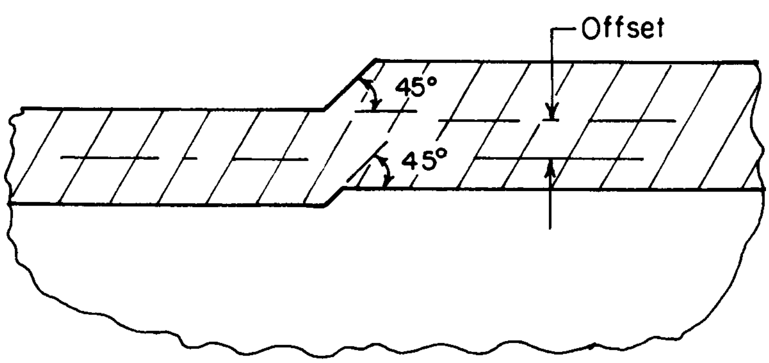
m = centerline offset, positive as defined by Figure 10.2(d), negative as defined by Figure 10.2(e)

Other symbols as defined under Equations (10.1) and (10.2).

The above theories for step transitions can also be extended to a special case of thermal gradients; i.e., where pipe of thickness $t_1$ is at

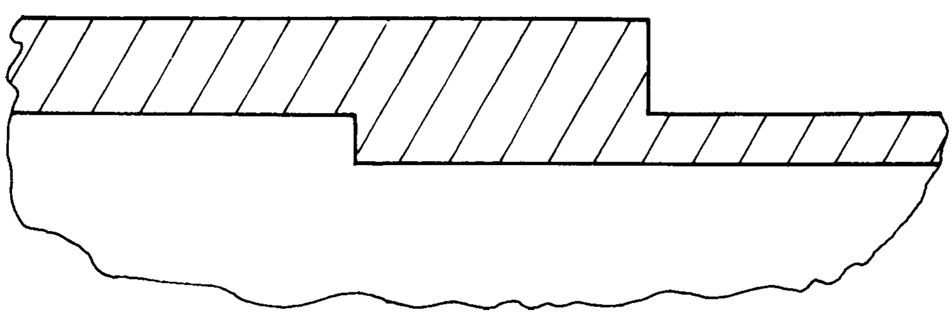



(a) Unfilleted butt joint




Offset can be in-
ward or outward
for either type
of butt weld

(b) Filleted butt joint



(c) Overlap joint

FIGURE 10.3 TRANSITION JOINTS INCLUDED IN THEORY BY BIZON[10.5] temperature $T_1$ and pipe of thickness $t_2$ is at temperature $T_2$. The axial bending stress at the juncture is given by

$$\sigma_{aT} = \left[\frac{\rho^2(\rho^2-1)}{f(\rho)}\right] 1.816\ E\ (\alpha_1\ T_1-\alpha_2\ T_2) \qquad (10.4)$$

where
$\sigma_{aT}$ = axial bending stress at juncture, thermal loading
$\alpha_1, \alpha_2$ = coefficients of thermal expansion, pipe of $t_1$ and $t_2$, respectively
E = modulus of elasticity (assumed to be the same for both pipes)
$\rho$ and $f(\rho)$ as defined under Equation (10.1).

It may be noted that when $\rho = 1.0$; i.e., $t_1 = t_2$, the axial bending stress ($\sigma_{aTm}$) at the juncture is zero. In this case (and for $\rho$ in the range of about 1.33 to 1.0), the maximum axial bending stress occurs away from the juncture; for $\rho = 1.0$ the value is $\sigma_{aTm} = 0.29\ E(\alpha_1 T_1 - \alpha_2 T_2)$.

In all of the above theories, both axial and circumferential, membrane and bending stresses can be obtained as a function of distance from the juncture, using relatively simple equations.

## 10.12 Peak Stresses

The shell-theory stresses cannot, of course, give stresses due to the re-entrant corners of the transition section. For a sharp corner as exists in Figures 10.2(d) and (e), infinitely large linear-elastic stresses would be calculated. If there is some finite radius at the corner, the work of Griffin and Thurman[(10.14)] indicates that a finite-difference computer program (DUZ-1) can be used to predict peak stresses. Work at Battelle-Columbus indicates that similar results can be obtained with a finite-element computer program (AXISOL). These results, particularly for small corner radii, require a very fine grid pattern to achieve accurate results; hence, parametric studies would be very expensive.

Leven[10.6] suggested that stress concentration factors such as given by Peterson[10.7] for a flat plate containing fillets may give some guidance to peak stresses. This approach, along with other comparisons with theory, are given in subsequent Section 10.3.

## 10.2 Test Data

### 10.21 Internal Pressure Loading

Available test data are limited to strain measurements with internal pressure loading. Three of the sets of test data were intended to measure only the primary and secondary stresses. These are tests by Rodabaugh and Atterbury[10.4], Morgan and Bizon[10.8], and Morgan and Bizon[10.9]. Rodabaugh and Atterbury give test data on tapered transitions with taper angle of 14 and 30 degrees, with the taper toward the inside of the pipe. Morgan and Bizon[10.8] give test data on step transitions, with and without fillet radii, and with all three types of transitions; i.e., balanced, outside and inside. Morgan and Bizon[10.9], in a later paper, give test data for various types of mismatched joints.

Three additional sets of tests are available in which measurements were made in sufficient detail so that peak stresses can be estimated. These are tests by Bynum and DeHart[10.10], Leven[10.6], and Heifetz and Berman[10.11]. Bynum and DeHart ran tests on a 17-1/4-inch-ID cylinder, with wall thickness transition from 3/8 to 3/16 inch with a 3/16-inch radius at the transition. The transition is of the "outward" type. Leven ran a photoelastic test on a 8.704-inch-ID cylinder, with wall thickness transition from 0.486 to 0.406 inch with a 0.050-inch fillet radius at the juncture. This was also an "outward" type transition. Heifetz and Berman ran tests on a 21.25-inch-ID vessel with wall thickness transitions from (a) 1.218 to 1.014 inch and

(b) 1.218 to 0.812 inch. Transition radii ranged from 1/16 to 3/8 inch. There were also "outward" type transitions. Some of the results of these tests are given and compared with theory in the following Section 10.3.

## 10.22 Other Loadings

There are no test data available to the writer giving stresses in girth transition joints due to other loadings such as thermal gradients or external bending loads.

Markl and George[10.12] and Meister, et al.[10.13] present results of fatigue tests in which cyclic external bending loads were applied to fillet welded joints.

Markl and George tested 4-inch size carbon steel pipe with various types of fillet welds to 4-inch 300-pound ASA B16.5 flanges. Stress intensification factors, referred to a typical girth butt weld in straight pipe, ranged from 1.09 to 2.36. For external fillet welds, failure occurred in the pipe at the toe of the fillet weld. The internal welds between the end of the pipe and ID of the flange, failures occurred through the weld itself. Because a typical girth butt weld in straight pipe has a stress intensification factor of about 2 with respect to polished bar fatigue strength, the stress indices from these tests ranged from about 2.2 to 4.7.

Meister, et al.[10.13], tested 2 and 4-inch 70-30 copper-nickel pipe either silver-brazed or welded to bronze couplings. Essentially all failures consisted of a crack through the pipe at the juncture with the braze-fillet or at the toe of the fillet weld. The brazing process produced a roughly circular cross-section fillet of braze material with a fillet radius of the order of one-half of the pipe wall thickness. The stress indices based on an endurance strength amplitude of 30,000 psi at $10^5$ cycles for 70-30 copper nickel polished bars, are:

| Nominal Size, in. | Stress Index, $\sigma_f/(M/z)$ Silver Brazed | Fillet Welded |
|---|---|---|
| 2 | 1.7 | 2.1 |
| 4 | 2.2 | 2.6 |

## 10.3 Comparison of Test Data With Theory

### 10.31 Internal Pressure Loading

The test data of References (10.4), (10.8), and (10.9) do not give any information on peak stresses but do show adequate agreement of shell theory and test data. The paper by Morgan and Bizon[10.8] is of particular interest because it gives test data showing the quite significant differences between:

(1) Balanced transition, with half of the step change in thickness inside and half outside the pipe

(2) Inside transition, with all of the step change in thickness inside the vessel, the outside being smooth

(3) Outside transition, with all of the step change in thickness outside the vessel, the inside being smooth.

It is pertinent to compare the peak stresses measured in References (10.6), (10.10), and (10.11) with shell theory stresses and with Peterson's stress concentration factors. These comparisons are shown in Table 10.1. Shell theory axial stresses at the juncture are shown for a step wall-thickness change with the step on the outside surface. Both test data and shell theory stresses are shown as a ratio to the nominal hoop stress, $S = pr/t_1$.

The column headed "Peak Stress Factor" is the ratio of maximum measured axial stress to the shell theory axial stress. This may be compared with the column "Peterson Stress Factor". In general, these columns are

TABLE 10.1 COMPARISON OF TEST DATA WITH SHELL THEORY AND PETERSON'S STRESS CONCENTRATION FACTORS

| Test Data Ref. No. | $t_2/t_1$ | $R_f/(t_2-t_1)$ (2) | $\sigma_a/S$ (1) Test Data | $\sigma_a/S$ (1) Shell Theory Membrane | Shell Theory Bending | Shell Theory Total | Peak (3) Stress Factor | Peterson (4) Stress Factor | $R_f/t_1$ |
|---|---|---|---|---|---|---|---|---|---|
| (10.10) | 2.00 | 1.00 | 0.68 | 0.50 | 0.17 | 0.67 | 1.01 | ~1.1 | 1.00 |
| (10.6) | 1.21 | 0.610 | 0.95 | ↓ | 0.11 | 0.61 | 1.56 | 1.88 | 0.124 |
| (10.11) | 1.50 | 0.154 | 1.15 | ↓ | 0.18 | 0.68 | 1.69 | 2.47 | 0.077 |
| ↓ | ↓ | 0.308 | 1.00 | ↓ | ↓ | ↓ | 1.47 | 1.98 | 0.154 |
| ↓ | ↓ | 0.616 | 0.85 | ↓ | ↓ | ↓ | 1.25 | 1.63 | 0.308 |
| ↓ | ↓ | 0.925 | 0.89 | ↓ | ↓ | ↓ | 1.31 | ~1.45 | 0.462 |
| ↓ | 1.20 | 0.306 | 1.20 | ↓ | 0.11 | 0.61 | 1.97 | 2.37 | 0.062 |
| ↓ | 1.20 | 0.612 | 1.03 | ↓ | 0.11 | 0.61 | 1.69 | 1.87 | 0.123 |

(1) $\sigma_a$ = axial stress, S = nominal hoop stress = $Pr/t_1$

(2) $R_f$ = fillet radius

(3) Peak stress factor = measured stress/total shell theory stress

(4) Peterson stress factor obtained from Reference (10.7), Figure 57 with:

$$D = 2t_2 - t_1,\ d = t_1,\ r = R_f$$

similar but application of the Peterson factor would consistently over estimate the stresses. From this aspect, use of Peterson's factors would be an acceptable design procedure because it appears to be conservative.

One interesting aspect is the series of four tests with increasing fillet radius given in Reference (10.11). The test data indicate a decrease in $\sigma_a/S$ as the fillet radius, $R_f$, is increased, except for the largest value of $R_f$ for which $\sigma_a/S$ increases slightly. A possible clue to this seemingly anomalous behavior is given in Figure 6 of Reference (10.4). This figure shows that for an outside taper, $t_2/t_1 = 1.5$, a small-length taper produces a larger (shell theory) axial stress on the outside surface than does a step wall thickness transition. The larger fillet radii used in the test models might be considered as equivalent to adding a short-length taper to the test model, thereby explaining the slight increase in stress for the largest value of $R_f$.

One note of caution concerning the test data involves the "outside" versus "inside" transition. The shell theory indicates that for some of the models a significantly higher axial stress will occur for the "inside" transition. For example, for $t_2/t_1 = 1.0$, the shell theory axial stresses at the juncture are:

| | $\sigma_a/(pr/t_1)$ | |
|---|---|---|
| Surface | Outside Step Transition | Inside Step Transition |
| Inside | 0.33 | 1.09 |
| Outside | 0.67 | -0.09 |

Figures 10.4 and 10.5 show how the theoretical stresses, calculated by Griffin and Thurman[10.14] using the DUZ-1 computer program, compare with test data by Heifetz and Berman[10.11]. Except for the slight increase in measured stresses at $R_f/(t_2-t_1)$ of 0.462, the agreement between theory and test

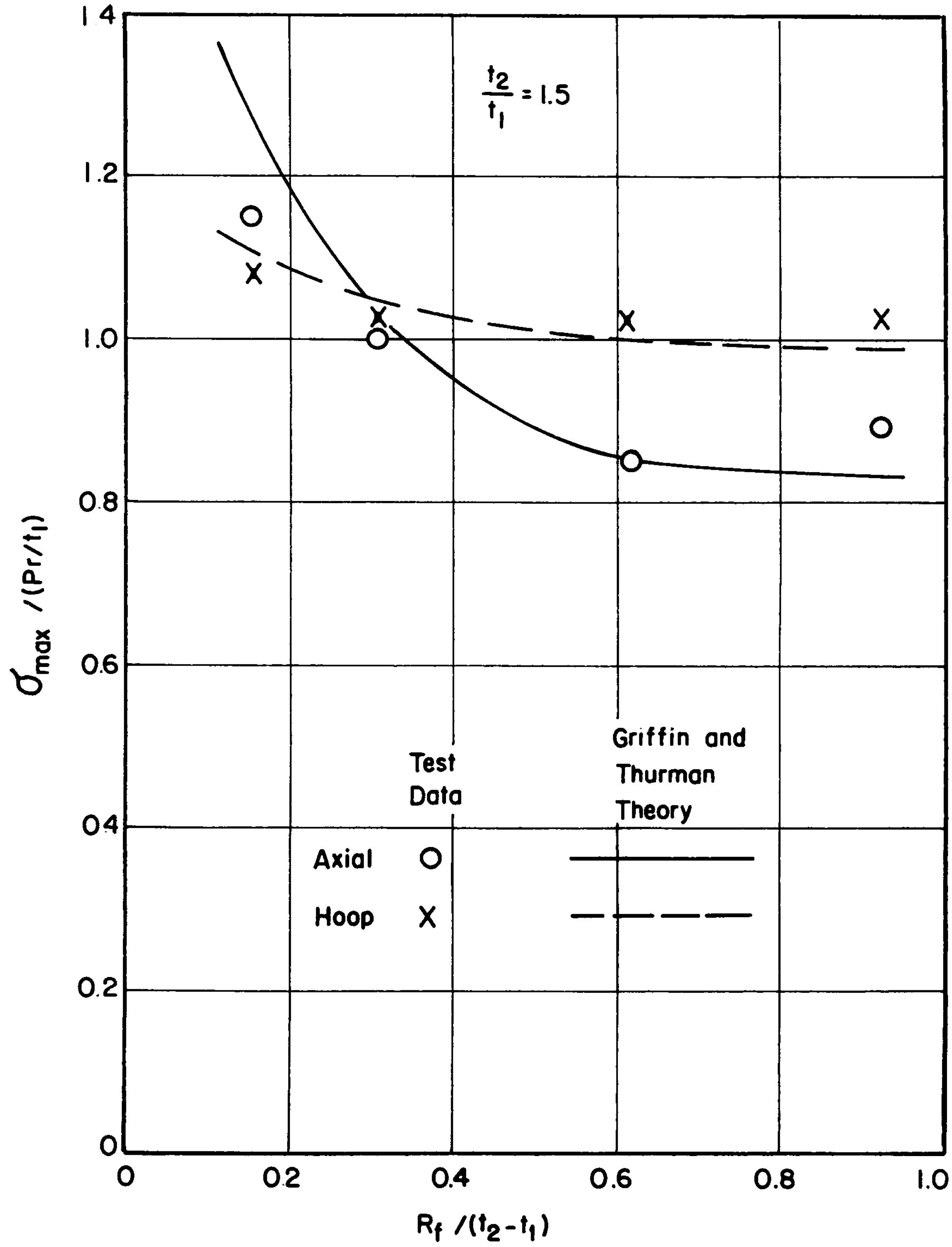



FIGURE 10.4 COMPARISON OF TEST DATA WITH THEORY, $t_2/t_1 = 1.5$

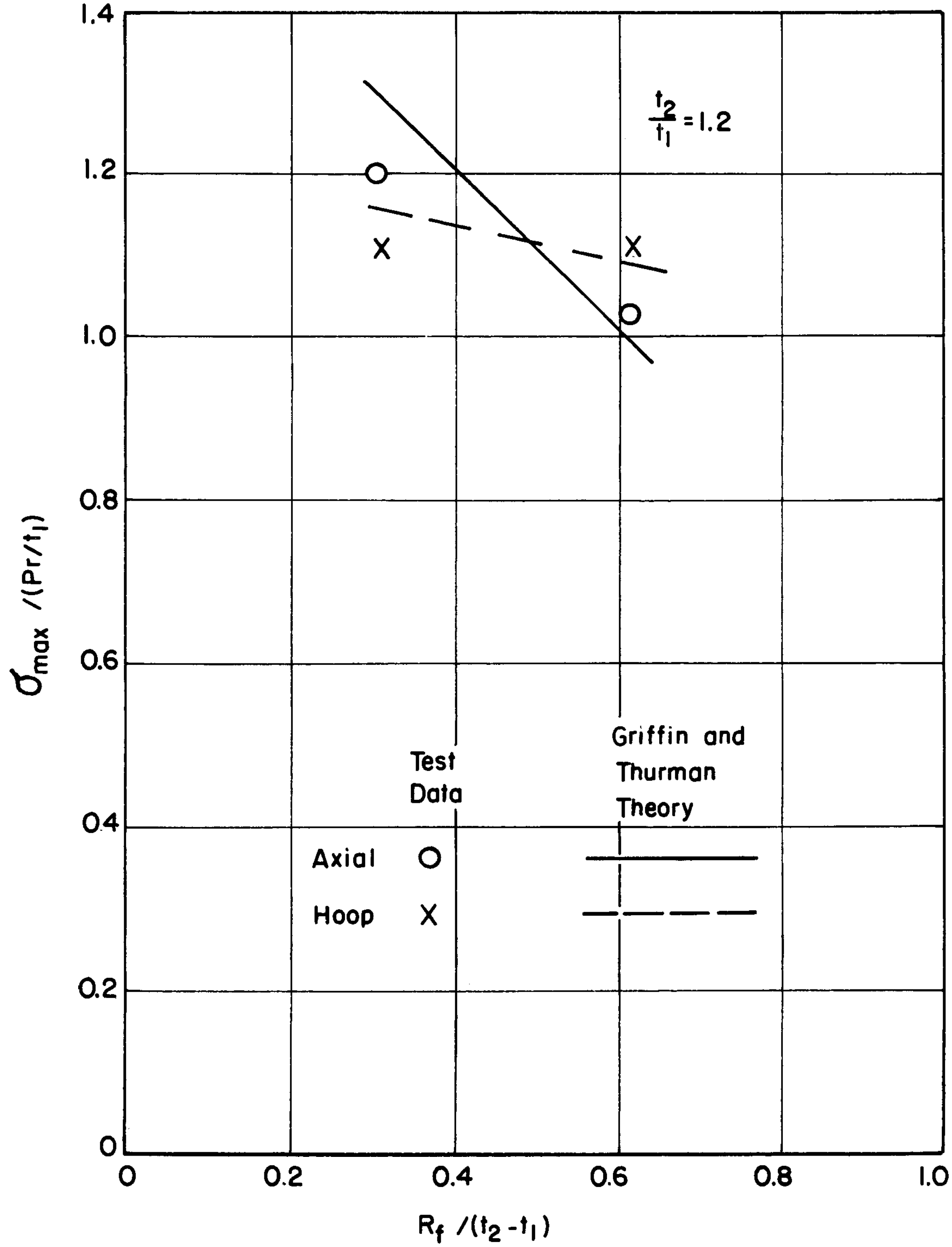
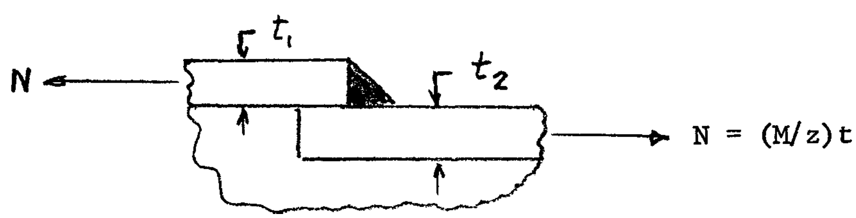



FIGURE 10.5. COMPARISON OF TEST DATA WITH THEORY, $t_2/t_1 = 1.2$

is very good. These graphs serve to illustrate another pertinent point concerning the significance of the test results; i.e., the maximum stresses are only slightly higher than the nominal hoop stress in the thin-wall cylinder.

## 10.32 External Moment Loading

While the bending fatigue tests reported in References (10.12) and (10.13) are not directly comparable with the theories discussed in the preceding, it is of interest to compare the results with the theory for a fillet weld as sketched below:




Using Equation (10.3), with $t_2 = t_1$, m = offset = $t_1$:

$$\sigma_{am} = \left(\frac{4}{8}\right)\left(6\right)\left(\frac{M}{z}\right) = 3(M/z).$$

The membrane stress is simply M/z; accordingly, the calculated stress index is 4.0 as compared to stress indices from 2.1 to 4.7 derivable from data in References (10.12) and (10.13).

## REFERENCES


10.1 USAS B31.7, American Standard Code for Pressure Piping, Section 7, Nuclear Power Piping, dated Feb., 1968, Issued for Trial Use and Comment, Published by the ASME, 345 E. 47th St., New York, N.Y., 10017.

10.2 AXISOL, See "Structural Analysis of Axisymmetric Solids", by E. L. Wilson, AIAA Journal, Vol. 3, pp 2269-2274 (1965).

10.3 DUZ-1, See "DUZ-1, A Program for Solving Axisymmetric and Plane Elasticity Problems on the Philco-2000", WAPD-TM-555 (1965).

10.4 Rodabaugh, E. C. and Atterbury, T. J., "Stresses in Tapered Transition Joints in Pipelines and Pressure Vessels", Trans. ASME, Series B, Vol. 84, pp 321-328 (1962).

10.5 Bizon, P. T., "Elastic Stresses at a Mismatched Circumferential Joint in a Pressurized Cylinder Including Thickness Changes and Meridional Load Coupling", NASA TN D-3609 (May 19, 1966).

10.6 Leven, M. M., "Stress Distribution in a Cylinder with an External Circumferential Fillet Subjected to Internal Pressure", Research Memo: 67-9D7-520-M1, Westinghouse R & D Center, Pittsburgh, Pa. (July, 1965).

10.7 Peterson, R. E., Stress Concentration Design Factors, 1953, John Wiley and Sons, Inc.

10.8 Morgan, W. C. and Bizon, P. T., "Experimental Investigation of Stress Distributions Near Abrupt Change in Wall Thickness in Thin-Walled Pressurized Cylinders, NASA TN D-1200 (Jan. 17, 1962).

10.9 Morgan, W. C. and Bizon, P. T., "Comparison of Experimental and Theoretical Stresses at a Mismatch in a Circumferential Joint in a Cylindrical Pressure Vessel", NASA TN D-3608 (May 19, 1966).

10.10 Bynum, D. J. and DeHart, R. C., "Fillet and Groove Stress Concentrations", Experimental Mechanics, pp 160-166, June, 1964.

10.11 Heifetz, J. H. and Berman, I., "Measurements of Stress-Concentration Factors in the External Fillets of a Cylindrical Pressure Vessel", Experimental Mechanics, pp 518-524, December, 1967.

10.12 Markl, A.R.C., and George, H. H., "Fatigue Tests on Flanged Assemblies", Trans. ASME, Vol. 72, pp 77-87 (1950).

10.13 Meister, Healy, Mindlin, Hyler, and Martin, "Evaluation of Fatigue Properties of Copper-Nickel, Silver-Brazed, and Socket-Welded Joints", Battelle-Columbus Report to Bureau of Ships, USN, Sept. 23, 1965.

REFERENCES (contd)

10.14 Griffin, D. S. and Thurman, A. L., "Calculation of Stresses in Pressurized Cylinders with External Fillets Using DUZ-1 Philco-2000 Computer Program", WAPD TM-654, January, 1967.

CHAPTER 11

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# 11. VALVES AND PUMPS

## 11.1 Valves

### 11.11 Introduction

Most valves are designed on the basis of existing national standards and the manufacturer's knowledge of areas of the valves that must be strain limited. Since, valve functioning is directly dependent upon the valve body's and part's deformations, valve stresses, could be termed, "secondary" in valve design. Because the strains (deformations) must be limited to make a valve function properly, the stresses due to internal pressure are usually relatively small. Also because deformation is usually the limiting consideration, the valve body is in many cases rigid to the extent that a pipe section attached to a valve will yield prior to its being able to impart sufficient forces to cause a pressure boundary failure of the valve, (assuming, of course, that the pipe is not extremely over-designed).

A brief search of technical literature showed, as expected, that very little has been published with regard to the structural adequacy of valves, either in the form of experimental data or analytical design methods. A paper by Jeffrey and Hanlon[11.1] gives some qualitative indication of the bending moments (applied through attached pipe) that will break a 6"-125 lb cast iron valve or will grossly deform a 6"-150 lb cast steel or nodular-iron valve body. A book by Pearson[11.2] presents a method of determining the stresses in noncircular valve-body sections. The method is based on elementary strength-of-materials equations and, at best, would give only crude estimates of stresses. No confirming experimental data are given nor is the existence of such data indicated.* Three-dimensional photoelastic analysis has been utilized to design valves[11.3]. This type of analysis is of considerable value in reducing peak stresses.

Another design approach for valve bodies may be described as the pressure-area method. This method has been used for the design of pipe fittings having complex shapes. Figure 11.1, taken from Page 71 of Kellogg's[11.4] book on "Design of Piping Systems", illustrates the concepts involved in the pressure-area method. Lind[11.5] has developed this method quantitatively for the special case of tees consisting of the intersection of two uniform-wall cylinders.

From a more rigorous analytical standpoint, some of the finite-element computer programs being developed (see Chapter 3) based on networks of beam, plate or shell elements may prove capable of determining stresses and displacements in valve bodies.

It might be remarked that numerous articles in the trade literature summarize valve types, factors involved in valve selection, and qualitative suggestions concerning installation and maintenance. One typical article of this type is contained in Reference (11.6). Similar information can be found in valve manufacturers' catalogs.

Based on contacts by the writer with technical representatives of several valve manufacturing companies**, it appears that valve bodies are designed principally on the basis of past experience and extrapolations

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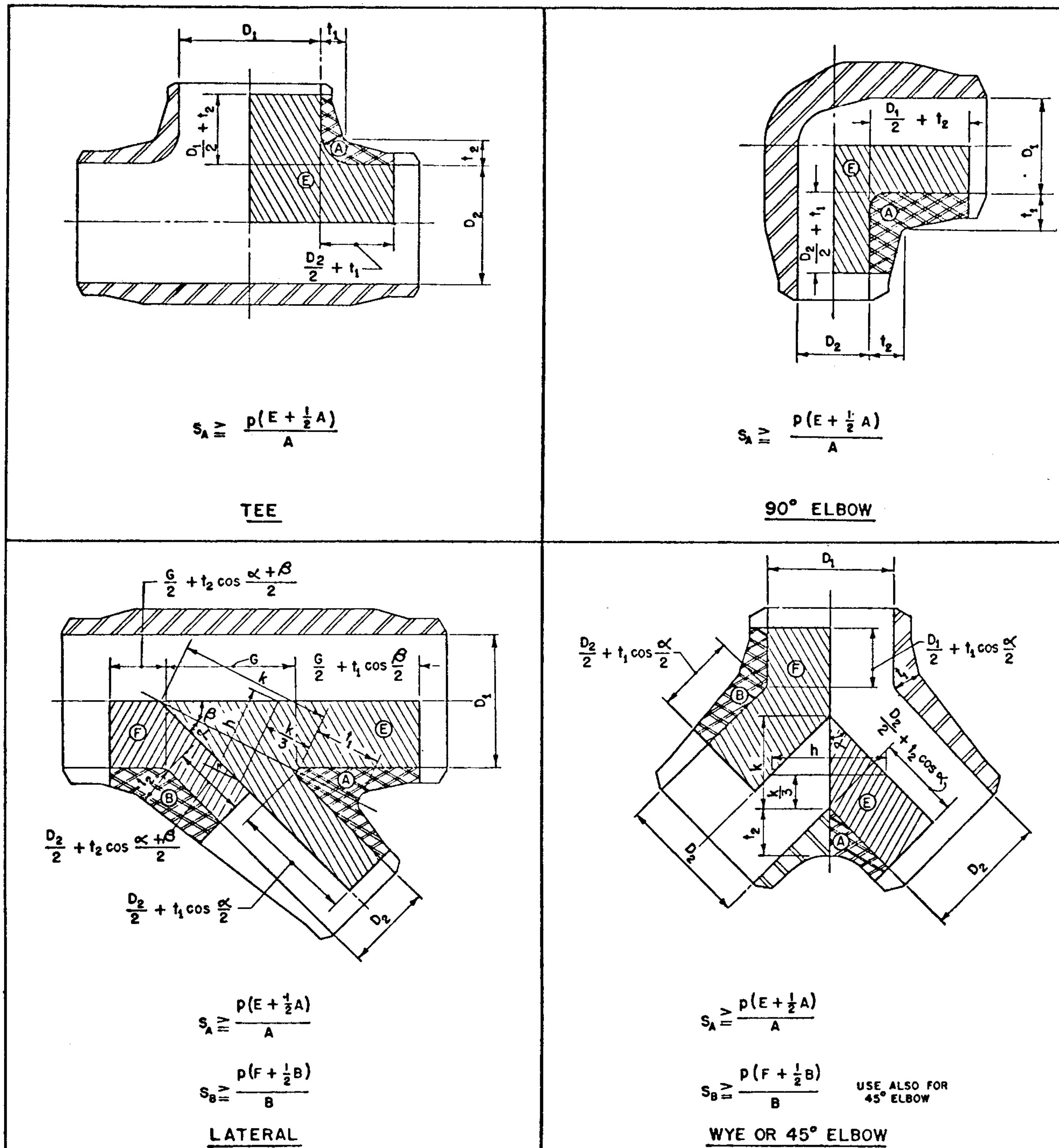
* Various valve manufacturers have determined stresses in valve bodies by means of strain gages; however, no such data have been found in the open literature.

** Crane Company
Wm. Powell Company
Rockwell Mfg. Company
Walworth Company




**NOMENCLATURE**

- A, B - METAL AREA, (SQ.IN.)
- $D_1$, $D_2$, - INSIDE DIAMETER OF FITTINGS, (IN.)
- E, F - INDICATED PRESSURE AREA, (SQ. IN)
- G, h, k - INDICATED LENGTHS, (IN.)
- p - DESIGN PRESSURE, AT DESIGN TEMPERATURE, (PSIG)
- $S_A$, $S_B$ - ALLOWABLE STRESS AT DESIGN TEMPERATURE, (PSI)
- $t_1$, $t_2$ - INDICATED METAL THICKNESS, (IN.)
- $t_3$ - AVERAGE METAL THICKNESS OF FLAT SURFACE, (IN.)
- $\alpha$, $\beta$ - INDICATED ANGLES.

FIGURE 11.1. ILLUSTRATION OF PRESSURE-AREA DESIGN METHOD, FROM FIG. 3.14 OF M. W. KELLOGG, REFERENCE (3)

thereof. There are, however, certain minimum requirements given in USAS, MSS, and API standards applicable to valves. These requirements will be discussed in the following five sections.

## 11.12 USAS Standard B16.5

While B16.5 is entitled, "Steel Pipe Flanges and Flanged Fittings", it is nevertheless the primary standard for steel valves sold in tne United States. B16.5 controls steel valve design by prescribing:

(1) Minimum wall thickness,

(2) Dimensions of flanges and bolting, for flanged-end valves,

(3) Dimensions of welding ends, for welding-end valves,

(4) Face-to-face or end-to-end dimensions (by reference to USAS B16.10, "Face-to-Face and End-to-End Dimensions of Ferrous Valves"),

(5) Pressure-temperature ratings,

(6) Minimum hydrostatic test pressure (for body; not a seat test).

Piping components included in B16.5 are divided into "series" or "classes", designated as a number followed by the symbol "lb". The seven classes in B16.5 are listed below, along with their $100^{\circ}$ F (secondary) rating pressure, primary rating temperature for carbon steel, and hydrostatic test pressure. The primary rating pressure is the pressure in psi that is the same as the class designation; e.g., the 150 lb class has a primary rating pressure of 150 psi.

| Class (Primary Rating) | Secondary Rating at 100°F psi | Test Pressure psi | Primary Rating Temperature for Carbon Steel, degrees F |
|---|---|---|---|
| 150 lb | 275 | 425 | 500 |
| 300 lb | 720 | 1100 | 850 |
| 400 lb | 960 | 1450 | ↓ |
| 600 lb | 1440 | 2175 | ↓ |
| 900 lb | 2160 | 3250 | ↓ |
| 1500 lb | 3600 | 5400 | ↓ |
| 2500 lb | 6000 | 9000 | ↓ |

The secondary rating (rating at 100 F) and test pressure, except for the 150 lb class, is 2.4* times and 3.6* times the primary rating pressure, respectively.

B16.5 gives pressure-temperature ratings for components made of 17 different steel materials. The temperature range is from -20 to 1200 F for ferritic steels; to 1500 F for austenitic steels. The background of these ratings is described by Rodabaugh[11.7]. As shown in Reference (11.7), there is no analytic tie-in of valve body designs with the ratings. The ratings are related to the capacity of the flanged joints which would be used with flanged-end valves.

The minimum wall thicknesses prescribed by B16.5 are all greater (by from 0.1 to 0.2 inches**) than those determined by the equation:

$$t = 1.5 \left[\frac{Pd}{2S - 1.2P}\right] \tag{11.1}$$

* Except for 304, 304L, and 316L materials.

** This extra thickness is usually considered as a corrosion/erosion allowance.

where $t$ = calculated thickness, inches

$P$ = primary rating pressure, psi

$d$ = inside diameter of valve (as taken from tables in B16.5)

$S$ = stress of 7,000 psi.

It can be seen that the factor in brackets in Equation (11.1) is an equation* for calculating the wall thickness of a straight pipe. The factor of 1.5 is usually considered as a shape factor, i.e., a valve body is not a pipe. This interpretation implies that an allowable stress of 7,000 psi can be used in establishing minimum wall thickness of valve bodies. For comparison, allowable stresses given in Section I of the ASME Boiler Code are shown below:

| Material | B16.5 Primary Rating Temperature F | ASME Section I Allowable Stress, psi |
|---|---|---|
| A216-WCB (Carbon Steel) | 850 | 7800 |
| A217-WC4 (1/2-Cr - 1/2-Mo) | 950 | 10000 |
| A217-WC9 (2-1/4 Cr - 1 Mo) | 975 | 9400 |
| A351-CF8 (Type 304) | 1000 | 9850 |

A casting quality factor of 0.80 might be applied to the ASME allowable stresses.

B16.5, under <u>Wall Thickness</u>, also states:

"Additional metal thickness needed for assembly stresses, valve

---

* For example, Equation (3) of USAS B31.1 with $E = 1.0$, $y = 0.4$, $A = 0$, $D_o = d + 2t$.

closing stresses, shapes other than circular, and stress concentrations must be determined by individual manufacturers since these factors vary widely."

## 11.13 MSS Standard SP-66

SP-66, "Pressure-Temperature Ratings for Steel Butt-Welding End Valves" is a relatively new standard (originally approved in January, 1964). Since valves in primary coolant loops are usually made of steel and have butt-welding ends, MSS-SP66 is pertinent, at least in indicating some opinions as to ratings of such valves. SP-66 contains two parts, Part I simply accepts the ratings given in B16.5. Part II provides an alternate method of rating valves which meet specific requirements.

The minimum wall thickness of pressure retaining components is given by the equation:

$$t = 1.5 \left[\frac{Pd}{2S - 2P(1 - y)}\right] + 0.1 \qquad (11.2)$$

where t = minimum wall thickness of pressure containing parts, inches

P = maximum allowable pressure at operating temperature, psi

d = minimum diameter of the flow passage, but not less than 90% of the I.D. at the welding ends, inches

S = allowable stress at operating temperature, psi (Table P7 - Section I ASME Boiler & Press. Vessel Code).

For y = 0.4 (applicable to ferritic materials to 900 F, austenitic materials to 1050 F), Equation (11.2) is identical in form to Equation (11.1), except that an additional thickness of 0.1" is explicitly required whereas the tabulated thicknesses of B16.5 are from 0.1 to 0.2" greater than required by Equation (11.1). The value of "d" in Equation (11.2) is not greatly different than in Equation (11.1). Accordingly, the significant difference lies in the value of S.

The rating pressure given in SP-66 is obtained by solving Equation (11.2) for P; i.e.,

$$P = \frac{S(t - 0.1)}{.75d + (1 - y)(t - 0.1)} \tag{11.3}$$

Application of Equation (11.3) to B16.5 valves would result in a substantial increase in pressure-temperature ratings, particularly near 650 F and for 150 lb valves. Some comparisons are shown in Table 11.1. These comparisons are based on the approximations that the minimum wall thickness is given by Equation (11.2) using S = 7000 psi and P = primary rating pressure, psi, and that the inside diameters given in ASA B16.5 are the same as defined by "d" under Equation (11.2). Additional comparisons of this kind are shown by Millville(11.8).

It may be noted that the 150 lb class ratings are not dependent upon the material and that those ratings are "different" than those assigned to all higher classes, the latter being proportional to the class number (e.g., rating for a 600 lb class is twice that of the 300 lb class for a given material and temperature.) There are a number of reasons for the "different" ratings of the 150 lb class; these are discussed in Reference (11.7). With respect to valves, one aspect is related to the shape of the valve body and bonnet flange. Because of the relatively short end-to-end dimensions prescribed for 150 lb valves, in larger sizes it is usually necessary to use an oval cross section for the upper part of the body, and use oval bonnet

TABLE 11.1 COMPARISON OF ASA B16.5 AND MSS-SP66 RATINGS FOR WELDING END VALVES WITH MINIMUM WALL THICKNESS AS REQUIRED BY ASA B16.5

| Material | Temp. F | 150 lb Class(1) B16.5 | 150 lb Class(1) SP-66 | 300 lb and Higher Class(2) B16.5 | 300 lb and Higher Class(2) SP-66 | S(3) ASME-I |
|---|---|---|---|---|---|---|
| A212-WCB (Carbon Steel) | 100 | 275 | 375 | 2.40 | 2.50 | 17,500 |
| | 650 | 120 | 375 | 1.72 | 2.50 | 17,500 |
| | 850 | 82 | 167 | 1.00 | 1.12 | 7,800 |
| A217-WC4 (1/2-Cr-1/2-Mo) | 100 | 275 | 375 | 2.40 | 2.50 | 17,500 |
| | 650 | 120 | 375 | 1.72 | 2.50 | 17,500 |
| | 950 | 82 | 214 | 1.00 | 1.43 | 10,000 |
| A217-WC9 (2-1/4 Cr-1 Mo) | 100 | 275 | 375 | 2.40 | 2.50 | 17,500 |
| | 650 | 120 | 375 | 1.72 | 2.50 | 17,500 |
| | 975 | 82 | 202 | 1.00 | 1.34 | 9,400 |
| A217-C5 (5 Cr-1/2-Mo) | 100 | 275 | 482 | 2.40 | 3.22 | 22,500 |
| | 650 | 120 | 482 | 1.72 | 3.22 | 22,500 |
| | 975 | 82 | 185 | 1.00 | 1.24 | 8,650 |
| A351-CF8 | 100 | 275 | 375 | 2.06 | 2.50 | 17,500 |
| | 650 | 120 | 246 | 1.23 | 1.64 | 11,500 |
| | 1000 | 82 | 211 | 1.00 | 1.41 | 9,850 |

(1) Ratings in psi.

(2) Ratings given as a multiple of P', P' = primary rating pressure P' is equal (in psi) to the class designation.

(3) S = allowable stress (psi) given in ASME Boiler Code, Section I, (1965) Table PG23.1.

flanges. This shape of structure is difficult to design for internal pressures significantly higher than the present 150 lb class ratings.

The requirement in MSS-SP66 that "...., further, all other components must be of suitable design and strength for the service conditions," should be considered in relationship to Table 11.1. The increase in ratings given by MSS-SP66 must be accompanied by a review and perhaps strengthening of other parts of the valve; e.g., stem, disc, seats, yoke, etc. This is a responsibility of the valve manufacturer.

## 11.14 API Std. 600

API-600, "Flanged and Butt-Welding-End Steel Gate and Plug Valves" is significant because it gives a number of qualitative and a few quantitative requirements for detailed design of valves of the types covered therein. It gives material requirements for body and bonnet, bonnet bolting, gland bolting, stem, gate, seat rings, handwheel and bonnet gasket. Some qualitative design guidance is given for the body and bonnet shape, stem packing, glands and yokes. It gives quantitative design requirements for minimum wall thicknesses of the body and bonnet, and for minimum stem diameters.

The minimum wall thickness of body and bonnet are all considerably greater than those given in ASA B16.5. These additional thicknesses represent corrosion/erosion allowances and are not used for rating purposes. The pressure temperature ratings are the same as those given in ASA B16.5.

The required shell test pressure is the same as given in USAS B16.5. The required seat test pressure is equal to the secondary (100 F) rating pressure. In addition, an air seat test (80 psig) is required.

## 11.15 Other Standards

In addition to the three standards discussed in the preceding, that apply to steel valves commonly specified for critical service conditions, there are a number of other standards applicable, to some extent, to valve design and/or ratings. These are listed below:

ASA Standards

| | |
|---|---|
| B 16b1 1931 (Reaffirmed 1952) | "Cast-Iron Pipe Flanges and Flanged Fittings"; For maximum non-shock working hydraulic pressure of 800 psi at ordinary air temperatures. |
| B16b2-1931 (Reaffirmed 1952) | "Cast-Iron Pipe Flanges and Flanged Fittings"; for maximum working pressure of 25 psi saturated steam pressure. |
| ASA B16.1 - 1960 | "Cast-Iron Pipe Flanges and Flanged Fittings, Class 125"; for 125 psi saturated steam and other pressure-temperature ratings. |
| ASA B16.2 - 1960 | "Cast-Iron Pipe Flanges and Flanged Fittings, Class 250"; for 250 psi saturated steam and other pressure-temperature ratings. |
| ASA B16.24 - 1962 | "Bronze Flanges and Flanged Fittings"; 150 and 300 lb, pressure ratings up to 500 psi at 150 F. |

(Valve manufacturers furnish valves to ratings and with minimum wall thicknesses as given in the above standards.)

MSS Standards

| | |
|---|---|
| SP-37-1959 | "125 lb Bronze Gate Valves" Ratings up to 125 psi at 353 F. Threaded, soldered or flanged ends. |
| SP-42-1959 | "150 lb Corrosion Resistant Cast Flanged Valves" Ratings up to 150 psi at 500 F. |
| SP-52-1957 | "Cast Iron Pipe Line Valves", B16.1, B16.2, B16.61 ratings, flanged ends. |

(These standards give some detailed requirements for stem diameters, and material requirements for some components; e.g., seating material, handwheels, bonnet or cover bolting, etc.)

There are some other MSS standards related to valves, e.g., SP-6 on flange face finishes, SP-25 on marking, SP-53, -54, and -55 on inspection standards for steel castings, SP-61 on shell and seat tests, etc.

API Standards

| | |
|---|---|
| 602-1964 | "Compact Design of Carbon Steel Gate Valves for Refinery Use" (2" and smaller) |
| 603-1962 | "150-lb, Light-Wall, Corrosion-Resistant Gate Valve for Refinery Use (1/2" to 12", inclusive). |
| 604-1966 | "Flanged Nodular Iron Gate and Plug Valves for Refinery Use" (150 lb and 300 lb, B16.5 ratings) |
| 6A-1966 | "Specification for Wellhead Equipment", (Includes special petroleum production valves.) |
| 6D-1964 | "Specification for Steel Gate, Plug, Ball and Check Valves for Pipeline Service" (Essentially, a supplement to the requirements given in ASA B16.5). |

## 11.16 Performance or Design Proof Tests

A review of the standards listed in the preceding indicates that the user obtains his primary assurance of structural adequacy from the specified shell and seat pressure tests. These pressure tests are generally prescribed to be 1.5 times the 100 F rated pressure for the shell test (valve seat open); 1.0 times the 100 F rated pressure for the seat test. The seat test is usually applied after the shell test, hence it can be assumed that no gross distortion of the body occurred during the shell test because such gross distortion would presumably lead to leakage in the subsequent seat test. In some valve manufacturing processes, the shell test is accomplished by using a hydraulic press to provide end closures; the test is not necessarily the equivalent of a valve in a pipeline, which would be subjected to axial loads as well as the distributed radial load due to internal pressure.

The seat test, while primarily aimed at establishing a satisfactorily low leakage rate*, also gives a test of the structural adequacy of the disc, stem, seats, bonnet, and bonnet flanges and other details of the valve construction.

In addition to internal pressure, a valve in a pipeline is subjected to loads applied to the valve by the attached pipe. For piping that undergoes temperature change in operation, loads of this type can hardly be avoided.

The standards cited above say nothing about such pipe-imposed loads. From the standpoint of safety of a valve in a primary-coolant loop in a nuclear-piping system, two questions arise:

* MSS SP-61 specifies maximum rates of 10 cc. per hour per inch of valve diameter for water seat tests; 0.1 std. cu ft per hour per inch of valve diameter for an air (80 psig) seat test.

(1) With a combination of rated internal pressure and some reasonable amount of external forces, will the pressure boundary of the valve body remain intact?

(2) With combined loads as in (1) above, can the valve be closed and/or opened?

An answer to these questions is not available although, as remarked in the first paragraph of the introduction, it seems likely that steel valves made in accordance with USAS B16.5 can sustain a bending load equivalent to the yield moment of attached pipe (the pipe wall thicknesses being designed for the same pressure as the valve) without pressure boundary failure of the valve body. The only available test data are given by Reference (11.1). A 6" cast steel (presumably 150 lb class) flanged gate valve was heated to 900 F, then subjected to increased bending moments up to 55,500 ft-lb. At this bending load, the valve body had grossly distorted but no fracture of the metal occurred. The nominal stress in 6" std. wt. pipe at a bending moment of 55,500 ft-lbs. is 77,800 psi. There is no information given as to what load would cause the valve to become in-operable; nor as to any time-effects (creep at 900 F).

On the other hand, there seems to be some reason to question whether all valves do have adequate strength and rigidity to resist even moderate external loads. References (11.6) contains the statement that "Most valves are not designed for external stressing". A quote from Reference (11.9) is: "Keep pipe strains off valves--don't let the valves carry the weight of the line. The distortion from this cause results inefficient operation, jamming, and early maintenance". A quote from Reference (11.10) is: "Use suitable hangers close to both sides of the valve in order to reduce stresses transmitted by the pipe. Most valves are not designed to cope with external stresses". Milleville(11.8) states: "It can be argued that a valve ought to have sufficient strength so that the adjacent end connection or pipe will fail before the valve is damaged by externally applied bending loads". He goes on to suggest that qualification tests might be used to insure adequacy in this respect.

In order to obtain assurance of the structural adequacy of valves for critical service applications, perhaps a feasible approach at this time would consist of the inclusion (in valve standards, or supplementary standards) of "design proof tests".* These design proof tests might consist of:

(1) A hydrostatic shell test to two times the rated pressure at temperature T, adjusted by the ratio $S_t/S_o$, where $S_t$ = allowable stress at test temperature, $S_o$ = allowable stress at the rated temperature, T.

(2) A combination loading of internal pressure equal to the rated pressure at temperature T, adjusted by the ratio of $S_t/S_o$, plus moments $M_i$ and $M_o$ (separately, but both with internal pressure), where

$M_i = M_o = f\ S_t\ Z$

$Z$ = section modulus of equivalent pipe = $0.0982(D_o^4 - D_i^4)/D_o$

$D_o$ = outside diameter of pipe to be attached to valve.

$D_i = D_o - 2t_p$

$t_p$ = required thickness of pipe attached to valve

---

* Inclusion of tests of this type are at present (July, 1969) being considered by the Task Force on Steel Ratings of USAS Subcommittee No. 4. While still in the early stages of Committee work, the present intent is to include such design proof tests in USAS B16.5.

$f$ = a factor indicating what part of the bending moment that can be transmitted by the pipe to the valve, as limited by $S_t$, must be supported by the valve. ($f = 1.5$ is tentatively suggested)

$M_i$ = applied moment in plane of valve stem and pipe axis.

$M_o$ = applied moment, 90° to plane of valve stem and pipe axis.

While the valve is subjected to $M_i$ or $M_o$, and with rated internal pressure:

(a) on one side of the seat only,

(b) with pressure on both sides of the seat,

the valve must be opened and closed without undue effort, or with the valve operator furnished for the valve.

(3) After completion of (1) and (2) above, seat tests would be required as given in API 600 or similar valve standards. (This step would be to obtain assurance that the loadings of Steps (1) and (2) did not produce gross structural distortion of the valve body, disc, seat, etc.).

The design proof test procedure described above would leave several questions unanswered, e.g., what is the effect of cyclic internal pressure and/or cyclic moments ($M_i$ or $M_o$)? What is the effect of thermal gradients? What is the effect of wear due to valve operation? What is the effect of creep at high temperatures? Answers to these questions by means of design proof tests, while technically feasible, would involve very high costs.

## 11.2 Pumps*

While pumps are not usually considered as a piping component or part of a piping system, the PVRC Subcommittee on Stress Indices does include pumps among the components of interest. Accordingly, a brief summary is given here regarding efforts to obtain assurance of the adequacy of pump casings as a pressure retaining boundary. In present day water-cooled nuclear reactors, the main circulating pumps are centrifugal; either bowl type or volute type. The following remarks are restricted to such pumps.

The design of pump bodies has been basically oriented around their fluid dynamic and strain properties. Since the basic function of a pump is to move fluids, most analytical and experimental work has been performed in the hydraulics field.

To make a pump perform its hydraulic duties, a designer must be deformation oriented, not necessarily stress oriented. To make impellers operate properly, the designer cannot allow pump casings or covers to deform to allow bearing and alignment malfunctions. Thus, bodies, splitters, and bearing retainers are design for minimum deformation. Consequently, average stresses in the pressure retaining boundary components are usually very low.

If stresses, per se, in a pump casing are examined, all of the ramifications and unknowns of other pressure vessel research studies are uncovered. For instance, to name some:

---

* This portion of the Survey Report was contributed by Mr. Byron J. Round, Combustion Engineering, Inc., Windsor, Connecticut.

(1) Irregular, nonaxisymmetric shapes.

(2) Nonradial nozzles (actually tangential).

(3) Stress concentrations due to:

(a) Bearing seats

(b) Splitters

(c) Material thickness transitions

(d) Openings.

(4) Huge bolted flanges.

(5) Thermal gradients.

(6) External forces, which may vary considerably depending on the method of support of the pump and/or piping system.

(7) Static and dynamic loads, due to the drive mechanism.

Certain parts, such as the bolted flange connection for the drive mounting can be calculated by existing code methods[11.14].

Girth welded joints and/or flanged connections to the inlets and outlets of the pump are under the jurisdiction of other codes[11.12,11.13,11.15]. Beyond these locations, there are no existing standards, at this time, for guidance to determine the pressure integrity of the pump casing except the proprietary knowledge of the manufacturers. The writer is not aware of any ruptures that have occurred in pump casings.

Efforts have been made to correlate theoretical solutions for some areas of pump casings with experimental strain gage readings on corresponding areas of the casings. These include:

(1) Aerojet-General, using a finite element program[11.18].

(2) Byron Jackson Pump, using a finite element analysis[11.19].

(3) Combustion Engineering, using a thick-shell program[11.3].

(4) Franklin Institute/Bingham Pump, using a finite element program[11.17].

(5) General Electric, using a shell program[11.16].

(6) Westinghouse Electric, using an axisymmetric analysis[11.20].

As indicated by the above, some test data do exist and efforts are underway to correlate those data with theoretical analyses. So far, the correlations have not been entirely satisfactory. Before a reliable correlation between theoretical solutions and experimental data can be determined for pumps, a considerable amount of time and money must be expended. Perhaps, when solutions are obtained for comparatively simple shapes such as tees, laterals and wyes, the pump criteria will evolve automatically.

## 11.3 ASME Code for Pumps and Valves for Nuclear Power

The ASME has issued a Draft Code for Pumps and Valves for Nuclear Power[(11.11)]. This code is issued for trial use and comment; however, it is invoked* by ASME Section III, Nuclear Vessels[(11.12)] and hence can be considered as an active document. This code is intended to be used in conjunction with ASME Section III and USAS B31.7, Nuclear Power Piping Code[(11.13)], so as to provide a standard of equivalent quality for the pressure equipment of a nuclear energy system of any water-cooled commercial nuclear power plant.

The ASME Pumps and Valves Code covers materials, manufacture, examination, pressure tests, marking, stamping, and reports in a manner quite similar to ASME Section III. The section of design, which is of primary interest in this report, is somewhat different than ASME Section III in that it presents relatively specific procedures, formulas, and charts for evaluating the adequacy of Class I valves. The reason for this is that the complex shapes of valves, and their attendant sharp discontinuities at valve seats and guides, make it essentially impractical to perform a rigorous ASME Section III type of analysis with presently developed theoretical techniques. There are, of course, certain parts of some valves which are essentially axisymmetric; e.g., body-to-bonnet joints, pressure seal, seat areas of an angle valve. Axisymmetric computer programs can be used for such parts with, in some cases, probably a reasonable degree of accuracy.

The ASME Pumps and Valves Code, in Article 4 on Design, gives rules applicable to valves larger than 4" nominal pipe size. The design method is based on the minimum valve body thickness criteria of USAS B16.5. Certain additional requirements are imposed to provide increased assurance of valve adequacy for Class I nuclear service.

* See Summer 1969 Addenda to ASME Section III.

Linear interpolation between USAS B16.5 pressure classes is permitted. Qualitative "Body Shape Rules" are given concerning corner and fillet radii, out-of-roundness and curved sections. The primary membrane stress (due to internal pressure is determined by a pressure-area method similar to that illustrated in Figure 11.1 The sum of the primary and secondary stresses are limited by the equation:

$$Q_p + P_e + 2Q_T \leqq 3S_m$$

where

$Q_p$ = primary and secondary stress, pressure loading

$P_e$ = secondary stress, external moment loading

$2Q_T$ = secondary stress, thermal gradients

$S_m$ = allowable stress for the valve body material at 500 F.

Equations are given for evaluation of $Q_p$, $P_e$, and $Q_T$. The first two terms involve factors derived from tests on tees and branch connections. The third term is based on a fluid temperature ramp increase and decrease of 500 F per hour.

The fatigue requirement for a standard design valve is satisfied provided $N_a \geqq 2000$ cycles. $N_a$ is found by entering the appropriate fatigue S-N curve for the material (curves taken from ASME Section III) with the value of peak stress, $S_p$. $S_p$ is defined as the larger of $S_{p1}$ or $S_{p2}$ determined by the equations:

$$S_{p1} = \frac{2}{3} Q_p + \frac{P_{eb}}{2} + Q_{T2} + 1.3\ Q_{T1}$$

$$S_{p2} = 0.4\ Q_p + \frac{K}{2} (P_{eb} + 2Q_{T2})$$

In the above, the $Q_p$ term represents stresses due to pressure; $P_{eb}$ stresses due to external moment; $Q_{T1}$ and $Q_{T2}$ stresses due to thermal gradients. Again, approximate but presumably conservative formulas are given in the Pumps and Valves Code for evaluation of these terms.

The Pumps and Valves Code also gives rules for valve design other than the body. Design of bolted-flanged joints and by-pass piping are referred to the design rules of USAS B31.7.

At this time (November, 1969), the Pumps and Valves Code does not include specific design guidance for pumps. It states that

> "During the period of time until the rules on design of pumps have been prepared any design method which has been demonstrated to be satisfactory for the specified design conditions shall be used."

# 11. REFERENCES

(11.1) Jeffrey and Hanlon, "Bending and Impact Tests of Cast Iron, Cast-Steel, and Nodular-Iron Valve Bodies", Mechanical Engineering, Vol. 78, No. 1, pp 23-27 (1956).

(11.2) Pearson, G. H., The Design of Valves and Fittings, Pitman & Sons, London (1953).

(11.3) Lawton, C. W., Voegli, E., and Tuppeny, W. H., "Design and Stress Evaluation of Non-symmetric Pressure Vessels", Presented at SESA Meeting, Houston, Texas, October, 1969.

(11.4) Design of Piping Systems, The M. W. Kellogg Company, John Wiley & Sons, New York (2nd Edition, 1956).

(11.5) Lind, N. C., "Approximate Stress-Concentration Analysis for Pressurized Branch Pipe Connections", ASME Paper No. 67-WA/PVP-7 (1967).

(11.6) Edwards, T. W., "Valves", Power, pp 69-92 (June, 1961).

(11.7) Rodabaugh, E. C., "Ratings of ASA B16.5 Flanges", Report to USAS Sub-committee B16.4 (Pressure-Temperature Ratings, Materials and Stresses for B16 Components).

(11.8) Milleville, B. J., "A New Look at Valve Design and Stresses", ASME Paper No. 67/MET/15 (1967).

(11.9) Handbook of Valve Information, The Wm. Powell Company, Cincinnati, Ohio 45214.

(11.10) Technical Reference Book on Valves for the Control of Fluids, The British Valve Manufacturers' Association, 24 Victoria Street, London SW1, England.

(11.11) "ASME Standard Code for Pumps and Valves for Nuclear Power", Draft ASME Standard, November, 1968, issued for Trial Use & Comment, Published by the American Society of Mechanical Engineers, 345 E. 47th Street, New York, N. Y. 10017.

(11.12) "ASME Boiler and Pressure Vessel Code, Section III, Nuclear Vessels", 1968, Published by the American Society of Mechanical Engineers, 345 E. 47th Street, New York, N.Y. 10017.

(11.13) "USA Standard Code for Pressure Piping, USAS B31.7, Nuclear Power Piping", Draft-January, 1969, Published by the American Society of Mechanical Engineers, 345 E. 47th Street, New York, N.Y. 10017.

(11.14) "ASME Boiler and Pressure Vessel Code, Pressure Vessels Section VIII, Div. 1", 1968, Published by the American Society of Mechanical Engineers, 345 E. 47th Street, New York, N. Y. 10017.

REFERENCES (contd.)

(11.15) USA Standards Institute, Code for Pressure Piping, USAS B31.1.0-1967, "Power Piping", USASI, New York, N. Y., Published by the American Society of Mechanical Engineers, 345 E. 47th Street, New York, N. Y. 10017.

(11.16) Hsu, L. C., Pump Analysis Study, General Electric Co., APED, Report 257HA842, General Electric Co., San Jose, California.

(11.17) Hopper, A., and Zudans, Z., "Comparison of Theoretical and Experimental Stress Analysis Results for a Pump Casing", Paper presented at Oak Ridge National Laboratories, April, 1969, "Studies in Applied Mechanics Meeting", Franklin Institute Research Laboratories.

(11.18) Reid, R. T., "Analytical Considerations for LMFBR Technology, Developed from Aerospace Technique", April, 1969, AGC-NRO 1367, Aerojet-General Corporation, Sacramento, California.

(11.19) Ball, R. E., and Clasby, G., "Structural Analysis of Volute-Type Pump Casings", Paper presented at Oak Ridge National Laboratories, April 19, 1969, "Studies in Applied Mechanics Meeting", Byron Jackson Pump, Inc., Los Angeles, California.

(11.20) Maloney, J. R., Westinghouse Electric Corp., Atomic Power Division, Cheswick, Pa., Private Communication to ASME Special Committee on Design of Pumps and Valves, May 9, 1968.

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# 12. BOLTED FLANGED JOINTS

## Introduction

Since the introduction of welding-end valves, bolted flanged joints are no longer used much in critical-service piping. However, there are some locations where, for purposes of construction or maintenance, a bolted flanged joint would be convenient or, in some cases, necessary.

The contents of this chapter are devoted primarily to flanged joints as used in pipelines; however, much of the theory and test data were developed for flanges in pressure vessels. The difference involves the greater degree of standardization of pipeline flanges and additional loadings that may be imposed on pipeline flanges. The theory and test data presented herein are restricted to axisymmetric flanges (i.e., circular); actual pipeline and pressure vessel flanges are usually* axisymmetric except for the bolt holes and local load application by the bolts.

The significant characteristics of bolted-flanged joints are different than those of most other piping components (e.g., pipe, elbows, tees) in that the failure criterion is usually leakage, not rupture of any part nor, necessarily, yielding of any part. The major stresses usually are developed by tightening the bolts in assembly of the joint. Subsequent application of internal pressure may not increase the stresses in the flanges or bolts significantly; in fact, for typical pipeline flanged joints, internal pressure usually reduces the bolt stresses. Cyclic stresses in flanged joints due to cyclic internal pressure or cyclic loads applied

---

* Some pipe flanges are made with gussets or large fillets between bolt holes.

by the pipe to the joint are usually small; provided there is an adequate initial bolt load. An exception consists of the region joining the flange to the pipe, where the stresses may vary significantly with pressure or pipe loads. Thermal gradients produce significant stresses which can, and often do, lead to leakage of the joint.

Accordingly, the analysis of flanged joints involves not only stresses but also small, elastic deformations; since these may lead to leakage. Test data are similarly concerned not only with stresses and load capacities, but also the determination of loadings which produce joint leakage and with the measurement and interpretation of small, elastic deformations.

## Nomenclature

This chapter does not give details of theoretical developments or experimental data. However, there are a number of terms which define bolted-flanged joints which may not be familiar to some readers. Accordingly, these terms are defined as follows:

Loads:

Initial bolt load - load applied by tightening bolts.

Residual bolt load - load in bolts after application of other loads or at some time subsequent to application of the initial bolt load. The residual bolt load can be either lower or higher than the initial bolt load.

Pressure - internal pressure applied inside the joint and presumed to act axially as well as radially.

Pipe Loads - see Figure 12.1.

Thermal Gradient - those (self-equilibrating) loads which arise due to temperature differences in a flanged joint.

Types of Flanges:

Integral: flange and pipe or pressure vessel shell are a continuous structure.

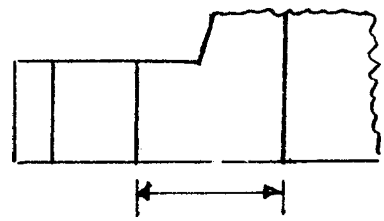
Loose: flange is separate from the pipe or vessel shell.

Welding Neck
Slip-on
Blind
Lap Joint
Ring

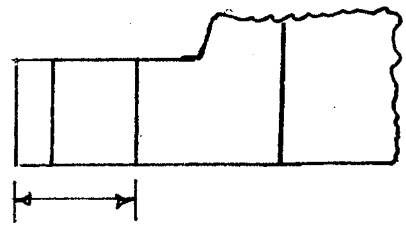
See Figure 12.2

Gaskets:

Inside Contact:



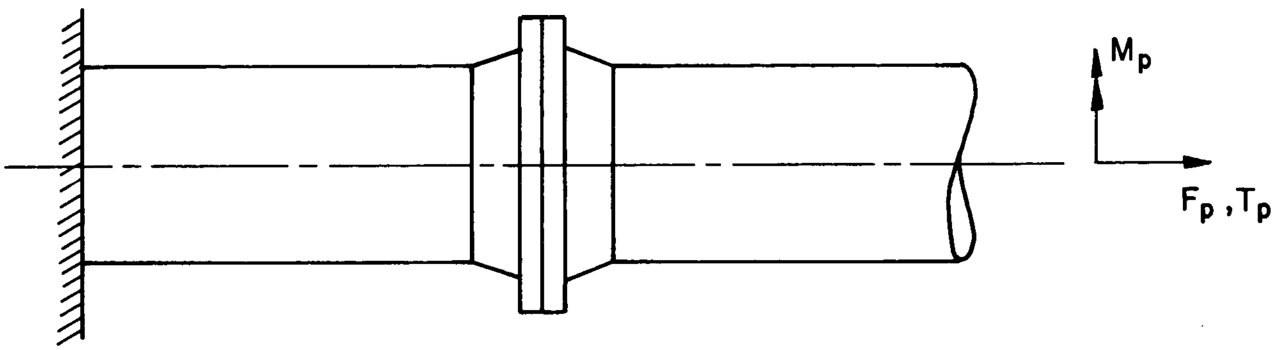

Full Face:




Flat: gaskets which have essentially no "self-sealing" characteristics e.g., a 1/16" thick compressed asbestos gasket. Prevention of leakage requires maintenance of some (usually small) load on the gasket.

Self Sealing: gaskets in which pressure aids in maintaining a seal. e.g., elastomeric O-rings, lens ring, bellows gaskets.

Ring: The term "ring gasket" used herein refers to metal ring gaskets which fit into grooves in the flange faces. These are standardized gaskets in USAS B16.5.




$M_p$ = bending moment (in.-lb.)

$T_p$ = torsional moment (in.-lb.)

$F_p$ = axial force ( other than that due to internal pressure ), lb.

FIGURE 12.1 PIPE LOADS ON FLANGED JOINT

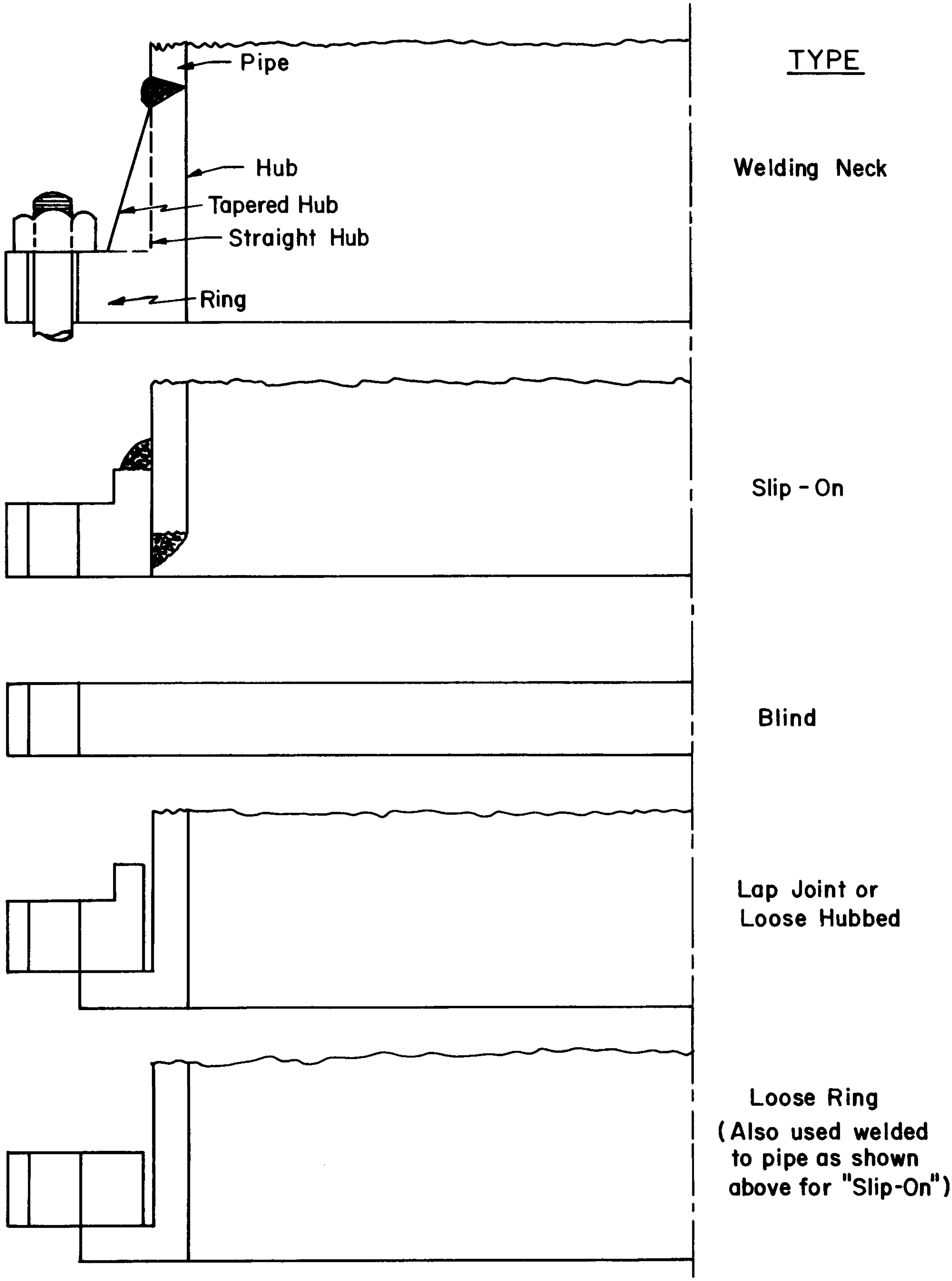



FIGURE 12.2 SOME COMMON TYPES OF FLANGES

## 12.1 Leakage of Flanged Joints

Because leakage of a flanged joint is the normal failure criterion, some discussion of such leakage is pertinent.

### 12.11 Joints with Inside Gasket Contact, Flat Gaskets

Most pipeline flanged joints are made using gaskets inside the bolt holes; the gasket being a flat annular ring made of, for example, asbestos or a spiral-wound metal strip (on edge) with an asbestos filler. Such gaskets have essentially no "self-sealing" characteristics. For such flanged-joints with pressure loading only, one may ideally describe the leakage characteristics as being analogous to a spring-loaded relief-valve. That is, there is some pressure at which the axial force due to pressure is equal to the bolt load. At this pressure, which we will call the "leakage pressure", the residual load on the gasket becomes equal to the internal pressure and a further small increase in pressure will result in profuse leakage. Below this pressure the leakage will be (ideally) zero. The analogy indicates that the leakage pressure should be proportional to the initial bolt load, corresponding to the spring setting on a relief-valve.

Experimental data shows that, under certain conditions, this ideal behavior is approached by actual flanged joints.

The first necessary condition is that sufficient initial bolt load must be applied to bring the gasket and flange surfaces into essentially complete contact with each other everywhere. This is the loading implied by the ASME Code[12.1] gasket y-factor. It is perhaps obvious that the y-factor must depend not only on the gasket material but also on the surface planeness and scratches of the flange faces. While ideally the leakage is zero for pressures below the leakage pressure, test results indicate that there is often some seepage of fluid around or through the gasket. This kind of seepage can be reduced to very low magnitudes by applying very high seating loads. Accordingly, the gasket y-factor also depends upon what is a tolerable seepage rate, which in turn depends upon the fluid and its temperature and pressure (e.g., water, steam, high pressure helium).

A second condition to achieve the "ideal relief-valve" behavior is that the flange rigidity must be sufficient to distribute the localized bolt loads to a nearly uniform load at the gasket surface.

A third condition, which leads into consideration of the ASME Code[(12.1)] gasket m-factor, is that the "seating" of the gasket must be plastic in nature; so that when the gasket load is partially removed by internal pressure, local seepage paths do not develop. Use of the gasket m-factor implies that if a certain load is kept on the gasket,an intolerable degree of seepage will not develop. Actually, test data indicates that the m-factor is dependent upon the seating load; and with adequate seating load the m-factor becomes equal to 0.5 for any "flat" gasket*.

There is, however, a good reason for using an m-factor greater than 0.5; although the phenomenon involved is related to the flanged joint as a whole rather than the gasket. By the relief-valve analogy, the leakage pressure is defined as that pressure which produces an axial force that balances the bolt load. However, the bolt load with internal pressure is not necessarily the initially applied bolt load. Wesstrom and Bergh[(12.2)]

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* In the equation for leakage pressure $P_L$; $P_L = W/[.785G^2 + 2\pi b(G + b)\ m]$, $m = 0.5$ is a theoretical minimum for a flat gasket; where $W$ = total bolt load, $G$ = mean gasket diameter, $2b$ = gasket width.

developed the theory for determining the bolt load in a flanged joint, which can be expressed by the equation:

$$W_P = W_i + \alpha P \tag{12.1}$$

where $W_P$ = bolt load at pressure, P

$W_i$ = initial bolt load

$\alpha$ = a parameter which depends upon the flange, gasket and bolting dimensions and material properties

P = internal pressure.

For most pipeline flanged joints, test data indicate that $\alpha$ is a negative number, i.e., the bolt load decreases as pressure increases. The test data roughly check with theoretical calculations, particularly with the use of an additional factor in Wesstrom & Bergh's equations, suggested by Rodabaugh[12.3]. Accordingly, use of an m-factor greater than 0.5 compensates to some extent for a negative value of $\alpha$, although the compensation is quite arbitrary and may be unconservative in some flanged joints, overconservative in others.

## 12.12 Joints with Full-Face Contact, Flat Gaskets

With gasket contact outside the bolt circle, the leakage pressure becomes a complex function not only of the bolt load but also of the deflection of the flanges, the flange bolts and the gasket. The deformation properties of common non-metallic gasket materials are not well known; hence it is difficult to analytically predict the leakage pressure. However, a great deal of experience exists with such gaskets used in cast iron flanged joints; these are proportioned so that usually satisfactory service is obtained. One might note that leakage of such joints normally occurs through the bolt holes and that, as compared to an equivalent inside-contact joint, much higher initial bolt loads are required for a given leakage pressure.

## 12.13 Joints with "Self-Sealing" Gaskets

The elastomeric O-ring is the most commonly used self-sealing gasket, although it is not used very much in pipeline flanged joints. For flanged joints, the design problem for elastomeric O-rings becomes that of:

(a) Providing sufficient rigidity and/or initial bolt load so that a gap does not develop which would permit the O-ring to extrude to the extent that the gasket "blows-out".

(b) For joints subjected to rapidly varying loads (e.g., a pipeline flanged joint with vibration in the piping) the design must be such as to keep the faces in contact. Even a small gap may result in progressive deterioration of the gasket and eventually to leakage.

The use of elastomeric O-rings are limited by temperature, roughly to the range of -100 F to + 250 F. There are two major advantages of elastomeric O-rings.

1. With an adequately designed flanged-joint and carefully machined O-ring groove and mating surface, the O-ring produces almost a perfect seal; even high pressure helium will only diffuse slowly through the elastomer.

2. If rapidly varying loads are not a design consideration, the initial bolt load can be almost zero. This is a particular advantage in a joint that must be opened periodically because the labor involved in loosening and tightening large flange bolts to a high stress can be significant.

On the other hand, it should be recognized that for high pressure joints, the flanges and bolts must be able to withstand almost the same load as the analogous joint with flat gaskets. As compared to joints with flat gaskets, O-rings involve additional machining costs and are perhaps less reliable in the sense that when a joint with a flat gasket develops a leak, that leak can often be stopped by additional tightening of the bolts; this usually cannot be done with an elastomeric O-ring gasket.

As mentioned previously, pipeline flanged joints seldom make use of elastomeric O-rings. However, there are two kinds of gaskets used in pipeline flanges which do have some degree of self-sealing. The first is known as the "ring-joint gasket", which consists of an oval or octagonal shaped metal ring which fits into grooves in the flange faces. This is widely used, particularly in the petroleum industry, and is a standardized type of gasket in USAS B16.5[12.4]. The second is the "lens-ring" gasket, which is occasionally used for very high pressures and is standardized in MSS SP-65.[12.70] There are many other types of metal gaskets which have some degree of "self-sealing"; none of which have obtained much popularity in pipeline flanged joints.

The leakage problem of flanged joints may also be overcome by means of a "seal weld". Usually, this involves a small weld between two relatively flexible lips extending around the periphery of the flange raised faces.

## 12.2 Analysis of Bolted-Flanged Joints

### 12.21 Analysis Classification

Flanged joints may be divided into two broad classifications for theoretical analysis:

(1) Joints with mating flange contact only inside the bolt circle.

(2) Joints with mating flange contact both inside and outside the bolt circle.

The majority of flanged joints in steel piping systems are made with USAS B16.5 [12.4] flanges. These flanges have a raised face inside the bolt circle, hence contact occurs only inside the bolt circle. Cast iron flanges normally do not have a raised face and the gasket normally covers the entire flange faces, hence contact occurs both inside and outside the bolt circle.

Theoretical analysis of flanged joints with contact inside the bolt circle is relatively well advanced. Flanged joints with contact inside and outside the bolt circle involve additional statical indeterminacies; theoretical analysis for such joints is not as well developed.

### 12.22 Analysis of Flanged Joints with Inside Contact

The analysis procedure given in the ASME Code[1] is representative of the present state of the art of flanged joint design. The design procedure consists of two steps.

(1) Calculation of the required bolt load, from which the minimum total bolt root area is established.

(2) Selection of flange dimensions, such that the flange has the necessary strength to withstand the bolt load.

These two steps are discussed in the following subsections, 12.211 and 12.222.

### 12.221 ASME Required Bolt Load

The bolting selected for the flanged joint must be sufficient to:

(a) withstand the axial force created by the internal pressure and also to maintain a specified residual load on the gasket. This is expressed by the equation,

$$W = \frac{\pi}{4} G^2 P + 2\pi bGmP \tag{12.2}$$

(b) provide an initial load sufficient to "seat" the gasket. This is expressed by the equation

$$W = \pi bGy \tag{12.3}$$

In equations (12.2) and (12.3):

W = total bolt load, lb

G = effective gasket diameter, in.

P = internal pressure, psi

b = effective gasket width, in.

m = residual load gasket factor

y = seating load gasket factor.

The significance of equations (12.2) and (12.3) are discussed in the preceding section on "Leakage of Flanged Joints". The definitions of G, b, m, and y were proposed by Rossheim and Markl[12.5] in 1943 and subsequently were introduced into the ASME Code. While considerable controversy exists concerning these gasket factors, the values given by Rossheim and Markl are still used in the 1965 edition of ASME Code, Section VIII.

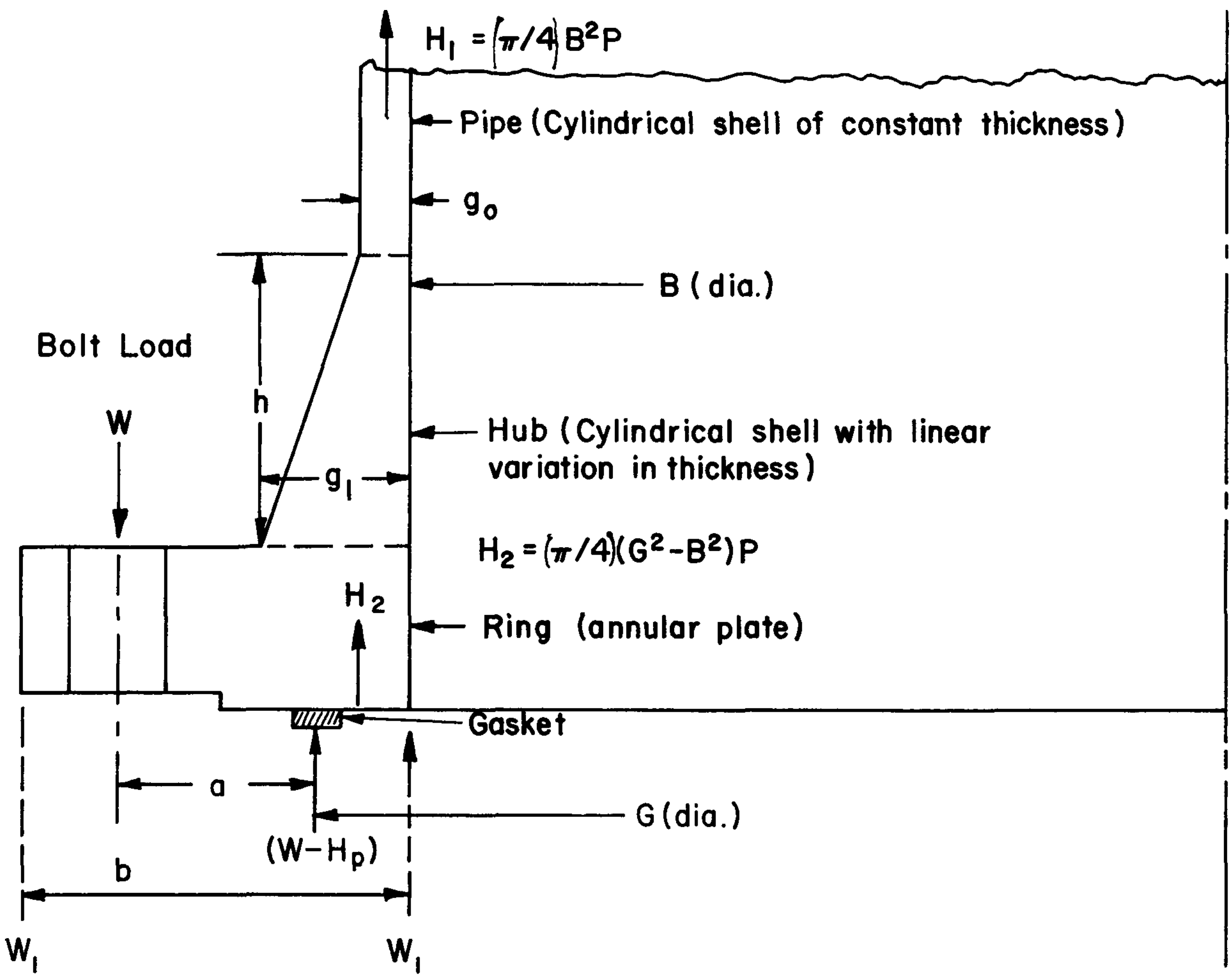
Having established a minimum required bolt load as the greater of the loads by Equations (12.2) and (12.3), one may proceed to the next step in the ASME Code design procedure; establishing a set of flange dimensions so the flanges will have adequate strength to withstand the imposed bolt load.

### 12.222 ASME Flange Strength Analysis (Bolt Loading)

The early history and development of the analysis of the load capacity of flanges is described by Waters, Wesstrom, Rossheim, and Williams[12.6]. This reference also outlines the flange strength analysis method used in the ASME Code[12.1].

In this analysis, the flange is considered as made up of plates and shells, joined together by the usual continuity assumptions. The bolt load is considered as a line load and the weakening produced by the bolt holes is ignored. As a further simplification, the flange ring is assumed to be loaded at the outer and inner edges by line loads which give a statically equivalent moment on the flange ring (see Figure 12.3).

The most general case of flange configuration is shown in Figure 12.3. Simplifications can be made for certain types of flanges, for example,




$W_1 = \frac{Wa}{b}$, for analysis

$H_p = H_1 + H_2$

$h_o = \sqrt{Bg_o}$

FIGURE 12.3 ILLUSTRATION OF FLANGE CONFIGURATION AND ASSUMPTIONS IN ANALYSIS, FLANGED JOINT WITH INSIDE CONTACT GASKET

Loose Ring Flange: Omit hub and pipe. The resulting stress equation is very simple; see Equation (9) of Reference (12.1).

Loose Hubbed Flange: Omit pipe, hub can be straight or tapered.

The relatively complex analysis is reduced to simple design equations by means of graphs of T, U, Y, Z, (flange ring parameters) and F, V, f, $F_L$, and $V_L$ (hub parameters). The hub parameters are (by a suitable choice of continuity boundary conditions) functions of two parameters: $h/h_o$ and $g_1/g_o$. The f-parameter is the ratio of stress at the thin-end of the hub to the stress at the thick-end of the hub; only the stresses at the hub ends are evaluated.

It is pertinent to note that the analysis is of an axisymmetric structure. Any axisymmetric shell computer program would be expected to duplicate the ASME Code analysis method. The writer has used the MOLSA(12.7) computer program to duplicate and extend the ASME analysis.

The design requirements of the ASME Code are satisfied if the average tensile stress in the bolts is less than the allowable bolt stress at the design temperature and if the maximum calculated flange stresses are less than the allowable stress for the flange material, except that the longitudinal hub stress can be equal to 1.5 times that stress. The later provision is significant in that it is an explicit provision in the ASME Code Section VIII which permits secondary type stresses to exceed the basic allowable stress by a specified amount.

It is recognized that, in actual installation, the bolts are usually prestressed to a much higher stress than the allowable bolt stress.

This load condition is almost ignored; however, some slight consideration is given in that the seating bolt load is:

$$W = \frac{(A_m + A_b)S_a}{2} \tag{12.4}$$

where $W$ = gasket seating bolt load

$A_m$ = minimum required bolt area

$A_b$ = actual bolt area

$S_a$ = allowable bolt stress at atmospheric temperature.

The preceding summarizes the analysis method given in the ASME Code. There are several quite significant aspects of flanged joint designs which are not covered by the ASME Code. These are discussed in the subsequent subsections 12.223 through 12.227.

12.223 Internal Pressure Loading

The ASME Code analysis gives stresses due to a moment applied to the flange ring. At "operating conditions", the moment is dependent upon the pressure. However, the stresses due to pressure are not included. For example, at the thin-end of the hub the longitudinal hub stress is:

$$S_H = \pm \frac{fM_o}{Lg_1^2B} + \frac{PB}{2g_o} \pm (S_b)_p \tag{12.5}$$

where $\frac{fM_o}{Lg_1^2B}$ = bending stress due to moment, $M_o$, which is given by the ASME Code analysis

$\frac{PB}{2g_o}$ = membrane stress due to internal pressure

$(S_b)_p$ = bending stress due to internal pressure.

The last two terms in Equation (12.5) are not given by the ASME Code analysis.

In addition to stresses due to internal pressure, the corresponding displacements may be significant in changing the bolt load. This aspect is discussed in section 12.11.

Internal pressure loading was considered by Waters, et.al.(12.6) and their method of analysis can be readily extended to include internal pressure loading. The writer has prepared a computer program(12.71) which includes internal pressure loading. Also, axisymmetric shell computer programs can be used to calculate stresses and displacements due to internal pressure.

### 12.224 Thermal Gradients

Leakage of flanged joints due to thermal gradients is sometimes encountered in field installations. The cause is often ascribable to the temperature difference between the bolts and the flange ring. A sudden increase in fluid temperature produces a transient condition during which there is a relatively higher temperature in the flange ring than in the bolts; consequently there is an increase in bolt load. Leakage will not occur at this time, however, if the bolt load is increased to the extent that yielding occurs, leakage may occur later as the bolt temperature approaches the flange ring temperature. A sudden decrease in fluid temperature may reduce the bolt load to the extent that leakage occurs.

Temperature differences between flange ring and the hub and pipe may also exist during a fluid temperature transient. The hub then wants to expand (or contract) with respect to the flange ring, resulting in an additional moment on the flange ring and changes in the bolt stress.

A principal advantage of flanged joints with inside contact gaskets is their better ability to withstand thermal gradients of the type described above. This is because the flange ring is able to act as a spring, absorbing part of the thermal displacements and thereby maintaining a more uniform bolt load, as compared to an analogous flanged joint with full face gaskets.

Part of the thermal gradient problem in flanged joints is discussed by Dudley[12.8]. The writer has prepared a computer program[12.71] which includes both the thermal gradient effects discussed above.

### 12.225 Loads Applied by Attached Pipe

The types of loadings involved are shown in Figure 12.1. Two aspects of flanged joint design are significant:

(1) Loads which produce joint leakage.

(2) Loads which produce excessive stresses.

#### Joint Leakage

The load $F_p$, if positive, can be considered as an addition to the ASME Code[12.1] load H. Ordinarily, the axial force in pipe lines is not a major joint design problem. The moment $M_p$ can be and sometimes is sufficient to cause joint leakage. Because this loading is not axisymmetric, it cannot be evaluated directly by the ASME Code analysis.

One approach which has been used assumes that the maximum axial tensile stress produced by $M_p$ ($S_{max} = M_p/Z$) exists everywhere around the circumference of the pipe. This can then be converted to an equivalent internal pressure and added to the ASME Code load H. The torsional moment $T_p$ does not produce any reduction in gasket load. It is resisted by frictional forces, or if frictional forces are not sufficient, by shear forces in the bolts.

The above procedure is probably a conservative method of evaluating leakage for a few cycles of load application. However, if the loads are cyclic, the possibility of progressive deterioration of the gasket must be considered.

Blick[12.9] discusses the problem of bending moments at flanged joints and gives suggested design procedures.

## Stresses

In following the philosophy of the ASME Code, the flange bolting should be sufficient so that the necessary initial load can be applied to withstand the pipe loads as well as internal pressure. Accordingly, inclusion of $M_p$ and $F_p$, which would then result in higher stresses due to the initial bolt load. Variation in flange stresses due to loads $M_p$ and $F_p$ would then be relatively small; and could be obtained by an analysis similar to that used to develop Equation (12.1) herein.

However, at the area of attachment of flanges to the pipe (butt welded, fillet welded, threaded, etc.), a significant stress may be present under cyclic loading conditions. These stresses can be estimated from stress intensification factors for the appropriate geometries; e.g., a girth butt weld, a fillet weld, or a threaded joint. Markl and George(12 give test data on these factors for the $M_p$ loading.

## 12.226 High Temperature Relaxation*

At operating temperatures where significant creep occurs, the analysis method should take into account the relaxation of the elastically stored forces which occur as a result of plastic flow. In the ASME Code analysis, this relaxation effect is taken into account in an indirect and approximate manner by the use of allowable stresses which are based on creep or stress-to-rupture properties of the material. These allowable stresses, however, do not necessarily reflect the relaxation characteristics of a bolted-flanged joint as a structure, and use of the ASME method may result in excessively conservative design or inadequate performance over the desired service life.

Reference (12.11) through (12.16) give discussions and simplified analysis methods for relaxation conditions in flanged joints. Reference (12.17) gives similar analytical methods, somewhat more general in application and also considers the important effect of "first stage" creep.

## 12.227 Bolt Holes and Local Loads

In the analytical methods discussed in the preceding, an assumption is made that the actual locally applied bolt loads can be replaced by an equivalent circular line load and that the bolt holes do not weaken the flange. For most flanges joints this is a valid approximation.

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* This is a relaxation rather than a creep problem, because the plastic flow reduces loads and stresses as a function of time.

However, in some standard flanges the bolt hole spacing is probably the cause of occasional field problems; e.g., the 3" and 8" sizes of USAS B16.5 150 lb flanges. There are two aspects of the problem:

(1) The bolt holes should be sufficiently close together so that the load on the gasket midway between bolts is not much less than the load at the bolt holes.

(2) The bolt holes should be sufficiently far apart so that they do not significantly weaken the flange as a load carrying device.

Roberts [12.18] discusses the first aspect, giving an approximate method for bolt hole spacing based on a "beam-on-elastic foundation" approach. Taylor Forge Design Manual [12.19] gives an empirical equation for maximum bolt spacing:

$$\text{Bolt Spacing (max)} = 2a + \frac{6t}{m + 0.5} \tag{12.6}$$

where a = diameter of bolts

t = flange ring thickness

m = ASME Code gasket factor.

Other empirical formulations are given by Labrow[12.20], Hill, et.al.[12.21], British Standard B.S. 1500[12.22], and Lake and Boyd[12.23].

The weakening caused by bolt holes is discussed by Bernhard[12.24]; he gives both an analysis method and test data for this effect. Ordinarily, weakening due to closely spaced bolt holes is not a problem because the minimum spacing is controlled by the clearances needed for the wrench head used in tightening the nuts.

## 12.23 Analysis of Flanged Joints with Full-Face Contact

Figure 12.4 illustrates a flanged joint with a full face contact. As compared to the inside contact, Figure 12.4, the full face contact involves a more complex distribution of reactions across the contact surface. Once these reactions are determined, the deformations and stresses of the flanges and bolts can be calculated. Rossheim and Markl[12.5] suggest one method of estimating the contact force distribution for full-face contact joints. Taylor Forge Co.[12.25] gives a basis for the design of flanges for full-face contact flanged joints; this involves a modification of the method proposed by Rossheim and Markl. These approaches employ the design procedure of the ASME Code[12.1]; however, the loading is modified so that the gaskets do not contribute to the moment causing flange rotation. These approaches were only intended as guides until a more accurate procedure could be established.

Murray and Stuart[12.26] give the theoretical development for full-face contact joints, with the assumption that the outer-contact occurs as a line load at the flange ring outside perimeter. Malkmus[12.27] also develops the theory with a similar assumption. Levy[12.28] developed an approximate method based on considering the flange ring as a cantilevered beam. He assumed that the line contact occurs where the slope of the beam is calculated to be zero under the imposed loads. This concept was developed further by Schneider[12.29] and by Waters and Schneider[12.70] who give test data to confirm Levy's hypothesis. These developments are applicable to a joint consisting of a pair of integral flanges with uniform wall hubs. The gasket is assumed to be rigid; the analysis would be applicable to a joint with an elastomeric O-ring seal and with full-face

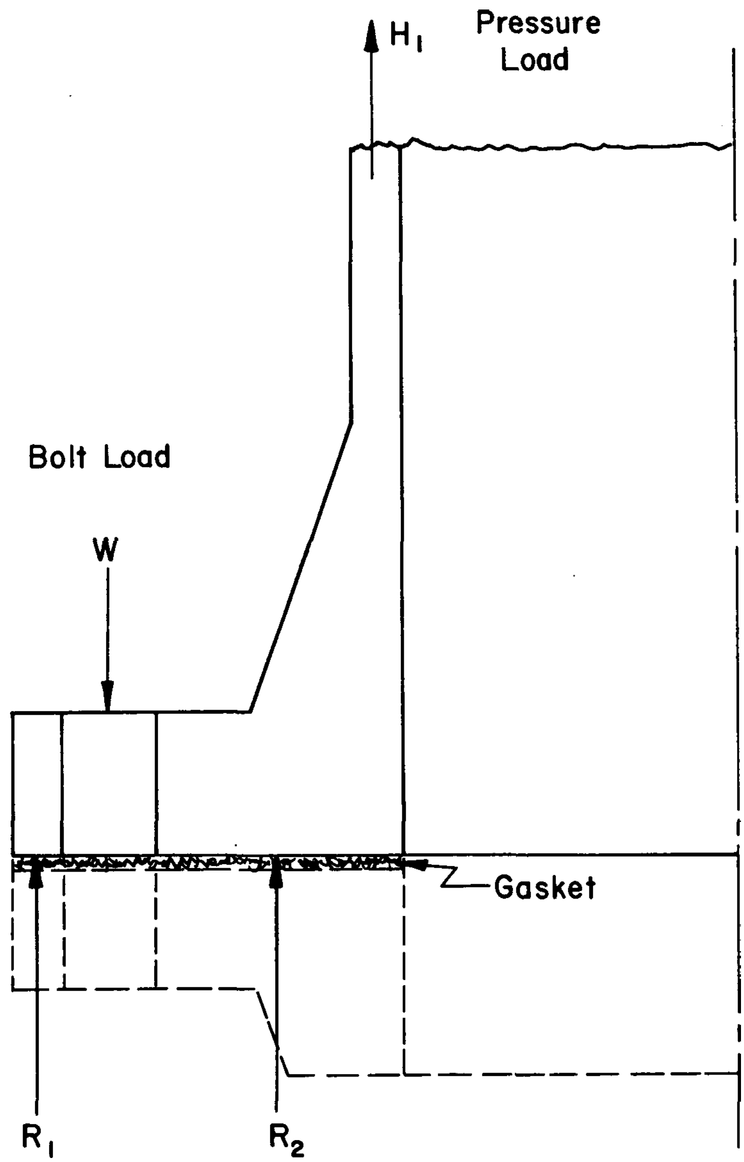


FIGURE 12.4 ILLUSTRATION OF FLANGED JOINT WITH FULL FACE GASKET AND STATICALLY INDETERMINATE REACTIONS, $R_1$ AND $R_2$

metal-to-metal contact between flange faces. Work is underway to extend the approach to tapered-hub flanges and to gaskets which are not rigid.

## 12.24 Fatigue Considerations

The ASME analysis method[12.1] is not suitable for determination of the variation in stresses due to variation in pressure, external loads or thermal gradients. Reference (12.71) gives details of a computer program for inside contact flanged joints which can be used to determine variations in stresses in the flanges and bolts due to pressure or thermal gradients. The theory presented in References (12.29) and (12.70), with some expansion, may be applicable for flanges with full-face metal-to-metal contact. A Piping Design Manual prepared by Teledyne Materials Research for Oak Ridge National Laboratory under the RDT Standards Program contains an Appendix C-1 which discusses the analytical techniques of bolt load determination in detail. This manual has been submitted in preliminary form to ORNL.

For flanges with inside contact, a number of calculations made by the writer on various USAS B16.5 flanged joints indicate that variation in flange and bolt stresses with internal pressure variations are relatively small. However, thermal gradients of moderate magnitude can produce quite large variations in bolt and flange stresses. External bending loads would be expected to produce stress variations analogous to those caused by internal pressure. In addition, the fatigue test data given in Reference (12.10) give information on the cyclic life under bending moments of the connection between pipe and flange.

## 12.3 Test Data on Bolted-Flanged Joints

The earliest test data known to the writer are contained in a report by Tanner(12.30) to the Working Committee of Sub-Committee No. 3 of the Standardization of Pipe Flanges and Fittings (November, 1923). This was part of an effort to standardize steel pipe flanges and which led, in 1927, to the publication of ASA B16.e. This standard eventually evolved into the present USAS B16.5. These tests were run on simulated loose ring flanges, simulated bolt loading was applied in increasing steps; the deflection of the flange and the "yield load", as determined by onset of a large increase in deflection per unit load, were determined. In 1924 through 1926, similar tests were run on simulated loose ring flanges as well as simulated loose hubbed flanges; these test results are contained in a paper by Waters and Taylor(12.31).

In 1936, the first of three reports of the (British) Pipe Flanges Research Committee was published by Gough(12.32). The second report by Tapsell(12.33) was published in 1939; the third and last report by Johnson(12.34) in 1954. These series of reports contain test data on many aspects of flanged joint performance. In particular, they contain the only available test data on the behavior of bolted-flange joints at temperatures in which creep (or relaxation) is the predominant cause of failure (leakage).

In 1936, Jasper, Gregersen, and Zoellner(12.35) published test data on flanges modeled using plaster of paris. This material was found to behave elastically under load up to the rupture load, with a modulus of elasticity of 800,000 psi. A significant range of sizes and proportions of both ring and hubbed flanges were tested by simulated bolt load, the deflections were measured and compared with theory.

In 1937, Petrie(12.36) published extensive test data on steel ring flanges of various diameters, bolting and ring thickness. Asbestos-composition, copper and oval- or rectangular-ring gaskets were included in the tests. Leakage pressures were determined as a function of initially applied bolt load.

The tests described above were on models typically representative of pipeline flanges. Pressure-vessel flanges are, of course, similar to pipeline flanges but usually the ratio of flange O.D. to flange I.D. is smaller than for typical pipeline flanges. In 1938, Rossheim, Gebhart, and Oliver(12.37) published results of tests on heat-exchanger flanges. Deflections of the flange due to both bolt load and internal pressure were determined. Also the change in bolt stress due to internal pressure was determined by measuring the change in length of the bolts. Stresses in the flange hubs due to both bolt load and internal pressure were determined by means of Huggenberger tensometers.

Waters and Williams(12.38) give test data for relatively thin flanges; 6" and 12" nominal pipe size. Bolt and flange stresses were measured by means of SR-4 strain gages. Barnard(12.39) gives test results for similar flanges, 6", 12", and 36" nominal pipe size. In addition to measured strains, Barnard gives leakage pressures.

George, Rodabaugh, and Holt[12.40] give test data for 8" - 150 lb and 12" - 300 lb USAS B16.5 flanged-joints made of steel or aluminum. The purpose of the test program was to compare aluminum flanged joints with steel flanged joints. Data includes flange deflection as a function of initial bolt load, leakage pressure as a function of initial bolt load and residual bolt stress as a function of initial bolt stress and pressure. Murray and Stuart[12.26] give results of pressure tests on a large (15 ft diameter) tapered hub, welding neck flange. Stresses and displacements were measured, both with an inside contact gasket and with a simulated full-face gasket. Because of the large size of the test specimen, stresses and deflections could be measured with good accuracy. This reference appears to contain the best set of test data available for checking test data with theory.

References (12.41) through (12.47) are a series of reports of tests run at Tube Turns (Division of Chemetron). These reports give data on the pressure capacity of a number of different sizes and types of USAS B16.5 flanged joints.

The references cited above are concerned with flanged joint performance with internal-pressure loading. For pipeline flanged joints, an equally significant loading often consists of the forces applied to the flanged joint by the attached pipe. Test data on this kind of loading are relatively scarce. Tests reported by Rodabaugh[12.47] are on 4" - 300 lb USAS B16.5 flanged joints. The results indicate significant reduction in leakage pressure for a relatively small bending moment; e.g., 25,000 psi bending stress in the attached 4" std. wt. pipe. George and Rodabaugh[12.48] give results of bending moment tests on 12" - 150 lb USAS flanged joints using full-face gaskets, along with results of tests on "tapered face" flanges.

## 12.4 Bolting and Gaskets

### 12.41 Bolting

The literature on bolting is considerably more voluminous than that on bolted-flanged joints; no attempt is made herein to review all such data. In connection with flange design, two questions often arise:

(1) What is the relationship between tightening torque and axial force in the bolts? Data on the aspect is given by Lenzen(12.49), Piping Handbook(12.50), and Fastener Standards(12.51).

(2) What is a "typical" bolt stress applied by a pipefitter in tightening flange bolts with an ordinary wrench (not a torque wrench)? Petrie(12.36) gives an empirical formula for this stress, which is widely quoted and probably is fairly correct.

$$S = \frac{45,000}{\sqrt{d}} \tag{12.6}$$

where S = "typical" field installation bolt (average tensile) stress

d = bolt diameter, equation is for bolts 1" or larger, 8-thread series.

In most flanged joints, the major stress applied to the bolts is that applied in tightening the nuts. Bolts are subjected to both tensile and bending stresses, the later being dependent upon the rotational rigidity of the flanges. If tightening is done using a wrench, a torsional load and shear stresses are also imposed. The bolt stresses usually do not change much as a result of internal pressure or loads imposed by the pipe (assuming adequate initial bolt load). However, thermal gradients can cause significant bolt load changes. Under cyclic loading conditions, some consideration of the fatigue strength of the bolts may be necessary. References (12.52) through (12.60) are several of many articles on the fatigue strength of bolts. Reference (12.52) gives an extensive bibliography of fatigue data prior to 1951. It might be remarked that the details of the thread form (rolled, machined, root radius, etc.) and nut dimensions can have major effects on the fatigue life.

Bolting dimensions for pipeline flanges are fairly well standardized; usually pressure-vessel flanges use the same dimensional standards. Paragraph 6.9 of Reference (12.4) prescribes these dimensional standards.

## 12.42 Gaskets

As for bolting, the literature on gaskets is quite extensive. Unfortunately, there is not much information in the literature on the properties of gasket materials of significance to flanged joint designs. This is perhaps reflected by the fact that the ASME Code[(12.1)] gasket factors proposed by Rossheim and Markl[(12.5)] in 1943 are still used in the 1965 ASME Code, almost without change.

Probably the most widely used gasket material for flanged joints is compressed asbestos with a suitable binder, usually a rubber compound. A discussion of pertinent (to flanged joint design) properties of one such material* is given by Whalen[12.61]. Qualitative and some quantitative data on non-metallic gaskets are given by Smoley[12.62]. Similar information is given by Dunkle[12.63] on metallic and semi-metallic gaskets.

There have been several recent investigations into the fundamentals of gaskets (or seals) sponsored by government agencies (12.64-12.68). These references are also pertinent to this chapter in that design procedures for bolted-flanged joints are discussed; principally from the standpoint of achieving minimum weight design.

Some desirable characteristics of a flat gasket are implied in the discussion of leakage of flanged joints, section 12.1. A more detailed discussion of these characteristics and other aspects of gasket characteristics are:

(1) The gasket should be "soft" so that it is capable of flowing into irregularities of the mating flange faces. This characteristic is related to the ASME Code y-factor; although obviously its value is a function of the seating surfaces and tolerable leakage rate as well as the gasket material properties.

(2) The gasket should be stable under load. Non-metallic gaskets are surprisingly stable under load, provided that seating surfaces are not too smooth. For example, a compressed asbestos gasket can usually withstand compressive stresses of 45,000 psi or higher. However, with time and/or temperature increase, some creep of the gasket may occur, leading to a reduction in the initial bolt load. Retightening of bolts after some short period of service is often used to overcome this effect.

* Johns-Manville Style 60. Gasket materials generally classified as "compressed asbestos with a suitable filler" can have a large range of properties.

(3) The gasket should either have sufficient strength to resist the radial pressure load, or should be restrained by the flange facing. This is an important practical aspect in that if the load on the gasket is lost for any reason, it is preferable to have leakage rather than a "blow-out" of the gasket. In the former case, the leakage can be stopped by retightening or additional tightening of the bolts; in the latter case the line must be shut down, the joint disassembled and a new gasket installed.

(4) The gasket should be sufficiently plastic so that it continues to provide an adequate seal as load is removed. This is related to the ASME Code m-factor. It should be noted that the gasket load varies significantly with pressure or pipe loads even though the bolt load remains almost constant.

In order to include the load-deflection behavior of the gasket in a flanged joint analysis, data on that characteristic would have to be established. This is quite difficult to do because the load-deflection characteristics of, for example, compressed asbestos, depends not only on the particular type of compressed asbestos material but also upon its thickness, width and seating surface finish. The behavior is both plastic and time dependent. However, the particular characteristics of flat gasket materials are usually not too significant in the joint performance, provided only that the gasket can be "seated" by the available bolt load. This is because the strain recovery on unloading the gasket is usually quite small compared to the displacements of other parts of the joint.

## 12.5 Flange Standards

Most flanged joints in pipelines are made with standard pipe flanges. While it is possible to design flanged joints which are significantly smaller and lighter than standard flanges, it is seldom economically worth while to do so, except if the weight per se is highly significant (e.g., aerospace applications).

A significant aspect of standard flanges is that they are usually provided to attach pipe to valves, pumps, compressors, etc. Accordingly, the bolt circle, number of bolt holes and size of bolts must match between these various components. The bolt circle is a critical dimension in so far as flange weight is concerned; decreasing the bolt circle permits a major decrease in flange dimensions and weight. However, bolt-circles in standard pipe flanges were established to accomodate requirements for cast valves, and cannot be reduced without major changes in valve design. Another significant aspect of standard flanges is that they are sold as being applicable to a range of temperatures and for "typical" conditions of installation. Accordingly, they must be suitable for the range of possible operating and installation conditions; including both high and low temperatures. This militates against the use of elastomeric O-ring gaskets as a standard since these are not suitable for high temperatures.

A list of standard flanges is given in Table 12.1. The background of the pressure-temperature ratings of USAS B16.5 flanged joints is given in Reference (12.69). As discussed in Reference (12.69), the dimensions of steel flanges of USAS B16.5 were based, in part, on prototype cast iron flanges, which were already well established by 1920. Accordingly, some of the dimensions of standard flanges were developed at least 50 years ago. There is a degree of "bolt-matching" between various standards. For example, 125 lb cast iron (B16.1), 150 lb steel (B16.5), 150 lb bronze (B16.24), and 150 lb corrosion resistant case (SP-51) are interchangeable in-so-far as bolting is concerned. Similar matching occurs between 250 lb cast iron (B16.1) and 300 lb steel (B16.5), etc.

TABLE 12.1: COMMONLY USED FLANGE STANDARDS

USAS B16.5[1], "Steel Pipe Flanges and Flanged Fittings"

Contains 7 pressure classes identified as 150, 300, 400, 600, 900, 1500, and 2500 lb. Sizes 1/2" through 24". Materials: 23 ferritic and austenitic ferrous alloys and 9 non-ferrous alloys. Includes applicable requirements for flanged end and butt welding end valves.

MSS[2]-SP44, "Steel Pipe Line Flanges"

An extension in size range of USA B16.5 to the 36" size in 300, 400, 600, and 900 lb pressure classes. Intended primary for attachment to thin-wall, high-strength pipe such as used in gas transmission lines.

USAS B16.1, "Cast Iron Pipe Flanges and Flanged Fittings"

Contains 4 pressures classes identified as 25, 125, 250, and 800 lb. Sizes 4" through 96" in 25 lb, 1" through 48" in 125 lb and 250 lb, 2" through 12" in 800 lb. Ratings depend upon size in all but the 800 lb class. Materials: gray iron castings per ASTM A126 or better.

USAS B16.24, "Bronze Flanges and Flanged Fittings"

Contains 2 pressure classes identified as 150 lb and 300 lb. Sizes 1/2" through 12". Materials: 2 grades of bronze.

MSS[2] SP51, "150 lb Corrosion Resistant Cast Flanges and Flanged Fittings"

Contains 1 pressure class, 150 lb. Sizes 1/4" through 12". Material: cast austenitic stainless steel. (Flanges are thinner than 150 lb B16.5 flanges and are intended for use with full face gaskets.)

MSS SP-65, "High Pressure Chemical Industry Flanges and Threaded Stubs for Use With Lens Gaskets"

Contains 1 pressure class, rated 10,000 psi at 100 F, 4200 psi at 850 F. Sizes 3/4" - 6". Materials: A105-II forged, A216 WCB cast.

API[3]-605, "Large Diameter Carbon Steel Flanges"

Contains 3 pressure classes identified as 75, 150, and 300 lb. Sizes 26" through 60". Materials: A181 or A105 Grade II (forged), A216 Grade WCB (cast).

TEMA[4]

Contains 5 pressure classes, identified as 75, 150, 300, 450, and 900 lb. Sizes (I.D.) 6" through 47". Materials: various carbon and alloy steels and non-ferrous alloys.

Footnotes on following page

TABLE 12.1 (contd)

AWWA C207

Contains 3 pressure classes, identified as B, D, and E. Sizes 6" through 48". Materials: ASTM A181 Gr. I. (for use with cloth-inserted rubber gaskets, extending from the inside diameter of the flange to the inside edge of the bolt holes or beyond).

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(1) USAS: (ANSI) United States of America Standards Institute (formerly, American Standards Association; now American National Standard Institute), standards published by the American Society of Mechanical Engineers, 345 E. 47th St., New York, N. Y. 10017.

(2) MSS: Manufacturers Standardization Society of the Valve and Fittings Industry, 420 Lexington Avenue, New York, N. Y. 10017.

(3) API: American Petroleum Institute, 1271 Avenue of the Americas, New York, N. Y. 10020.

(4) TEMA: Tubular Exchanger Manufacturers Association, 53 Park Place, New York, N. Y.

(5) AWWA: American Water Works Association, 2 Park Avenue, New York, N. Y. 10016.

## 12. REFERENCES


(12.1) "ASME Boiler and Pressure Vessel Code, Section VIII, Unfired Pressure Vessels, Appendix II, Rules for Bolted Flanged Connections", Published by American Society of Mechanical Engineers, 345 E. 47th St., New York, N. Y., 10017.

(12.2) Wesstrom, D. B. and Bergh, S. E., "Effect of Internal Pressure on Stresses and Strains in Bolted-Flanged Joints", Trans. ASME, J. of Applied Mechanics, Vol. 73, p. 553 (1951).

(12.3) Rodabaugh, E. C., Discussion of Reference (12.2).

(12.4) "Steel Pipe Flanges and Flanged Fittings", USAS B16.5, Published by American Society of Mechanical Engineers, 354 E. 47th St., New York, N. Y. 10017.

(12.5) Rossheim, D. B. and Markl, A.R.C., "Gasket-Loading Constants", Mechanical Engineering, 1943.

(12.6) Waters, Wesstrom, Rossheim, and Williams, "Formulas for Stresses in Bolted Flanged Connections", Trans. ASME, Vol. 59, p. 161, (1937).

(12.7) MOLSA, See "Analysis of Shells of Revolution Subjected to Symmetrical and Nonsymmetrical Loads", by A. Kalnins, ASME J. of App. Mechs., Sept., 1964.

(12.8) Dudley, W. M., "Deflection of Heat Exchanger Flanges as Affected by Barreling and Warping", Trans. ASME, Series B, Vol. 83, pp. 460-466 (1961).

(12.9) Blick, R. G., "Bending Moments and Leakage at Flanged Joints", Petroleum Refiner, Vol. 29, p. 129 (1950).

(12.10) Markl, A.R.C. and George, H. H., "Fatigue Tests on Flanged Assemblies", Trans. ASME, Vol. 72, p. 77 (1950).

(12.11) Baumann, K., "Some Considerations Affecting Future Developments of the Steam Cycle", Engineering, Vol. 130, pp. 597-599, 661-664, 723-727 (1930).

(12.12) Bailey, R. W., "The Utilization of Creep Test Data in Engineering Design", Proceedings of the Institution of Mechanical Engineers, Vol. 131, pp. 131-349 (1935).

(12.13) Bailey, R. W., "Flanged Pipe Joints for High Pressures and Temperatures", Engineering, Vol. 144, pp. 364-365, 419-421, 538-539, 615-617, 674-676 (1937).

(12.14) Waters, E. O., "Analysis of Bolted Joints at High Temperature", Transactions of the ASME, Vol. 60, pp. 83-86 (1938).

REFERENCES (contd.)


(12.15) Marin, J., Mechanical Properties of Materials and Design, McGraw-Hill Book Co., Inc., New York (1942).

(12.16) Finnie, I., and Heller, W. R., Creep of Engineering Materials, McGraw-Hill Book Co., Inc., New York(1959).

(12.17) Cassidy, Rodabaugh, Roach, and Trainer, "Relaxation Design of Separable Tube Connectors", NASA Report S-67-1157, Vol. 1, March 15, 1967.

(12.18) Roberts, I., "Gaskets and Bolted Joints", Trans. ASME, J. of App. Mechs., p. 169, June, 1950.

(12.19) Modern Flange Design, Taylor Forge Co., Chicago, Ill, Bulletin No. 502, 1964 (Fifth) Edition.

(12.20) Labrow, S., "Design of Flanged Joints", Proc. J. Mech. Engrs., Vol. 156, p. 66 (1967).

(12.21) Hill, Lewis, and Easter, "Fabricated Steel Ring Flanges for Water Pipe Service for Low Pressures and Low Temperatures", Journal of the AWWA, Vol. 36, p. 968 (1944).

(12.22) Fusion Welded Pressure Vessels, British Standard 1500:1958, Part 1. Carbon and Low Alloy Steels. Published by British Standards Institute, 2 Park St., London, W.1.

(12.23) Lake, G. F. and Boyd, G., "Design of Bolted Flanged Joints of Pressure Vessels", Proc. J. Mech. Engrs., Vol. 171, p. 843 (1957).

(12.24) Bernhard, H. J., "Flange Theory and the Revised Standard B.S. 10:1962 - Flanges and Bolting for Pipes, Valves and Fittings", Proc. Instn. Mech. Engrs., Vol. 178, Pt. 1, No. 5, (1963-64).

(12.25) "The Design of Flanges for Full-Face Gaskets", Taylor Forge Co., Chicago, Illinois, 1958.

(12.26) Murray, N. W. and Stuart, D. G., "Behavior of Large Taper Hub Flanges", Instn. Mech. Engrs., Symposium on Pressure Vessel Research, January, 1961.

(12.27) Malkmus, M. V., "Theoretical Stress Analysis of a Flat Face Bolted Flange Connection with an O-ring Gasket", M. S. Thesis, U. of Louisville, June, 1963.

(12.28) Levy, S., "Design Criteria for Zero-Leakage Connectors for Launch Vehicles, Vol. 4, "Design of Connectors", General Electric Co., Report No. 63GL44, March 15, 1963, Section 41.

(12.29) Schneider, R. W., "Flat Face Flanges with Metal-to-Metal Contact Beyond the Bolt Circle", ASME Paper No. 67-WA/PVP-2.

REFERENCES (contd.)


(12.30) Tanner, J. R., "Report to the Working Committee of Sub-Committee No. 3 of the Standardization of Pipe Flanges and Fittings", November, 1923.

(12.31) Waters, E. O. and Taylor, J. H., "The Strength of Pipe Flanges", Mechanical Engineering, Vol. 49, pp. 531 and 1340 (1927).

(12.32) Gough, H. J., "First Report of the Pipe Flanges Research Committee", Proc. Instn. Mech. Engrs., Vol. 132, p. 201 (1936).

(12.33) Tapsell, H. J., "Second Report of the Pipe Flanges Research Committee", Proc. Instn. Mech. Engrs., Vol. 141, p. 433 (1939).

(12.34) Johnson, A. E., "Pipe Flanges Research Committee - Third Report", Proc. Instn. Mech. Engrs., Vol. 168, p. 423 (1954).

(12.35) Jasper, Gregerson, and Zoellner, "Strength and Design of Covers and Flanges for Pressure Vessels and Piping", Heating, Piping, and Air Conditioning, Vol. 8, pp. 605, 672 (1936), Vol. 9, pp. 43, 109, 112, 174, 243, 246, 311 (1937).

(12.36) Petrie, E. C., "The Ring Joint, Its Relative Merit and Application", Heating, Piping, and Air Conditioning, Vol. 9, No. 4, (April, 1937).

(12.37) Rossheim, Gebhardt, and Oliver, "Tests of Heat Exchanger Flanges", Trans. ASME, Vol. 60, p. 305 (1938).

(12.38) Waters, E. O. and Williams, F.S.G., "Stress Conditions in Flanged Joints for Low Pressure Service", Trans. ASME, Vol. 74, p. 135, (1952).

(12.39) Barnard, R. E., "Design of Steel Ring Flanges for Water Works Service - A Progress Report", Journal of the AWWA, Vol. 42, p. 931 (1950).

(12.40) George, Rodabaugh, and Holt, "Performance of 6061-T6 Aluminum Flanged Pipe Assemblies Under Hydrostatic Pressure", ASME Paper No. 56-PET-19.

(12.41) Rodabaugh, E. C., "Bolt-Up and Pressure Tests of an 18" - 600 lb ASA Blind Flange to an 18" - 600 lb Welding Neck Flange", Report No. 2.027, Tube Turns, Louisville, Ky., October, 1964.

(12.42) O'Toole, W., "Tests on 3" - 150 lb ASA Welding Neck and Slip-On Flanges", Report No. 2.024, Tube Turns, Louisville, Ky, June, 1955.

(12.43) Rodabaugh, E. C., "Hydrostatic Tests of a 2" - 2500 lb Welding Neck Flanged Joint with Lens Ring Gasket", Report 2.008, Tube Turns, Louisville, Ky., September, 1957.

REFERENCES (contd.)

(12.44) Rodabaugh and Tate, "Effect of Flange Material Yield Strength on the Pressure Capacity of a Flanged Joint", Report 2.036, Tube Turns, Louisville, Ky., April, 1958.

(12.45) Rodabaugh, E. C., "Bolt-Load Capacity of 4" - 150 lb ASA B16.5 Flanges Made of 3003-F Aluminum", Report 2.049, Tube Turns, Louisville, Ky., May, 1958.

(12.46) O'Toole, W., "Pressure and Vacuum Tests of an 8" - 150 lb Lap Joint Flange Assembly", Report 97.005K, Tube Turns, Louisville, Ky., May, 1960.

(12.47) Rodabaugh, E. C., "Bending Tests on 4" - 300 lb ASA Welding Neck and Lap Joint Flange Joints", Report No. 8.010a, Tube Turns, Louisville, Ky., April, 1953.

(12.48) George and Rodabaugh, "Welding Neck Tapered Face Flanges for Use with Cast Iron Flanged Components", Pipe Line News Directory, 1959-60 Edition.

(12.49) Lenzen, K. H., "Strength and Clamping Force of Bolts", Product Engineering, December, 1947.

(12.50) <u>Piping Handbook</u>, 4th Edition (1945), p. 536, Mc-Graw-Hill Book Company, New York.

(12.51) <u>Fasteners Standards</u>, 4th Edition (1965), p. 343, Industrial Fasteners Institute, 1517 Terminal Tower, Cleveland, Ohio 44113.

(12.52) Thurston, R.C.A., "The Fatigue Strength of Threaded Connections", ASME Transactions, Vol. 73, p. 1085 (1951).

(12.53) Heywood, R. B., "Longer Fatigue Life for Bolts and Studs", Engineering, Vol. 189, p. 494 (1960).

(12.54) Eaton, I. D.,"Comparitive Fatigue Strength of Seven Screw-Thread Forms", Product Engineering, February 4, 1963.

(12.55) Viglione, J., "Nut Design Factors for Long Life", Machine Design, August 5, 1965.

(12.56) Mordfin, L., "Some Problems of Fatigue of Bolts and Bolted Joints in Aircraft Applications", NBS Technical Note, 136, PB 161637, 1962.

(12.57) Yeomans, H., "Programme Loading Fatigue Tests on a Bolted Joint", Technical Note No. Structures 327, Royal Aircraft Establishment, Ministry of Aviation London W.C. 2, 1963.

(12.58) Sproat, R. L., "A Checklist on Fastener Reliability", Missile Design and Development, June, 1960.

REFERENCES (contd.)

(12.59) Richter, G., "Factors Affecting the Failure of Screws and Bolts in Service", Engineer's Digest, March, 1965.

(12.60) Snow, A. L. and Langer, B. F., "Low-Cycle Fatigue of Large Diameter Bolts", ASME Paper No. 66-PET-8.

(12.61) Whalen, J. J., "Leakage and Elastic Characteristics of Compressed Asbestos Sheet Packing", ASME Paper No. 58-SA-28.

(12.62) Smoley, E. M., "Nonmetallic Gaskets", Chapter 12 of Seals Reference Issue, Machine Design, March 9, 1967.

(12.63) Dunkle, H. H., "Metallic Gaskets", Chapter 13 of Seals Reference Issue, Machine Design, March 9, 1967.

(12.64) Rodabaugh, Adam, Goobich, and Trainer, "Development of Mechanical Fittings, Phases I and II", Technical Documentary Report RTD-TDR-63-1115, Battelle Memorial Institute, December, 1963.

(12.65) Goobich, Adam, Baum, and Trainer, "Development of AFRPL Threaded Fittings", Technical Documentary Report AFRPL-TR-65-162, Battelle Memorial Institute, November, 1965.

(12.66) Fang, B. T., et. al., "Design Criteria for Zero-Leakage Connections for Launch Vehicles", Vols. 1-6, General Electric Company, Report No. 63GL46, March 15, 1963.

(12.67) Bain, J. A., et. al., "Zero Leakage Design for Duct and Tube Connectors for Deep Space Travel", Vols. 1-6, General Electric Co., Report S-67-1157.

(12.68) Weitzel, Robbins, Ludtke, and Ohori, "Elastomeric Seals and Materials at Cryogenic Temperatures", Technical Documentary Report No. ML-TDR-64-50, March, 1964.

(12.69) Rodabaugh, E. C., "Ratings of ASA B16.5 Flanges", Report to Subcommittee 4 of ASA B16., June 10, 1966.

(12.70) Waters, E. O., and Schneider, R. W., "Axisymmetric, Nonidentical, Flat Face Flanges with Metal-to-Metal Contact Beyond the Bolt Circle", Trans. ASME, J. of Eng'g. for Industry, Vol. 91, No. 3, pp 615-622.

(12.71) Rodabaugh, E. C., "Computer Programs for the Analysis of Flanged Joints", Preliminary Draft of Phase Report 115-7 from Battelle Memorial Institute (Columbus, Ohio) to Oak Ridge National Laboratory.

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# 13. OTHER MECHANICAL CONNECTIONS

Requirements of USAS B31.7 for threaded joints, expanded joints, flared, flareless and compression joints and for sleeve-coupled and other patented joints are shown in Table 13.1. These are given for Class I piping; the same requirements apply (by reference back to these paragraphs) to Class II and Class III piping.

## 13.1 Threaded Joints

USA standard taper pipe threads (USAS B2.1, such threads are usually identified as NPT) are widely used for small size piping. The seal is normally provided by the threads; B31.7 requires a seal weld. Alternately, straight threads can be used, with the pressure seal made either with a gasket or with a sealing surface formed on the end of the pipe. Rarely, and not to be recommended, taper pipe threads are used with an auxiliary gasket or seal at the end of the pipe.

Because threaded pipe is necessarily used with a female counterpart (coupling, threaded fitting), the static pressure capacity of such joints is seldom in question. However, fatigue failure of such joints due to cyclic bending is sometimes a problem. The failure may either consist of leakage or a fatigue crack through the pipe wall, normally starting at one of the exposed thread roots. The seal weld, to the extent that it is strong enough*, should solve the leakage problem, however the seal weld may not help the fatigue crack problem, particularly if the seal weld does not cover the exposed threads. The seal weld may reduce fatigue life if it is of poor quality (root cracks. etc.).

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* B31.7 does not give any dimensional requirements for a seal weld, and states (Par. 1-711.5) that "Seal welds shall not be considered as contributing any strength to the joint".

TABLE 13.1. REQUIREMENTS FOR JOINTS FROM USAS B31.7 (FEBRUARY, 1968)

1-713 EXPANDED JOINTS

Expanded joints shall not be used in Class I nuclear piping systems.

1-714 THREADED JOINTS

Screwed joints in which the threads provide the only seal may not be used in Class I nuclear piping systems. If a seal weld is employed as the sealing medium, the stress analysis of the joint must include the stresses in the weld resulting from the relative deflections of the mated parts.

1-715 FLARED, FLARELESS, AND COMPRESSION JOINTS

Flared, flareless, and compression-type tubing fittings may be used for tubing sizes not exceeding 1 in. OD within the limitations of applicable standards and specifications listed in Table 1-726.1 and requirements (b) and (c) below. In the absence of such standards or specifications, the designer shall determine that the type of fitting selected is adequate and safe for the design conditions in accordance with the following requirements.

(a) The pressure design shall meet the requirements of Subdivision 1-704.7.

(b) Fittings and their joints shall be suitable for the tubing with which they are to be used in accordance with the minimum wall thickness of the tubing and method of assembly recommended by the manufacturer.

(c) Fittings shall not be used in services that exceed the manufacturer's maximum pressure-temperature recommendations.

1-718 SLEEVE-COUPLED AND OTHER PATENTED JOINTS

Mechanical joints for which no standards exist and other patented joints may be used provided that adequate provision is made to prevent separation of the joints; they are accessible for maintenance,

TABLE 13.1 (Continued)

removal, and replacement after operation; and that a prototype joint has been subjected to performance tests to determine the safety of the joint under simulated service conditions. When vibration, fatigue, cyclic conditions, low temperature, thermal expansion, or hydraulic shock is anticipated, the applicable conditions shall be incorporated in the tests. The mechanical joints shall be sufficiently leak tight to satisfy the requirements of the design specification.

The only fatigue test data on threaded joints known to the writer are given by Markl and George[13.1]. These are tests on 4" Sch 40 pipe threaded into 4" - 300 lb USAS B16.5 threaded flanges. Failures of threaded joints consisted of:

(1) Persistent leakage along the threads. Even with only 25" head of water, this occured long before structural failure, and with 600 psi pressure and some of the higher bending moments, around 100 reversals were enough to start a dribble of water.

(2) Structural failure, consisting of a crack through the wall. Apparently all cracks started from the root of one of the exposed threads in the pipe.

The data are summarized in terms of stress intensification factors (i-factors) relative to the fatigue strength of a typical girth butt weld in straight pipe. The i-factors (depending upon how tight the joint was initially) range from 2.48 to 2.83 for leakage; 1.74 to 1.83 for structural failure. The i-factor for threaded joints found in USAS B31.1 is 2.3. For comparison, the i-factor for a girth fillet welded joint is 1.3. It should be recalled that the stress intensification factor of a typical girth butt weld in straight pipe, as compared to the fatigue strength of the pipe material tested as a polished coupon, has a stress intensification of about two. Accordingly, on a B31.7 or elastic basis, the stress intensification factor for threaded joints is about double those listed above, i.e., about 4.6.

A stress intensification factor for threaded joints is not yet included in B31.7, Appendix D, because of the following questions:

(1) To what extent are the test data for a 4" std wt threaded pipe joint applicable to other wall thicknesses and/or sizes? One notes that the

thread depth/wall thickness ratio is not constant; obviously not for different wall thicknesses of a given size and even for a constant pipe schedule, the ratio of thread depth to wall thickness varies as shown by the following tabulation:

| Nom Size | Thread Depth, h | Sch 40 Wall, t | $\frac{h}{t}$ |
|---|---|---|---|
| 4 | .0800 | .237 | .337 |
| 2 | .0696 | .154 | .453 |
| 1 | .0696 | .133 | .523 |

(2) It is not apparent that a seal weld will necessarily increase the structural fatigue strength of the threaded joint. It would be preferable, in the writer's opinion, to require a full fillet weld that also covers all exposed threads. Then the joint probably could be given the stress intensification factor presently assigned to girth fillet welds.

## 13.2 Expanded Joints

Presumably, expanded joints refers to a joint made by expanding (or rolling) the pipe into a flange or fitting similar to a boiler tube joint (without welding). Some limited data on pull-out strength of such joints exist. The strength of the joint depends upon the skill of the workman as well as the detail designs (type of grooves, if any). Present day use in pipelines of such joints appears to be limited to Sch 5 or Sch 10 stainless steel pipe for special locations when welding or brazing cannot be permitted because of potentially explosive environments.

## 13.3 Flared, Flareless and Compression Joints

These joints would normally find major applications in pipelines for connections to instruments. Such joints are widely used in the automotive, aircraft, and aerospace industries. As critical components in aircraft and aerospace vehicles, they have been subjected to extensive experimental investigation and must pass rigorous qualification tests.

Briefly, a flared fitting involves a seal made on the conically-flared end of the tubing itself. A separate flaring tool is used to make the flare on the tube end. A flareless fitting involves a "ferrule" which bites into the tubing when the joint is assembled. A compression fitting (in this general type of joint) involves a ball-sleeve which is idented into the tubing when the joint is made-up.

Some common commercial standards for these joints (and the associated fittings)are:

USAS B16.26 - Brass fittings for flared copper tubes

USAS B70.1 - Refrigeration flare-type fittings*

SAE J512d - Automotive tube fittings (flared or compression)

SAE J513c - Refrigeration tube fittings (flared)
(conforms in general to USAS B70.1)

SAE J514b - Hydraulic tube fittings (flared and flareless)

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* This is the only standard on tube fittings included in Table 1-726.1 of B31.7.

The standards listed above give dimensional and material requirements but no performance requirements. Military standards of the AN- or MS- series give comparable dimensional and material requirements. MIL-F-18280 gives performance requirements (proof pressure, burst pressure, vibration and cyclic bending fatigue) for flareless fittings. This standard is sometimes applied to performance requirements for flared fittings.

In addition to these "standard" fittings, there are a number of proprietary variants sold by various manufacturers.

During the past few years, considerable effort has been made to improve the reliability and performance characteristics of tube joints. One result is the so-called "MC" fitting, an improved flared fitting established by NASA, standard MC-146. Another development, aimed at tube connectors with very low helium gas leak rates, is the AFRPL threaded fitting[13.2].

Reference 13.3 through 13.20 is a partial bibliography of data on flared or flareless fittings; included herein principally to indicate the scope of information available on such joints. It might be remarked that problems are encountered either with (a) leakage of the seal or (b) fatigue failure of the attached tubing. The fittings themselves appear to be amply strong.

## 13.4 Sleeve-Coupled and Other Patented* Joints

Two widely used types of joints which presumably would fall in this classification are Dresser or Dayton couplings and Victaulic couplings. The Dresser coupling is made with plain end pipe, the seal being made by compressible gaskets at each end of the sleeve. The Victaulic coupling uses grooved-

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* The writer does not know if any of the joints discussed are actually patented.

end pipe with (usually) a two piece circumferential clamp and a circumferential cup-type gasket. Among other proprietary joints are "Greyloc" and "Marman Clamps". The writer does not have available any quantitative performance data on any of these or similar types of joints.

## 13.5 Unions

These types of joints, while quite extensively used in small size piping, seem to be orphans both in the B31.7 classifications and in the usual piping component standard organizations such as USAS, MSS, and API. Dimensional standards for unions are published by AAR-M-404 (Association of American Railroads) and Federal Specifications WW-U-516, WW-U-531, and WW-U-536. No specification for unions is listed in Tables 726.1 of B31.7.

Unions are commercially available as:

| | |
|---|---|
| Brass or Bronze | 125 lb |
| | 200 lb |
| | 300 lb |
| Malleable iron | 150 lb |
| | 250 lb |
| | 300 lb |
| Carbon steel | 300 lb, with bronze seats |
| | 2,000 lb |
| | 3,000 lb |
| | 6,000 lb |

Usually, unions are furnished with threaded ends. Carbon steel unions in the 2,000, 3,000, or 6,000 lb classes can be obtained with socket welding or butt-welding ends.

There are no quantitative data on performance characteristics of unions available to the writer.

## 13. REFERENCES


(13.1) Markl, A.R.C., and George, H. H., "Fatigue Tests on Flanged Assemblies", Trans. ASME, Vol. 72, p 77-87 (1950).

(13.2) Goobich, Adam, Baum & Trainer, "Development of AFRPL Threaded Fittings for Rocket Fluid Systems", AFRPL-TR-65-12 November 1965, Battelle (Columbus) Report to Air Force Rocket Propulsion Laboratory, Edwards Air Force Base, California.

(13.3) Allin, F. R., and Courtot, L. B., "Evaluation of Flareless Fittings for Low Density Gas Applications", Weatherhead Company, Test Report No. 69, 460-F (September 8, 1958).

(13.4) Beachley, N. H., "Survey of Hydraulic Fittings in Air Force Ballistic Missile Programs", Space Tech. Lab., Report 7431.2-289, 1-9 (August 25, 1960).

(13.5) Cornish, H. E., and Bloom, J. C., "Development of High Pressure Seals For AN Straight Thread Fittings", Applied Hydraulics, (18-24) (November, 1949).

(13.6) Courtot, L. B., "Refinement of Precision Flareless Fittings", Weatherhead Company, Engineering Progress Report (September, 1958).

(13.7) "Design Criteria for Zero-Leakage Connectors for Launch Vehicles", General Electric Co., General Engineering Lab, Schenectady, New York, Quarterly Progress Report No. 4, Contract NAS 8-4012, June 15, 1963.

(13.8) Hallesy, H. W., "Development of a Permanent and a Reconnectable Tube Fitting for High Pressures and/or High Temperature", Boeing Aircraft, Report D6-5327 (March, 1960).

(13.9) Lewis, S., "Leakage Problems with Conventional Fittings", Space Tech. Lab. Report 9733.5-460, 1-7 (May 24, 1961).

(13.10) Lewis, S., "MS & AN Fittings", Space Tech. Lab., Report GM60-7640.5-507, 1-4 (August 22, 1960).

(13.11) Mayhew, W. E., "Design and Development of a 1000 F Hydraulic System", Republic Aviation Report, AD 257 940 (June, 1960).

(13.12) Nicol, J., "Hot Gas Line Fittings 1800 F for Dynasoar Reaction Control Systems", Weatherhead Company (July, 1961).

(13.13) Phillips, R. W., "Flareless Fittings", Applied Hydraulics and Pneumatics, 6, 5, May, 1953, pp 84-86

(13.14) Richards, C. M., "Positive Gas Sealing With Flared Fittings", SAE Journal, 77-79 (October, 1960).

(13.15) Richards, C. M., "Precision Sleeves Improve Flareless Fittings", Hydraulic & Pneumatics, 120-122 (April, 1962).

(13.16) Seibel, L. L., and McGillen, V. W., "Hydraulic and Pneumatic Fitting and Tubing Test Program", North American Aviation Report, AD 235 024 (November, 1959).

(13.17) Davies, R. H., "The J.I.C.Performance Standards for Tubing and Tube Fittings", Applied Hydraulics, 9-27 (November, 1949).

(13.18) Dubrow, A., "Investigation of the Effect of Pre-Stress on Fatigue-Vibration Life of High Pressure Hydraulic System Tubing", Aeronautical Materials Laboratory, Philadelphia, Pennsylvania, AML NAM AE 6272, 1-13 (April 5, 1955).

(13.19) Lenhart, H. G., and Gartside, W., "Flared Tubing Fatigue Test", Boeing Aircraft, T 2-1432, 1-33 (November 11, 1957).

(13.20) Lundback, A. V., "Evaluation of Annealed Stainless Steel Tubing and 'AN' Fitting Joints", Aerojet-General SCR 56 (June 21, 1961).

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# 14. EXPANSION JOINTS

Compensation for the thermal expansion of a pipeline may be obtained by the inherent flexibility of the piping system itself; i.e., by loops, offsets, etc. Alternately, and sometimes preferably, the thermal expansion can be absorbed by means of expansion joints. There are two general classes of expansion joints:

1) Those using a convoluted or disc-like metal* member, and
2) Those using relatively moving parts, pressure-tightness being obtained by some type of packing or seal.

These classes are identified herein as "bellows expansion joints" and "slip, swivel and ball joints". An alternate, commonly used nomenclature identifies these classes as "packless" and "packed" expansion joints.

The following discussion is concerned primarily with bellows expansion joints for two reasons. First, bellows joints can be designed for "zero leakage", an important aspect in piping for radioactive fluids. Second, very little quantitative data are available on the performance characteristics of packed joints. Because of the numerous types and applications of bellows joints, the discussion begins with a general description of such joints and gives the nomenclature used later.

Insofar as the writer is aware, bellows expansion joints are not being used in the primary coolant loops of water-cooled reactors. Apparently it has been possible to compensate for thermal expansion by the flexibility of the pipe. The relatively high pressures involved in water-cooled reactors would pose a difficult design problem for bellows joints. They are, however,

* Bellows made of non-metallic materials are not included herein.

being used in penetrations of containment vessels, suppression chambers, dry wall-to-reactor seals and refueling seals. Bellows expansion joints have been used in gas-cooled reactors in England and are being used in the Ft. St. Vrain gas-cooled reactor in Colorado. Bellows expansion joints may find application in liquid metal-cooled reactors since the higher operating temperatures require greater thermal expansion capacity and lower pressures, which somewhat eases the bellows design problem.

## 14.1 Bellows Expansion Joints

### 14.11 Types of Bellows Expansion Joints

Bellows have been used in expansion joints for piping systems for many years. The Expansion Joint Manufacturers Association, which was founded in 1955, published the Association's Standards[14.1] in 1958. An enlarged third edition was published in October, 1969. The 62-page 1969 edition contains definitions of pertinent nomenclature, descriptions of the principal types of expansion joints and installations, and comments on installation techniques and performance characteristics. The Standards, adopted by the eight member companies, provide a good summary of the many significant aspects of pipeline expansion joint design. The following expansion joint descriptions were taken from the 1969 standards.

#### Single Expansion Joint

The simplest form of Expansion Joint consists of a single bellows that is designed to absorb all of the movement of the pipe section in which it is installed.

Double Expansion Joint

A double Expansion Joint consists of two bellows joined by a common connector which is anchored to some rigid part of the installation by means of an anchor base. The anchor base may be attached to the common connector either at installation or at time of manufacture. Each bellows acts as a single expansion joint independent of the other bellows and absorbs the movement of the pipe section in which it is installed. Double expansion joints should not be confused with universal expansion joints.

Internally Guided Expansion Joint

An internally guided expansion joint is designed to provide axial guiding within the expansion joint by incorporating a heavy telescoping internal guide sleeve, with or without the use of bearing rings. (Note: The use of an internally guided expansion joint does not eliminate the necessity of using adequate external pipe guides.)

Universal Expansion Joint

A universal expansion joint is one containing two bellows joined by a common connector for the purpose of absorbing any combination of the three basic movements, i.e., axial movement, lateral deflection and angular rotation. Universal expansion joints are usually furnished with limit rods to distribute the movement between the two bellows of the expansion joint and to stabilize the common connector. This definition does not imply that only a double bellows expansion joint can absorb universal movement.

Hinged Expansion Joint

A hinged expansion joint contains one bellows and is designed to permit angular rotation in one plane only by the use of a pair of pins through hinge plates attached to the expansion joint ends. The hinges and hinge pins must be designed to restrain the thrust of the expansion joint due to internal pressure and extraneous forces, where applicable. Hinged expansion joints should be used in sets of two or three to function properly.

Swing Expansion Joint

A swing expansion joint is designed to absorb lateral deflection and/or angular rotation in one plane. Pressure thrust and extraneous forces are restrained by the use of a pair of swing bars, each of which is pinned to the expansion joint ends.

Gimbal Expansion Joint

A gimbal expansion joint is designed to permit angular rotation in any plane by the use of two pairs of hinges affixed to a common floating gimbal ring. The gimbal ring, hinges and pins must be designed to restrain the thrust of the expansion joint due to internal pressure and extraneous forces, where applicable.

Pressure-Balanced Expansion Joint

A pressure-balanced expansion joint is designed to absorb axial movement and/or lateral deflection while restraining the pressure thrust by means of tie devices interconnecting the flow bellows with an opposed bellows also subjected to line pressure. This type of expansion joint is normally used where a change of direction occurs in a run of piping. The flow end of a pressure-balanced expansion joint sometimes contains two bellows separated by a common connector, in which case it is called a universal pressure-balanced expansion joint.

## 14.12 Expansion Joint Selection

The following material, which was also taken from the 1969 Standards of the Expansion Joint Manufacturers Association, summarizes the major steps in the selection of expansion joints.

"The first step in the selection of expansion joints is to choose tentative locations for the pipe anchors. By means of anchors, any piping system, regardless of its complexity, can be divided into a number of individual expanding pipe sections having relatively simple configurations (i.e., straight runs, "L" shaped bends, "Z" shaped bends, etc.). The number of pipe anchors selected, as well as their locations, will depend upon the piping configuration, the amount of expansion which can be accommodated by a single expansion joint, the availability of structural members suitable for use as anchors, the location of various pipe fittings, the location of connected equipment the location of branch connections, etc.

"In most applications, the major pieces of connected equipment such as turbines, pumps, compressors, heat exchangers, reactors, etc., can be considered as anchors. However, it is usually necessary to supplement these equipment anchor points by locating additional anchors at valves, at changes in the direction of the pipe, at blind ends of pipe, and at major branch connections. Unless there are obvious advantages to be gained from another approach, it is generally advisable to start out with the assumption that the use of single and double expansion joints in straight axial movement will provide the simplest and most economical layout. Wherever possible, the distance between anchors should be kept to a uniform dimension so that the expansion joints in the various pipe sections will be interchangeable. In order to minimize the number of expansion joints used, the distance between anchors should be selected so as to utilize expansion joints having a maximum number of corrugations in each bellows.

"After the anchor points have been tentatively located, the resulting pipe configurations should be reviewed to determine whether they conform to the standard pipe sections shown in Sections 2.4 and 2.5. At this point, consideration should be given to the relative merits of systems utilizing single and double expansion joints for axial movement only, as opposed to those utilizing universal, pressure-balanced, hinged and gimbal expansion joints. A final decision regarding anchor locations and the types of expansion joints to be used can only be made after comparison of various alternative solutions with respect to cost, the ability to comply with cyclic life and force requirements, space restrictions, etc.".

## 14.13 Bellows Convolutions

### 14.131 Formed Bellows

Formed bellows are usually made from longitudinally butt-welded tubing that has been fabricated from sheet metal with closely controlled thickness. They can be produced in many materials and sizes, and at a cost much lower than that for other types of bellows. Diameters up to 50 feet have been supplied. There seems to be no apparent upper limit on the size that can be made. In comparison with welded bellows (see below), formed bellows have a higher spring rate and require more ductile materials. However, because of the absence of circumferential welds, they are more reliable than welded bellows.

Although Table 14.1 shows only single-ply configurations, most formed bellows can be made with multiple plies. Three- and four-ply bellows are common. Multiple plies are used to provide a greater resistance to pressure and a lower spring rate than would be obtained with a single ply equal in thickness to the total thickness of the multiple plies. The major types of formed bellows are described briefly.

## Semitoroidal

Semitoroidal bellows are attractive for materials with relatively low ductility. The form also offers good pressure capability and stability. The convolutions may be truly semicircular, elliptical, or some combination of curves. A low deflection capability per convolution and a high spring rate are major limitations of this configuration.

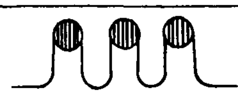
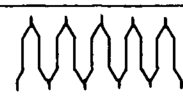
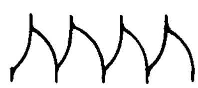
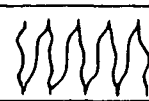
## U-Shaped

When flat sections are placed between the semitoroidal sections, a U-shaped, or flat-plate bellows configuration is formed. Over 50 percent of all the bellows are of this type. The shape is amenable to any of the methods for manufacturing formed bellows, a variety of performance characteristics can be achieved by varying the radii and depth of convolution, and supporting devices are easily installed externally or internally. When sized, the shape is more appropriately described as an "S-shape".

## Toroidal

Toroidal bellows have been developed to reduce the pressure-induced stresses in the bellows. By using a shape which is essentially circular, the effects of pressure are more evenly distributed along the convolution. In

TABLE 14.1. MAJOR TYPES OF BELLOWS CONVOLUTIONS

| | Convolution Shape |
|---|---|
| FORMED | |
| Semitoroidal |  |
| U-shaped |  |
| Sized U-shape (S-shape) |  |
| U-shaped, external ring support |  |
| U-shaped, internal ring support |  |
| U-shaped, external T-ring support |  |
| Toroidal |  |
| WELDED | |
| Flat |  |
| Stepped |  |
| Single sweep |  |
| Nested ripple |  |

addition, the stresses in the convolution are less affected by an increase in bellows diameter than is the case with the other convolution shapes. The Marquette Coppersmithing Company claims that their OMEGA shape distributes the stresses more evenly than a true toroidal bellows. Zallea Brothers advertise a HyPTor, or modified toroidal shape which is satisfactory for intermediate pressures and is more flexible than a true toroidal shape. Although the toroidal bellows permit high operating pressures, they are more difficult to manufacture than the other formed bellows and have a high spring rate.

### 14.132 Welded Bellows

The most commonly manufactured welded bellows are made up of shaped diaphragms which are alternately welded together at the inner and outer diameters. Although they are more expensive to manufacture than formed bellows, welded bellows offer three significant advantages over formed bellows: (1) a wider choice of materials, (2) more deflection per unit length, resulting in shorter assemblies or longer strokes, and (3) a wider choice of performance characteristics because of a greater variety of convolute dimensions and shapes. In general, welded bellows are available in sizes from 1/2 inch to 7 inches outside diameter. Bellows in excess of 12 inches in diameter have been produced. When forming limitations prevent the fabrication of formed bellows from tubing, bellows with convolutions similar to formed bellows are sometimes built up of welded sections.

Despite the impressive welding techniques that have been developed by the manufacturers of welded bellows, the large amount of welding required (approximately 18 inches per convolution in a 3-inch-OD bellows) makes fatigue failure less predictable for welded bellows than for other types of bellows.

### 14.133 Machined Bellows

Machined bellows are turned or ground from bar stock, tubing, or forged rings of most materials used in other types of metallic bellows, as well as of materials not found in sheet stock. High-strength, high-endurance, heat-treatable tool steels, in addition to high-strength, low-modulus titanium alloys can be used. The design of machined bellows is customized, with most machined bellows having high spring rates. Machined bellows have been made from 1/4 inch to 60 inches in diameter for pressures as high as 12,000 psi.

## 14.14 Manufacturing Considerations

Although manufacturing considerations have been omitted for most of the report chapters, these aspects are so important to the successful performance of bellows that a review of this subject has been included.

### 14.141 Formed Bellows

The formed-bellows manufacturing process begins with the fabrication of a thin metal cylinder from flat sheet or strip having a high-quality surface and containing no visible damage to the edges. After the sheet has been cut to size by a shearing operation, it is roll-formed to a cylindrical shape. Typically, the cylinder is somewhat overformed in order to assure that the edges will meet satisfactorily.

#### Longitudinal Seam Welding

The formed cylinder is placed in a welding fixture and a butt weld of the gas-tungsten-arc type (GTA; also known as TIG) is made along the mated edges of the sheet. The technology of making such welds is well advanced, and manufacturers are capable of making welds in material as thin as 0.003 inch.

Depending on the cylinder wall thickness and material, it may be necessary to add metal while making the weld. Metal addition is usually required for welds in sheets over about 0.10-inch thick to maintain a weld bead thicker than the base metal. If welding rod or wire of suitable composition is available for the material comprising the bellows, it can be fed into the arc as the weld is made. This procedure is known as cold-wire addition. Metal to form the weld bead can also be obtained by the melting during welding of a flange that has previously been bent up along the edge of the metal. Cold-wire addition gives better dimensional control of the resulting cylinder, but flange burndown is less likely to introduce contaminants into the weld. When conditions permit, a square-butt joint is made without any metal addition. Some bellows manufacturers are able to make seam welds in stainless steels without additions, but must make additions to welds in other alloys of the same thickness.

### Planishing

Many manufacturers cold work the weld zone with a pair of crowned opposed rolls in a planishing operation. Planishing must be carefully controlled in order that the wall thickness in the vicinity of the weld zone is not reduced below the base-metal wall thickness. Some manufacturers do not use planishing because of the danger of wall thinning, while others use it only for certain materials. Planishing may be desirable because tests[14.24] have shown that bellows with planished welds have higher cyclic life than similar bellows with welds left unplanished.

### Multi-Ply Bellows

In the fabrication of multi-ply bellows, a series of tubes, sized to fit closely one inside another, are cleaned and assembled ready for the forming operation. The cleaning at this stage is important since it is exceedingly difficult, if not impossible, to remove contaminating materials that have been trapped between the plies once the convolutions have been formed.

Forming

Almost every manufacturer uses a unique forming machine of proprietary design. Although these machines fall into several basic categories, there are differences in detail which may significantly affect the performance of the fabricated bellows. The basic categories of forming machines are as follows:

(1) Hydraulic, simultaneously formed convolutions

(2) Hydraulic, individually formed convolutions

(3) Hydrostatic, rubber pressure medium

(4) Mechanical rolls

(5) Mechanical expansion tools.

In the hydraulic process with simultaneous-convolution formation, the ends of the tube are first closed by movable platens. The end sections of the bellows are constrained in cylindrical dies that may be part of the platens. A series of split rings, one less than the number of convolutions desired, is carefully spaced along the length of the tube. Hydraulic pressure is then applied to the interior of the tube, causing the tube to bulge outward between the split rings.

From this point the processes of the various manufacturers differ. Some manufacturers leave the rings in place throughout the entire convolution-formation operation. Some manufactuers attach the rings to a pantograph during forming to maintain uniformity. Others remove the rings completely at this point and complete the convolution formation with the tube entirely free to restrictions except at the ends. During the formation of the convolutions, the platens must be moved together to accommodate the shortening of the tube. Some manufacturers accomplish the movement of the platens and the regulation of the hydraulic pressure by hand, while others have applied automatic controls to the process. Automatic controls are desirable from the standpoint of product uniformity.

It may be necessary to form the convolutions in several stages, depending upon the material and upon the depth of convolution desired relative to the tube diameter and wall thickness. Some manufacturers process-anneal their tubes following the initial bulging operations. Others find it necessary to stop several times during convolution formation, remove the split dies, clean, process-anneal, and reassemble the tube in the forming machine. Still other manufacturers restrict their product line to convolution depths that can be formed in their materials using a single operation, thus eliminating process-annealing. Manufacturers' processes also appear to differ widely in the amount of forming that can be accomplished between anneals.

Some manufactuers form each convolution individually, using essentially the same process as described above but with the hydraulic fluid confined to that region of the tube where the convolution is to be formed. The tube is first bulged. Then the external clamp holding the unformed portion of the tube is moved forward a preset distance to form a convolution. The operation is repeated after the tube is indexed to the next convolution position.

A variant of the hydraulic process is one in which the hydraulic oil is replaced by a rubber form. Under pressure, the rubber acts as a hydrostatic fluid. Its use eliminates the need for the presence of oil. Oil can cause carburization and possible embrittlement of the metal if it is not completely removed prior to process annealing or final heat treatment. Residues from oil have also been known to cause pit-type corrosion.

Perhaps the oldest method of forming bellows is that of shaping the convolutions by mechanical tools while rotating the tube (called roll forming). As in the hydraulic processes, there is considerable variety among the machines for roll forming. Some roll-form tooling resembles a lathe on which the tube to be formed is slipped over a centered rotating grooved die. An external tool is then used to press the tube into the grooves in the die, one groove at a time. Another type of rolling makes use of two small coaxial wheels over which the tube is placed. While these wheels are rotated, thus rotating the tube, a third wheel is brought down between the other wheels, thus forming a convolution. The tube is then indexed one pitch distance, and the operation is repeated. Considerable ingenuity by the manufacturers who use the roll-forming process has led to the ability to roll form the convolutions outward as well as inward. However, roll-formed bellows are currently in disfavor among some users because of the possibility of creating surface defects and smearing metal over these defects in such a way that they are hidden. A second objection to roll-formed bellows that is often cited is the excessive wall thinning at the roots or crowns of the convolutions that may be encountered if forming is not done carefully enough. Successful hydraulic forming of bellows, on the other hand, constitutes a proof test of sorts.

It may be necessary to set the pitch of the formed convolutions in a separate operation if the manufacturing method used results in unacceptable variations in pitch. This is done using shaped rolls similar to roll-forming tooling, but using them in such a way that they are merely run around the circumferences of successive convolutions without deepening them.

Sealing of Multi-Ply Bellows

Some multi-ply formed bellows, particularly those intended for steam piping service, are often left with the space between plies unsealed. Vent holes are even provided in the outer plies in some bellows. Unsealed construction is likely to be found in multi-ply bellows made from alloys that must be heat treated after forming. The reason for this is that air and moisture trapped between plies of a sealed bellows may create sufficient internal pressure between the plies at high temperature to cause gross deformation and ballooning of the bellows. Unsealed multi-ply bellows have the disadvantage that corrosive agents can get between the plies, where they may cause premature failure by stress corrosion.

The best practice for the manufacture of multi-ply bellows would seem to be to weld the clean, formed plies together around most of their circumference at each end of the bellows, heat treat, and then complete the seal welds as soon as possible. The heat treatment should never be used as a method of burning out oil or other contaminants on or between plies of bellows. Such residues can cause carburization and embrittlement of the metal and may cause local corrosion.

Multi-ply bellows intended for low-temperature service should be sealed with only dry gas or vacuum between the plies, since moisture will freeze out in service, affecting the spring rate. Electron-beam welding in vacuum is probably the best method of sealing multi-ply bellows.

## 14.142 Welded Bellows

The steps described below are for welded bellows made with formed diaphragms. The steps are similar for a bellows that is built up from welded sections to have convolutions similar to formed bellows.

### Blanking

The process begins with the blanking of doughnut-shaped disks, called diaphragms, from sheet material. The blanking operation must be carefully done, using dies that are in good adjustment to minimize the formation of burrs. Any burrs which are formed on the edges of the diaphragms must be removed to obtain good fitup for subsequent welding.

### Forming

Corrugations in diaphragms are introduced by spinning, stamping, or by hydrostatic pressure. The spinning is done on a lathe by pressing the metal against a corrugated form. This results in a certain amount of cold working which improves the life of the diaphragm. Some manufacturers stamp the diaphragm first and finish them by spinning. Spinning is subject to the same possible objections as roll forming of formed bellows.

In the stamping process, two mating steel dies are generally used. Some dies are made so that they make contact only with the material on concave sides of the corrugations. The depth can be adjusted through a wide range. The die can be made such that the corrugations are formed in succession from the inside to the outside, thus drawing the material gradually from the outside. In order to reduce friction, a lubricant may be used between the material and the polished die.

In the hydrostatic process, a metal blank is clamped against a corrugated die and hydraulic pressure or pressure from steel-backed rubber forces the blank against the die. Small bleed holes relieve the pressure between the blank and the die.

The material may be annealed prior to forming to make it more easily worked. After formation, the diaphragm may be heat treated to reduce the residual stresses created by the forming operation and, for some materials, to increase the strength. The type of heat treatment required before and after forming is a function of the material and of the diaphragm shape.

### Inner-Diameter Welding

A pair of diaphragms are placed in a welding jig with the inner diameters in contact and clamped with chill blocks on either side of the joint. An edge weld is then made around the inner circumference. This operation is usually accomplished with the gas-tungsten-arc process (GTA or TIG), but some manufacturers claim more uniform welds with the electron-beam process. The welded pair of diaphragms is referred to as a convolution. The welding operation is repeated for the number of convolutions desired in the bellows.

### Outer-Diameter Welding

The convolutions are stacked in another welding fixture with the outer diameters of adjacent convolutions in contact, split chill rings are used between the mated pairs of surfaces, and the outer diameters are welded in the same manner as the inner diameters.

Most welded-bellows manufacturers use a semiautomatic form of the GTA welding process in which the material to be welded is rotated beneath a stationary torch. Upon completion of a weld, the fixture is moved to the next weld position and the process is repeated. Some manufacturers now use electron-beam welding, at least for the outer-diameter welds. Small-scale plasma-arc welding equipment has recently become commercially available. Both of these latter processes are less sensitive to slight changes in power or arc length than the GTA process for welding thin metals.

Many of the welding difficulties that occur in welded bellows are related to the bellows materials, some of which are not as weldable as the alloys used for formed bellows. Heat-resistant alloys, most of which are vacuum melted, typically contain two or more phases and undergo various solid-solution and precipitation reactions during the thermal cycle associated with welding. In some alloys, these reactions may result in loss of ductility or strength in the weld heat-affected zone. These materials problems will not be entirely eliminated regardless of which welding process is used.

### 14.143 Heat Treating

Bellows for use in pipelines are usually made of austenitic stainless steel. Ordinarily the bellows are furnished "as-formed"; as such, the material is significantly cold worked both by membrane stretching as well as plastic bending. Fatigue tests of austenitic stainless steel bellows in air indicate that a heat treatment subsequent to forming reduces the fatigue life. Monel and Inconel bellows materials are sometimes used, particularly for fluids which are known to cause stress-corrosion cracking of austenitic stainless steel. Monel bellows are sometimes furnished in the "as-formed" condition, again because fatigue tests in air show better life for the as-formed bellows. A counter argument in favor of heat treatment after forming is based on the assumption that heat treatment will improve the stress-corrosion-cracking resistance. This point has not been clearly established; further typical pipeline bellows used at the manufacturer's rated displacements normally involve plastic strains so that the bellows material can become cold worked in service despite the heat treatment after forming.

### 14.144 End-Fitting Design

The design and attachment of end fittings to the bellows is critical from the standpoint of fatigue resistance. The problem is to attach a relatively thick-wall pipe segment to the ends of the relatively thin-wall bellows so that excessive stress concentrations are avoided. Manufacturers have various ways of making these attachments; usually such that fatigue failures (in tests or service) occur in the bellows and not at the attachment weld. However, as mentioned in several references giving bellows fatigue test data (see paragraph 14.173), fatigue tests do sometimes result in failure at the attachment to the end fitting. Also, both fatigue tests and service data indicate a tendency for fatigue failures to occur in the end convolutions; possibly due to the effect of the cylinder-to-torus transition.

The end fittings themselves are normally either a pipe segment for butt-welding into a pipeline, or flanged ends for flanging into the line. These parts are designed by the usual methods for pipe or flanged joints.

## 14.15 Theory

### 14.151 Elastic Stresses

Ideally, bellows expansion joints are thin shells of revolution; hence, the relatively well-advanced theory of axisymmetric structures is applicable. The earliest known application of shell theory to bellows is given by Salzmann[14.2] in 1946. He investigated the "U-shaped" convolution, using an energy method to obtain force-deformation relations for bellows subjected to axial displacement; internal pressure loading was not included. Clark[14.3] obtained asymptotic solutions for the semitoroidal convolution shape with either axial loading or internal pressure. Dahl[14.4] also obtained solutions for the semitoroidal convolution using energy methods. Turner[14.5] gives, in outline form, an analysis method applicable to U-shaped convolutions. He includes both the asymptotic solutions analogous to those developed by Clark[14.3] and the series solutions analogous to those developed by Dahl[14.4]. Laupa and Weil[14.6] give solutions for the U-shaped convolution with axial loads or internal pressure using energy methods for the toroidal sections and plate theory for the annular plate connecting the root torus to the crown torus.

The M. W. Kellogg Company, according to McKeon[14.7], has prepared a computer program based on the analysis of Laupa and Weil. Anderson[14.8], at Atomics International, has prepared a computer program based on the work of Clark[14.3]. It might be noted that the asymptotic solutions given by Clark are limited to certain values of the parameter $\mu = \sqrt{12(1-\nu^2)b^2/ah}$, where $\nu$ = Poisson's ratio, b = torus cross-section radius, a = distance from axis of revolution to torus center, and h = wall thickness. Roughly, $\mu$ must be larger than 6 for the asymptotic solutions to be valid. Some discrepancies between stresses calculated by the Kellogg program and those by the Atomics International program may be due to this aspect.

The development of general-purpose shell of revolution programs such as MOLSA[14.9] provides an analysis tool which includes all of the above developments plus:

(1) The capacity to analyze for offset or rotational displacement of one end of the bellows with respect to the other end.

(2) The capacity to analyze an arbitrarily-shaped convolution and wall-thickness variation. This aspect is significant because bellows convolutions which are nominally "U-shaped" or nominally "toroidal-shaped" usually are not actually so shaped. The deviation from the

assumed shape can produce large stresses, particularly for the nominally toroidal-shape with internal pressure loading.

The above discussion is concerned with linear elastic theory. In most pipeline bellows, nonlinear effects are significant. At least one nonlinear elastic shell of revolution program, called NONLIN,[14.10] exists for analysis of this aspect. Trainer, et al.[14.11], used both MOLSA[14.9] and NONLIN[14.10] computer programs in the analysis of both "formed" and "welded" bellows. These or other similar computer programs provide tools to include in the analysis the rather complex corrugation shapes and thickness variations as determined by inspection of actual bellows.

The following general comments on stresses in bellows are based on work done in Reference (14.11):

Discussion of Stresses in Formed Bellows

The formed bellows designed for a given application is usually a compromise between a deep U-shaped bellows with larger deflection capability and a shallow convolution semitoroidal-type bellows with more pressure capability. Within the constraints imposed by spring-rate requirements and minimum buckling loads, the selected bellows should have the lowest maximum stresses under the most severe combinations of operating pressure and deflection.

As shown in Appendix D of Reference (14.11), the deflection and pressure stress patterns vary greatly, depending on the general bellows configuration. Because the pressure and deflection stresses are combined algebraically, the parametric curves given in Appendix D of Reference (14.11) can be used to estimate the best approximate configuration for each application.

To determine the pressure and deflection stresses accurately in the final bellows configuration, however, it is necessary to calculate the stresses for the exact bellows dimensions.

Discussion of Stresses in Welded Bellows

Welded bellows are used in applications involving moderate pressures and large axial movement at low spring rates. In contrast to formed bellows, the maximum pressure and deflection stresses in welded bellows of standard design always occur near the root and crown welds. This is undesirable since it means that the maximum stresses occur in a notched heat-affected zone. The change in section resulting from the weld bead also represents a possible source of stress concentration.

One of the most significant results of the Air Force program covered in Reference (14.11) was the discovery that it is possible to redesign nested-ripple welded bellows so that the stresses near the crown and root welds are virtually eliminated. This design change involves tilting the bellows flats with respect to the axis of the bellows. By reducing the stresses near the welds, so that the maximum stresses occur away from the weld areas and in an area where the metal has the properties of the original sheet material, the fatigue life of welded bellows should be significantly improved. It is believed that this slight design change alone would result in a major improvement in the operating characteristics of welded bellows if optimum tilted flat configurations can be found for most types of welded bellows convolution shapes.

### 14.152 Elastic-Plastic Analysis

Typical pipeline bellows, when used at the full rated pressure and displacements given by manufacturers, are subject to strains well up into the plastic range. To the extent that only a few hundred cycles will be imposed during the desired lifetime, such strains are quite acceptable and give an economical means of designing for thermal expansion. As discussed later in Paragraph 14.173, some of the elastic stress calculation equations described in Paragraph 14.151 have been used to correlate and possibly extrapolate fatigue tests on bellows.

The question arises as to how reliable are elastic-stress calculations (without benefit of adjustments based on fatigue and data) when used to predict low-cycle fatigue of bellows. For a cyclic life of the order of 1000 cycles, the calculated stresses may be far above the 3 $S_m$(or 2 $S_y$) limit for secondary stresses used in the ASME Nuclear Vessels Code or the USAS Nuclear Piping Code. This aspect would seem to cast some doubt on the direct applicability of elastic-stress calculation. However, the significant strain-hardening capacity of austenitic stainless steels (at least at low to moderate temperatures) may be sufficient to insure "shakedown" to essentially elastic behavior after a few cycles; in which case the elastic-stress calculations may directly indicate the fatigue life. The applicability of an elastic analysis may also be questioned when the temperature is sufficiently high so that creep occurs.

Computer programs which include elastic-plastic analysis capability may provide guidance in answering the above questions. The programs FEELAP[14.12] (Finite Element Elastic-Plastic) and NONLEP[14.13] (General Shell of Revolution, Nonlinear, Elastic Plastic) are examples of programs which may be applicable. These programs can be extended, without major change, to cover creep.

### 14.153 Elastic/Plastic Buckling or Squirm

Multiconvolution bellows with internal pressure loading are subject to a type of instability known as "squirm". In part, the behavior is analogous to a beam-column with a compressive axial load. While this design limitation was probably known to bellows manufacturers for many years, Haringx(14.14), in 1952, was apparently the first to publish a theoretical explanation and equations for calculating pressure limits to avoid this instability. The following comments on the problem are based on the work of Reference (14.11):

The Euler critical load for a perfectly straight bellows may be calculated from the formula:

$$P_{cr} = \frac{4\pi^2 D}{L_c^2},$$

where D = the lateral bending stiffness, lb-in.$^2$

$L_c$ = total live convolution length, in.

For a bellows under internal pressure and axial compression, the equivalent axial load $P_{cr}$ is a combination of a pressure force and a compression force as given by Equation (J-16) in Appendix J of Reference (14.11). If the bellows were perfectly made, these would be the conditions that would cause gross buckling, or squirm, of the bellows.

Although the critical buckling pressure calculated for one experimental formed bellows was more than 330 psi, the bellows specimens tested were found to exhibit detectable sidewise movement at pressures of less than 80 psi. The reason for this was that instead of being perfectly straight, the bellows were actually bowed slightly, so that they had the appearance of a slightly bent beam. Because of this imperfection, internal pressure in the bellows induced a bending moment that tended to increase the bow, and the bellows began to deform sideways from the onset of the pressure loading.

The sidewise movement of a bellows introduces additional strains and stresses in the convolutions of the bellows. Thus, the total stress is:

$$\sigma_T = \sigma_p + \sigma_\Delta + \sigma_M,$$

where $\sigma_p$ and $\sigma_\Delta$ are the usual axisymmetric stresses from internal pressure p (psi) and compressive deformatiom $\Delta$ (in.), and $\sigma_M$ is the additional asymmetric stress from sidewise bending of the bellows due to beam-column buckling. As shown in Appendix J of Reference (14.11), the stress $\sigma_M$ can be determined from computer calculations using the mathematical model of the bellows if measurements are made of the bellows imperfections.

At the inner surface of an inner convolution of a 5-inch bellows, the meridional stresses were calculated to be:

$$\sigma_p = 40{,}000 \text{ psi}$$

and

$$\sigma_M = 11{,}500 \text{ psi}$$

for p = 78.6 psi and $\Delta$ = 0.0 in. The stress value 11,500 psi corresponded to a modest sidewise deflection of 0.004 in. Thus, elastic beam-column buckling of a bellows may result in an appreciable increase in stress in the bellows. If this stress fluctuates with the other fluctuating stresses, the fatigue life of the bellows may be significantly reduced.

If the elastic beam-column buckling loads are exceeded, the highly stressed parts of the bellows convolutions will deform plastically and a state of permanent squirm deformation will result. This mode of failure is reasonably well known and squirm-producing combinations of pressure and reasonably well known and data on squirm-producing combinations of pressure and deflection can be obtained from some manufacturers.

A second type of instability involves the "in-plane buckling" of individual corrugations. This is analogous to the instability of a circular shell which buckles under the action of external pressure in four half-waves. Column (or lateral) squirm is the most common in pipeline size expansion joints, whereas "in-plane buckling" generally occurs when the convoluted length is less than the bellows diameter. Some references have implied that a bellows which is "square" (length equal to or less than the diameter) will not squirm. On the contrary, some "square" bellows may squirm at less than their maximum compression rating when the internal pressure is equal to the maximum rated operating pressure with the squirm being of the "in-plane buckling" type.

It should be noted that "squirm" can develop almost instantaneously into a complete and catastrophic deformation of the bellows. A typical picture of a deformed shape is shown in Figure 14.1.

## 14.154 Limit Loads

As in most piping components, some indication is desirable of those loads which lead to gross deformation of bellows joints. In many piping components, the "burst pressure" is a significant limit load because the component is serviceable up to the pressure that causes rupture. The burst pressure of bellows is usually not significant because the convolutions are normally quite grossly deformed before rupture occurs. Limit loads (axial, rotational, and offset loads) are usually not of interest in pipeline bellows because deflections are applied rather than loads. What is of interest is the pressure which will cause gross deformations (assuming squirm does not occur). This aspect is discussed briefly in the following.

The elastic solution for stresses in shells has been employed by Marcal and Turner[14.15] to obtain a lower bound on the axisymmetric collapse


FIGURE 14.1. EXAMPLE OF SQUIRM IN BELLOWS,
INTERNAL PRESSURE LOADING, ENDS FIXED

pressure for bellows. As a first approximation, this method was also tried in the Air Force program(14.11). The method consists of scaling up the maximum elastic stress state at a point in a shell to the plastic collapse value. In a 5-inch bellows the maximum stress occurred at the roots of the convolutions and was predominantly a bending state of stress. Scaling up the corresponding bending moment to the plastic collapse value gave a plastic collapse pressure of 116 psi.

Tests with these bellows showed that collapse occurred at internal pressures of about 260 to 270 psi. Even if allowance was made for strain hardening due to forming and fatigue-test cycling at the root, it did not appear that this accounted for the larger observed collapse pressure--particularly since the root area was also observed to remain relatively rigid at collapse. Thus, use of the elastic solution to predict lower bounds based upon maximum elastic stress did not provide sufficient accuracy.

Marcal and Turner had much better success. This was believed to be due to two different kinds of plastic collapse which are related to two different ranges of diameter-to-thickness ratios. The diameter-to-thickness ratio for the 5-inch bellows was $d/h = 5.0/0.010 = 500$, whereas the ratio for the bellows tested by Marcal and Turner ranged from 8.2 to 23.6. It was reasoned that a membrane stress state predominates at plastic collapse of the thin-walled bellows ($d/h = 500$), and that a bending-stress state predominates at plastic collapse of thick-walled bellows ($d/h \simeq 10$).

If the above reasoning were correct, then the maximum membrane stress calculated elastically was expected to result in a better prediction of the collapse pressure. The following method was tried. The membrane stress resultants from the elastic computer solution were taken at the inflection point where the bending moment was 0, and were scaled up to the collapse value. The resulting calculation of 313 psi was quite close to the experimental values. It was believed that this was as close an approximation as could be made without conducting a complete detailed theoretical-plastic analysis, which was beyond the scope of the program.

Theoretical predictions of collapse pressures were then made for other bellows. The pressures causing axisymmetric plastic collapse were found to be significantly higher than the elastic buckling (squirm) pressures. Thus, squirm may be the controlling mode of failure in many bellows.

An interesting use of restraint has been reported by Newland[14.16], who analyzed the buckling resistance of a universal expansion joint. He has shown that, by providing a correctly designed supporting structure, the critical buckling pressure can be increased up to four times the value for the same system without supports.

Some manufacturers list burst pressures for bellows. This may be the pressure at which axisymmetric collapse is expected. Since the material usually does not rupture at the initial stage of collapse, this value represents a safety factor for burst. A burst-pressure value can also represent a calculation based on the ultimate tensile strength of the bellows wall. As such, it may have little practical meaning since rupture may take place at a lower pressure in a location where the bellows was creased during deformation.

### 14.155 Vibration

The life of a bellows may be drastically reduced if resonance causes amplitudes greater than those estimated for the normal operating conditions. Resonance can occur in response to vibration of the supporting structure, or to the movement of fluid through the bellows.

Structurally Induced Vibration

The general approach to a structurally induced vibration problem is to use a bellows which will not resonate with the structure, or to apply various dampening devices to the bellows. Formulas can be used to estimate resonant frequencies in a bellows. Except in unusual circumstances, vibration of pipeline expansion joints does not appear to have been a problem.

The problem of structurally induced vibration in bellows was investigated in some depth by Daniels[14.17] as a part of a program for the design of expulsion* bellows. Daniels was able to predict the accordion and beam vibration modes using formulas for a solid bar and beam when the constants used in the formulas were interpreted correctly. Calculations of the natural frequency for bellows clamped at both ends, undamped, and vented to atmosphere can be made by substituting the appropriate values in the frequency equations shown below:

Accordion Mode

$$f_n = 1/2 \sqrt{\frac{kg}{W_m}}$$

Beam Mode

$$f_L = \frac{A_n}{2\pi} \sqrt{\frac{kR_o^2 g}{2L_c^2 (W_m)}}$$

where $f_n$ = fundamental natural frequency for the accordion mode, cps

$f_L$ = fundamental natural frequency for the lateral beam mode, cps, when the constant $A_n$ = 22

k = axial spring rate of bellows, lb/in.

g = acceleration due to gravity, 386 in./sec$^2$

$W_m$ = weight of metal in the convolutions, lb

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* Expulsion bellows are used in zero-gravity environments for obtaining a positive displacement of fluid from a tank by decrease of its volume.

$R_o$ = outside diameter of the convolutions $\div$ 2, in.

$L_c$ = live length of bellows.

The applicability of Daniel's accordion and beam-mode formulas for small bellows was evaluated during the Air Force program[14.11], through theoretical and experimental vibration analyses of a number of test bellows. Each analysis consisted of: (1) determining the weight and spring rate of the bellows, (2) calculating the natural frequency for the accordion and beam modes of vibration, and (3) subjecting the bellows to axial and transverse vibrations on a Caladyne shaker table.

For the formed bellows, the experimental results for the accordion mode correlated very closely with the formula predictions. The experimental results for the lateral beam mode did not correlate well with the theoretical predictions, and it was concluded that the beam formula is not applicable to the type of bellows tested. This was attributed to the effects of shear deformation and rotary inertia. These effects are known to result in lower frequencies than predicted by the classical theory and cause a greater reduction for shorter length beams.

The results for the welded bellows were essentially the same as for the formed bellows. The calculated and observed values for the accordion mode were quite close, while the calculated and observed values for the beam mode disagree even more than for the formed bellows.

All the bellows tested exhibited low internal damping and extremely narrow resonant periods. Except for nonstandard modes of vibration caused by noncentroidal excitation, bellows response to inputs other than true harmonics was practically negligible. Light applications of Coulomb damping eliminated bellows vibration altogether.

In addition to the analysis method discussed previously, it should be noted that general shell of revolution programs, such as SHOREF[14.18], that permit calculation of natural frequency are available. The analysis of dynamic response of bellows can be further extended for a known end displacement forcing function.

Flow-Induced Vibration

Unfortunately, little theoretical work has been done on flow-induced vibration in bellows. Such vibration can often be prevented by a liner in the bellows which separates the convolutions from the flow stream. However, flow-induced vibration causes bellows failures in piping systems, and this failure mode should be investigated more extensively.

## 14.16 Corrosion

Because of the combination of high stresses and thin-wall material, problems with stress-corrosion cracking or corrosion-accelerated fatigue are highly significant in bellows. Austenitic stainless steels are ordinarily considered quite resistant to corrosion by such fluids as steam or condensate. When such steels are used for bellows, however, only a few parts per million of chloride ion may lead to failure of the bellows in a short time. Changing to Monel material will not necessarily solve the problem. It might be remarked that for the majority of bellows installed in steam or condensate lines, problems with corrosion do not arise. In other seemingly comparable installations, many failures occur. Insofar as the writer is aware, the conditions leading to failure have not been isolated. Bellows expansion joints have potential applications in liquid-metal-cooled reactor piping. The corrosion problem, when the fluid is a liquid-metal, may be a major uncertainty in assessing the reliability of the bellows for this application.

## 14.17 Test Data

### 14.171 Measured Strains

Feely and Goryl (14.19) give data on "welded" bellows obtained by SR-4 strain gages. Loadings consisted of axial displacements, rotational displacement, and internal pressure. Turner and Ford (14.20) give test results on "formed" bellows of various convolution shapes*. Loading consisted of axial compression. Turner (14.5) gives additional data on formed bellows, with loadings consisting of axial compression, offset compression (producing a rotational displacement) and internal pressure. Bowden and Drumm (14.21) give data on a U-shaped formed bellows with 60-inch inside diameter, 4.5-inch convolution height, and wall thickness of about 0.25 inch. Loadings consisted of rotational displacement and internal pressure. Because of the large size, and apparently clearly established dimensional data, this set of tests should

Winborne (14.22) gives data on ten 20-inch inside-diameter formed bellows of various convolution shapes, manufacturing methods, and reinforcing details. Loadings consisted of either axial displacement or internal pressure. The following comparison of test data with theory is quoted from Reference (14.22). "Calculations were made on the basis of specified dimensions and compared to the maximum measured stress on an inner convolution induced by axial deflection, and to the direct measurement of spring constants. The results are shown below:

*The test bellows were actually fabricated with a circumferential weld at the convolution crests.

| Bellows No. | Convolution Type | Maximum Stress, ksi* Calculated | Measured |
|---|---|---|---|
| 6 | U-shaped | 147 | 174 |
| 7 | U-shaped, inner rib | 91 | 66 |
| 8 | U-shaped | 39 | 34 |
| 9 | U-shaped | 65 | 60 |
| 11 | Toroidal shaped | 100 | 141 |

"The accuracy of calculated stress was influenced by fabrication details, such as the type of bellows-to-pipe end connection, the degree of wall metal thinning caused by the drawing process, and the deviation from specified dimensions.

| Bellows No. | Axial Spring Constant, lb/in. Calculated | Measured |
|---|---|---|
| 6 | 5320 | 6260 |
| 8 | 510 | 475 |
| 9 | 2340 | 1640 |
| 11 | 6630 | 6390 |

"Equations for calculation of stress and spring constants of bellows are contained in Appendix B of this report and References** (1) and (2) (Anderson's Equations)."

There are several points to note about the comparisons:

(1) No comparisons are given for internal pressure loading although test data were obtained and the theory cited covers internal

*This is presumably the stress for an axial displacement of 1 inch for the total bellows. For example, bellows No. 6 had five convolutions; the axial displacement would be 0.2 inch per convolution for the stresses shown.

**References (14.8) and (14.23) herein.

pressure. Possibly this is because the test data for internal pressure loading indicates a significant nonlinear effect.

(2) Comparisons are made with stresses on "an inner convolution". Higher stresses were measured at end convolutions.

(3) It is not apparent why comparisons were made for bellows Nos. 6, 7, 8, 9, and 11, but not for bellows Nos. 2, 3, 5, and 10 for which test data are also given.

Trainer, et al.[14.11], give data on both formed and welded bellows. These were relatively small bellows, from 1 to 5-inch nominal inside diameter. The formed bellows were all nominally U-shaped without reinforcing. Loadings consisted of axial displacement and internal pressure. Experimental determination of strains in the 5 and 3-inch bellows presented no great difficulties although the wall thicknesses were such (0.010 inch for 5 inch and 0.008 inch for 3 inch) that a correction to the measured bending strain for the distance from the metal surface to the strain gage foil surface was significant. For the 1-inch formed bellows and the welded bellows, however, the strain gradients were such that, even with the 0.016-inch gage length strain gages used, the experimental results may be questionable.

Comparison of test data with theory is shown by Table 14.2, taken from Reference (14.11). Results for the 5-inch size show remarkably good agreement between test data and theory. For the 3-inch size, agreement is reasonably good. The 1-inch size shows some major disagreements. Comparisons of spring rates are shown in Table 14.3. The data in Reference (14.11) appear to be the only set of data in which a careful effort was made to incorporate in the stress calculations the shape and thicknesses of the actual bellows subjected to strain gage tests. The theoretical calculations were made using the NONLIN[14.10] computer program.

TABLE 14.2 COMPARISON OF THEORETICAL AND EXPERIMENTAL STRESSES FOR TYPICAL FORMED BELLOWS (14.11)

| | | | Theoretical and Experimental Stresses, psi(a) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 5-inch, One-Ply SS Bellows | | 3-inch, One-Ply SS Bellows | | 1-inch, One-Ply SS Bellows | | 3-inch, One-Ply Inconel Bellows | | 1-inch, One-Ply Inconel Bellows | |
| | | | Stresses | % Diff. | Stresses | % Diff. | Stresses | % Diff. | Stresses | % Diff. | Stresses | % Diff. |
| Deflection Stresses (d) | Convolution Crown | Meridional: | | | | | | | | | | |
| | | Theoretical | +19,294 | +0.2 | +16,987 | -14.1 | +35,048 | -8.5 | +22,879 | -0.3 | +58,568 | -25.3 |
| | | Experimental | +19,330 | | +14,580 | | +32,075 | | +22,177 | | +43,741 | |
| | | Circumferential: | | | | | | | | | | |
| | | Theoretical | +10,791 | +12.6 | +9,887 | -1.1 | +19,803 | +14.9 | +9,742 | +10.4 | +21,684 | -15.2 |
| | | Experimental | +12,150 | | +9,780 | | +22,750 | | +10,867 | | +18,392 | |
| | Convolution Root | Meridional: | | | | | | | | | | |
| | | Theoretical | +22,943 | +6.7 | +20,623 | -8.1 | +48,931 | -14.8 | +25,592 | -0.3 | (c) | (c) |
| | | Experimental | +24,482 | | +18,963 | | +41,695 | | +24,724 | | (c) | |
| | | Circumferential: | | | | | | | | | | |
| | | Theoretical | +2,727 | (b) | +2,935 | (b) | +9,097 | (b) | +7,445 | (b) | (c) | (c) |
| | | Experimental | +2,911 | | +2,697 | | +7,750 | | +7,187 | | (c) | |
| Pressure Stresses (d) | Convolution Crown | Meridional: | | | | | | | | | | |
| | | Theoretical | -57,112 | -4.3 | -50,446 | -13.8 | -35,865 | -28.1 | -57,348 | -10.8 | -23,790 | -17.4 |
| | | Experimental | -54,655 | | -43,500 | | -25,775 | | -51,133 | | -19,660 | |
| | | Circumferential: | | | | | | | | | | |
| | | Theoretical | -16,242 | -9.9 | -20,098 | -1.5 | -18,570 | -54.0 | -16,490 | +5.8 | -6,845 | -6.1 |
| | | Experimental | -17,850 | | -19,800 | | -8,550 | | -17,476 | | -6,425 | |
| | Convolution Root | Meridional: | | | | | | | | | | |
| | | Theoretical | +66,024 | -13.4 | +58,709 | +0.6 | +44,235 | -29.6 | +45,599 | -32.5 | (c) | (c) |
| | | Experimental | +57,145 | | +59,064 | | +31,150 | | +30,761 | | (c) | |
| | | Circumferential: | | | | | | | | | | |
| | | Theoretical | +26,268 | (b) | +19,559 | (b) | +13,620 | (b) | +13,470 | (b) | (c) | (c) |
| | | Experimental | +22,767 | | +19,688 | | +9,585 | | +10,500 | | (c) | |

(a) Plus values indicate tensile stresses; minus values indicate compressive stresses.
(b) These values are similar to the meridional values because of the method of calculating the experimental circumferential stresses at the convolution root--no circumferential strain gages were used at this location.
(c) Strain gages could not be placed on the convolution root of this bellows.
(d) Stresses are for comparative purposes only. The deflection or pressure at which comparisons are made varies for the different bellows listed.

TABLE 14.3. COMPARISON OF THEORETICAL AND EXPERIMENTAL SPRING RATES FOR TYPICAL FORMED BELLOWS[14.11]

| 5-inch, One-Ply Stainless Steel, 12 Convolutions | | | | 3-inch, One-Ply Stainless Steel, 10 Convolutions | | | | 1-inch, One-Ply Stainless Steel, 8 Convolutions | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Bellows | Compr. Spring Rate, lb/in. | Extens. Spring Rate, lb/in. | Comb. Spring Rate, lb/in. | Bellows | Compr. Spring Rate, lb/in. | Extens. Spring Rate, lb/in. | Comb. Spring Rate, lb/in. | Bellows | Compr. Spring Rate, lb/in. | Extens. Spring Rate, lb/in. | Comb. Spring Rate, lb/in. |
| JD87 | 294 | 332 | 313 | JD61 | 138 | 152 | 145 | JD23 | 73 | 80 | 77 |
| JD88 | 302 | 316 | 309 | JD62 | 155 | 179 | 167 | JD24 | -- | -- | -- |
| JD89 | 287 | 316 | 302 | JD63 | 152 | 193 | 173 | JD25 | 76 | 93 | 85 |
| JD90 | 296 | 324 | 310 | JD64 | 148 | 174 | 161 | JD26 | 78 | 89 | 84 |
| JD91 | 288 | 324 | 306 | JD65 | 163 | 184 | 174 | JD27 | 78 | 85 | 82 |
| JD93 | 272 | 302 | 287 | JD66 | 168 | 187 | 178 | JD28 | 70 | 85 | 78 |
| JD94 | 279 | 314 | 297 | JD67 | 155 | 182 | 169 | JD30 | 76 | 85 | 81 |
| JD95 | 280 | 321 | 301 | JD69 | 151 | 166 | 159 | JD31 | 80 | 85 | 83 |
| JD96 | 287 | 322 | 305 | JD70 | 156 | 187 | 172 | JD32 | 68 | 82 | 75 |
| JD97 | 283 | 317 | 300 | JD71 | 172 | 198 | 185 | JD33 | 82 | 89 | 86 |
| JD98 | 284 | 316 | 300 | JD72 | 174 | 194 | 184 | JD34 | 74 | 78 | 76 |
| Exp. Avg. | 287 | 319 | 303 | | 158 | 181 | 170 | | 77 | 85 | 81 |
| Theoretical | -- | -- | 325 | | -- | -- | 161 | | -- | -- | 86 |

| 3-inch, One-Ply Inconel, 14 Convolutions | | | | 1-inch, One-Ply Inconel, 16 Convolutions | | | |
|---|---|---|---|---|---|---|---|
| Bellows | Compr. Spring Rate, lb/in. | Extens. Spring Rate, lb/in. | Comb. Spring Rate, lb/in. | Bellows | Compr. Spring Rate, lb/in. | Extens. Spring Rate, lb/in. | Comb. Spring Rate, lb/in. |
| JD119 | 139 | 145 | 142 | JD107 | 76 | 76 | 76 |
| JD120 | 125 | 131 | 128 | JD108 | 82 | 82 | 82 |
| JD121 | 137 | 143 | 140 | JD109 | 78 | 78 | 78 |
| JD122 | -- | -- | -- | JD110 | -- | -- | -- |
| JD123 | 139 | 145 | 142 | JD111 | 90 | 90 | 90 |
| JD125 | 131 | 133 | 132 | JD112 | 79 | 75 | 77 |
| JD126 | 136 | 140 | 138 | JD113 | 77 | 79 | 78 |
| JD127 | 133 | 139 | 136 | JD114 | 79 | 81 | 80 |
| JD128 | 136 | 142 | 139 | JD115 | 82 | 82 | 82 |
| JD129 | 136 | 142 | 139 | JD116 | 79 | 79 | 79 |
| | | | | JD118 | 89 | 89 | 89 |
| Experimental Average | 134 | 140 | 137 | | 81 | 81 | 81 |
| Theoretical Average | -- | -- | 140 | | -- | -- | 89 |

## 14.172 Limit Loads

Marcal and Turner (14.15) give experimental axial limit loads for two bellows; however, these data are not particularly pertinent to pipeline bellows because a deflection rather than a load is applied.

Limit pressures are of interest. As discussed in paragraph 14.153 and 14.154, the pressure that causes "squirm" is a significant limit pressure. This limitation applies to multiconvolution bellows. For a single convolution bellows, failure probably will consist of rupture. For toroidal-shaped convolutions, the burst pressure may be reached without significant distortion of the bellows shape.

The only published* data on limit pressures known to the writer are given in References (14.8) and (14.11). These data are principally concerned with "squirm" in multiconvolution bellows.

## 14.173 Fatigue

Presumably a number of major bellows manufacturers have each run a significant number of fatigue tests on bellows. With one exception, however, none of the manufacturers known to the writer definitely assign permissible displacements on the basis of fatigue tests. The one exception is Tube Turns (Division of Chemetron).

The following excerpts are from available published data on fatigue tests of bellows. Unless otherwise specifically noted, and insofar as can be determined from the published data, the following general conditions apply to the fatigue tests:

(1) Bellows material--austenitic stainless steel

(2) Test temperature--room

*Probably several bellows manufacturers have data of this type. At least one manufacturer is known by the writer to have run fairly extensive tests on "squirm" pressures and burst pressures on formed bellows.

(3) Environment: air, nitrogen or helium, or mixtures thereof

(4) Failure is defined as a crack through the bellows wall, detected by leakage.

Samans and Blumberg[14.24] give fatigue test data on ten types of 12-inch nominal size bellows joints. Four of the types are welded; six are formed; the formed type all had external T-ring support (see Table 14.1 for nomenclature).

Probably the most thorough and well-documented set of fatigue tests on U-shaped formed bellows are those performed by Tube Turns under the direction of A.R.C. Markl. These constitute 214 cyclic tests on bellows of nominal sizes 3 through 20 inches; 107 on U-shaped without reinforcing and 107 on U-shaped with external ring support. One-hundred and sixty six tests are with axial displacement; 48 with offset displacement. Unfortunately, the detailed documentation of these data is considered proprietary.

Winborne[14.22] gives fatigue test data on ten 20-inch nominal size bellows; one tested in air at 70 F, the other nine tested with sodium at 1200 F on the inside, air entrapped in thermal insulation on the outside. Loading consisted of cyclic axial displacements*. The displacement rate was 0.05 inch/second, with a hold time of 20 seconds at each end of the displacement. With three exceptions, the fatigue tests were run in increasing magnitude steps of axial displacement. Three of the ten bellows tests were discontinued prior to occurrence of fatigue failure. The ten test specimens were made up of (nominally) seven different bellows. These are classifiable, per Table 14.1, as one semitoroidal, six U-shaped, one U-shaped with an

*No mention is made of internal pressure; presumably the internal pressures were close to zero in all tests.

external tee-ring support, and one toroidal. The tenth bellows, of a type not included in Table 14.1, was U-shaped with an internal rib at the convolution roots--presumably welded thereto.

Anderson[14.8] gives an extensive collection of fatigue test data on bellows. Anderson states that: "Bellows manufacturers* submitted most of the data on toroidal and reinforced bellows. Results from the bellows test program of the Rocketdyne Division of North American Aviation provided the major source of data on convoluted bellows." The source of data is not indicated in the detailed results; however, the data from Samans and Blumberg[14.24] and Winborne[14.22] can be recognized.

The fatigue test data are grouped as follows:

A. Convoluted--45 tests

These are classifiable as semitoroidal or U-shaped per Table 14.1. Loadings consist of:

- Axial displacement--14 tests
- Angular displacement--27 tests
- Axial plus offset displacements--4 tests.

In most tests, a (presumably) static pressure was maintained during the cyclic displacement tests.

B. Ring reinforced--43 tests

These are probably**classifiable as U-shaped with external T-ring support per Table 14.1. Loadings consisted of:

- Axial displacement--28 (one with static internal pressure)
- Axial displacement plus cyclic pressure--12 (these are Samans and Blumberg tests)

---

*Anderson specifically acknowledges Zallea Brothers and Solar Aircraft Company for data from tests on bellows.

**Details of the ring reinforcement are not given.

- Axial plus offset displacement--1
- Angular displacement--2 (both with static internal pressure)

Anderson states that "one of these bellows was tested at 1050 F; all others were tested at room temperature." However, the tabulated results indicate that two tests were run at 1050 F.

C. Toroidal--20 tests

These are classifiable as toroidal per Table 14.1. Loadings consisted of:

- Axial displacement--3 (one with static internal pressure)
- Axial displacement plus cyclic pressure--15
- Angular displacement--2 (both with static internal pressure)

D. High temperature--18 tests

These are tests run with liquid sodium at 1200 F on one surface of bellows, either the inside or outside surface. The first nine tests listed are taken from Winborne(14.22). The second nine tests are on small bellows (3-inch nominal size). These are described as "convoluted" and are classifiable as U-shaped per Table 14.1. Details of the test procedure (in particular, cyclic rate or hold time) are not given.

Apparently, the loading in all tests consisted of axial displacement with either zero or negligible internal pressure. All except two tests were run in increasing magnitude steps of axial displacement. Six of the 18 tests were discontinued prior to failure.

Anderson uses the fatigue test data at room temperature to develop some semiempirical correlation equations for estimating the fatigue life of bellows. The correlation method starts with the equations for elastic stresses as developed by Clark(14.3) plus some semiempirical corrections for dimensional parameters where Clark's equations are not applicable. The elastic stresses due to internal pressure and displacement are then combined so as to give a "best-fit" correlation between combined stresses and fatigue life. On the whole, the author has succeeded quite well in achieving correlations between the data and his equations. McKeon(14.7) discusses the correlation method used by Anderson and compares it with that used by M. W. Kellogg. McKeon's Table 3 appears to be for the set of bellows tested by Tube Turns.

The writer questions, in particular, Anderson's correlation procedure with respect to internal pressure. Apparently, the correlation procedure considers static pressure as entirely equivalent to the peak pressure of a pressure cycle. That is, for example, a static pressure of 150 psi is equivalent to the cyclic pressure of 0 to 150 psi. This leads to the obviously unsatisfactory conclusion that a static internal pressure will cause fatigue failure of a bellows.

With respect to the cyclic life at room temperature and (presumably) air environment relative to cyclic life at 1200 F with sodium on one surface (air or helium on other surface), one notes that comparisons are difficult because in all except two of the high-temperature tests, axial displacements were varied during the cyclic test. However, for these two tests (bellows Nos. 8 and 10 of Winborne's (14.22) data), the following comparisons are pertinent.

Bellows No. 8 was U-shaped without reinforcing; No. 10 was U-shaped with an external tee-ring support. The correlation combined stress equations developed by Anderson for room temperature are different for these two types of bellows; however, consistently applying the same correlations to the two bellows tested at 1200 F give the following comparisons:

| Bellows, No. | Cycles to Failure, N | S | $N_{70}$ for S | $S_{70}$ for $N_{1200}$ |
|---|---|---|---|---|
| 8 | 1590 | 82,800 | 398,000 | 302,000 |
| 10 | 2730 | 71,700 | 480,000 | 254,000 |

The value under "$N_{70}$ for S" is the number of cycles expected at room temperature by Anderson's correlation. The cyclic life at 1200 F is reduced by a factor of 250 for bellows No. 8; a factor of 175 for bellows No. 10. The column $S_{70}$ for $N_{1200}$ gives the stress required at room temperature to produce failure in the observed number of cycles at 1200 F. This is significant in relationship to Anderson's suggestion to use, as allowable stresses, the B31.1 (USAS Power Piping Code) relationship:

$$S_A = 1.25\ (S_c + S_h).$$

For Type 304 steel at 70 F, $S_c + S_h$ = 37,500 psi; at 1200 F, $S_c + S_h$ = 18,750 + 5500 = 24,250 psi. This would imply that reducing stresses by a factor of 1.54 would result in equal fatigue lives. The tabulated values of $S_{70}$ for $N_{1200}$ would indicate that stresses must be reduced by a factor of 3.65 (bellows No. 8) or 3.54 (bellows No. 10) to obtain equal fatigue lives. These two* tests indicate that Anderson's procedure may be quite unconservative in this respect.

In these tests, it is not possible to isolate that part of the reduction in fatigue life due to temperature (1200 F) from that part due to environment (sodium versus, presumably, air).

Trainer, et al.(14.11) give fatigue test data on both formed and welded bellows. The data include 69 tests on U-shaped bellows and 71 tests on welded

* Most of the other tests at 1200 F, when evaluated using Minor's hypothesis for variable fatigue loads, give relative (100 F to 1200 F) results more nearly in agreement with $S_A = 1.25(S_c + S_h)$.

bellows. Loading consisted of axial displacement plus various magnitudes of static internal pressure. Tests of welded bellows gave wide scatter in fatigue life for nominally identical bellows, depending upon minute details of weld irregularities. These will not be discussed further herein.

Test results are summarized in Figures 14.2 for austenitic stainless steel bellows and in Figure 14.3 for Inconel-718 bellows. Also shown in these figures are strain-controlled fatigue tests on coupons. In Figure 14.2, sufficient data were available to establish a "scatterband" for such coupon tests. In Figure 14.3, only one set of data were available. For the bellows tests, the strain ranges due to the axial displacement were calculated using NONLIN[14.10]. One notes that the bellows fatigue results are significantly below the coupon data. The authors offer the following possible reasons for this relationship:

(1) Variations in convolution shape and in material thickness may result in strain ranges different from those calculated for a representative (average measured dimensions) model.

(2) Each failure would be expected to occur at the "weakest" convolution and material point; the results would be analogous to the shortest fatigue life of some 50 coupon specimens for these 8 to 12 convolution test bellows.

(3) The strains in the bellows were biaxial, in contrast to the uniaxial straining of the metal coupons.

(4) In some of the bellows tests there was a non-zero mean strain.

(5) The surface finish of the bellows was not equivalent to the highly polished condition of the coupon tests.

The writer would suggest one additional possible reason. Strains were computed on an elastic basis. Most of the tests were conducted at strain

8

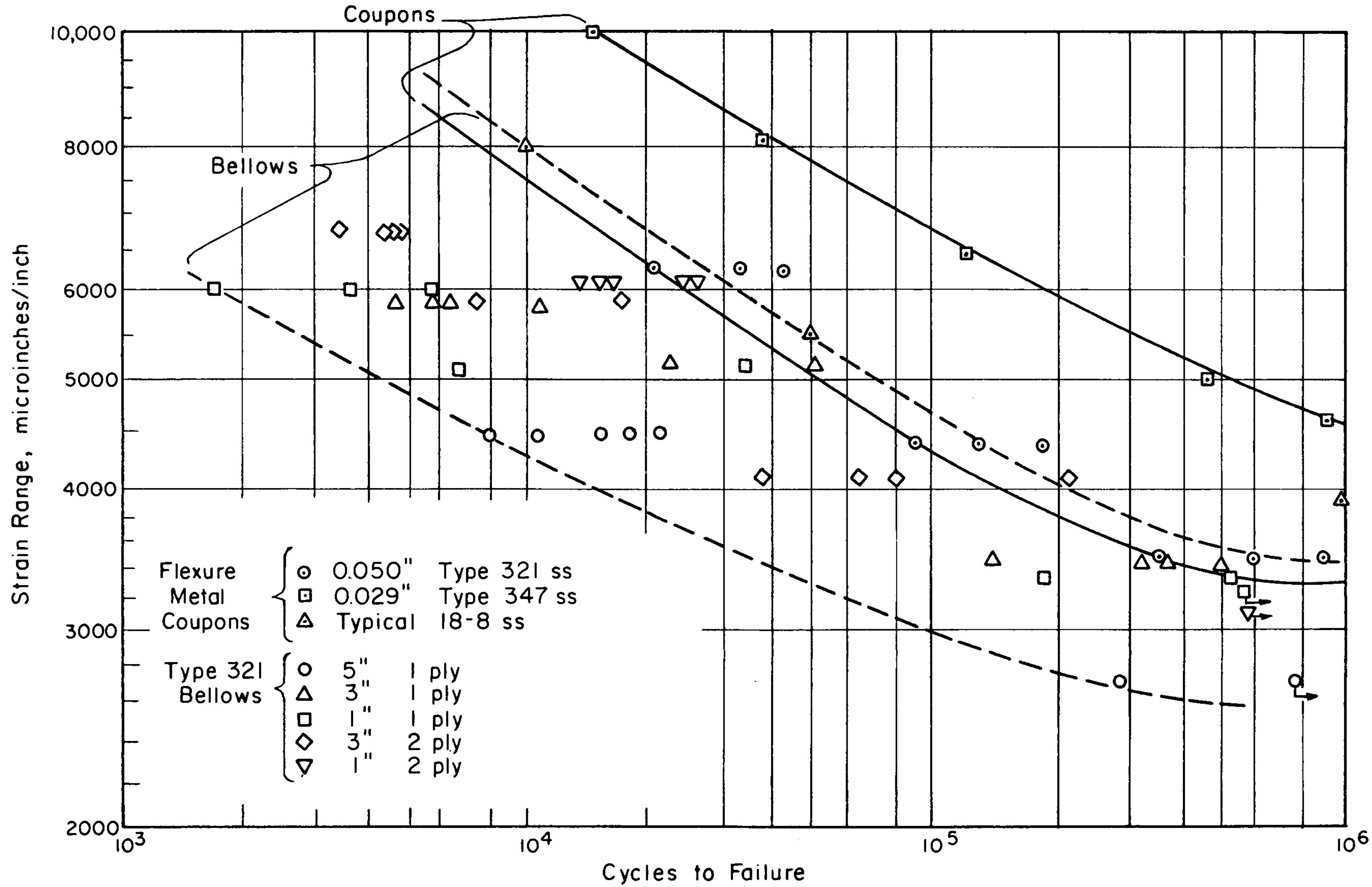


FIGURE 14.2. FATIGUE RESULTS OF STAINLESS STEEL FORMED BELLOWS AND STAINLESS STEEL COUPONS(14.11)

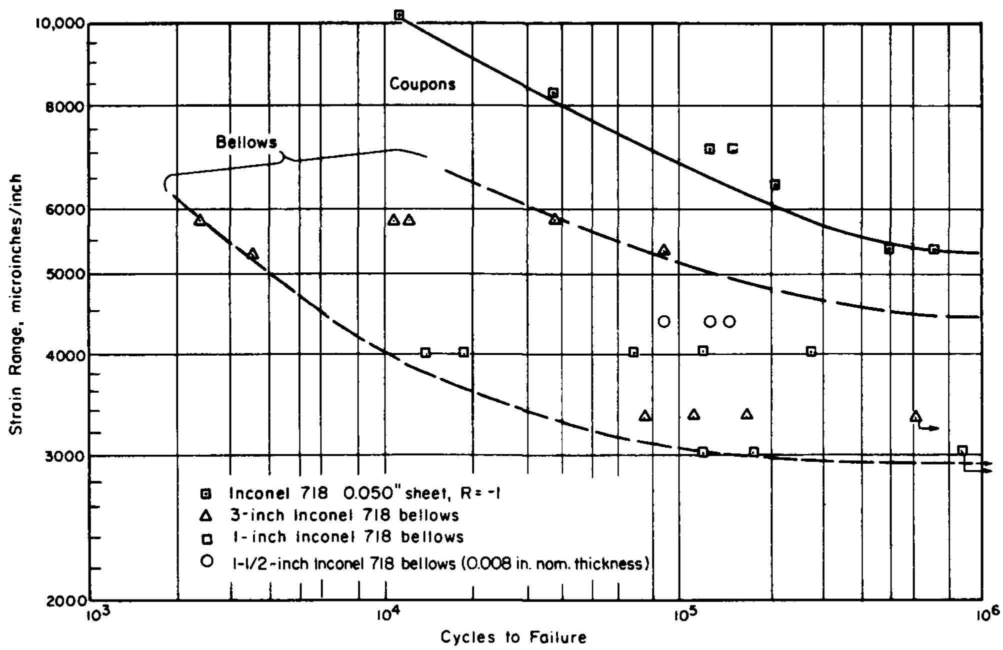


FIGURE 14.3. FATIGUE RESULTS OF INCONEL FORMED BELLOWS AND INCONEL COUPONS (14.11)

ranges from 0.003 to 0.006 in./in. This implies a stress range of around 90,000 to 180,000 psi. While the authors present some test data indicating that "shake-down" is almost complete after 6 cycles of ±0.375-inch axial displacement on a 5-inch bellows made of 321 material, there was still some evidence of plastic straining at 6 cycles. For higher axial displacements used in the tests, and for all bellows made of Inconel-718, continued plastic straining may have occurred throughout the tests. An elastic-plastic analysis, including strain hardening, might give some theoretical guidance in this respect. The problem is complicated by the varying degree of cold work present in bellows material furnished "as formed".

## 14.18 USAS B31.7 Requirements for Bellows

Expansion joints are not currently permitted in B31.7 for Class I piping (see paragraph 1-704.7.1). Table 14.4 is an abstract of requirements for bellows expansion joints from B31.7 for Class II piping. The design requirements contained in Table 14.4 appear to be based on the report by Anderson(14.8).

There are several questions concerning this procedure:

(1) In 2-709.1.2(a), what is an acceptable method of calculating membrane stresses in, for example, a ring-reinforced bellows with three convolutions?

(2) In 2-707.1.2(b), why are membrane plus bending stresses restricted for unreinforced bellows <u>only</u>?

(3) In 2-707.1.2(c):

(a) Is S and/or $S_f$ intended to be a stress range or a stress amplitude?

(b) Why is the "squirming effect" included only for bellows with angular rotation?

(c) How does one handle combinations of cyclic pressure and cyclic displacements? Must the test data include cyclic pressure for determining $S_f$?

(d) Is $(S_c + S_h)$ an appropriate parameter for extrapolating room temperature tests to elevated-temperature service? (See discussion under Paragraph 14.173 on Anderson's data.)

The data presented herein indicate that, aside from instability or squirm, elastic stresses in bellows expansion joints (of known dimensions) can be calculated with accuracy equal to or exceeding the accuracy of stress calculation of a number of other piping components. Accordingly, one might apply the same kinds of calculated elastic stress limits to bellows expansion joints as used for all other pressure vessel or piping components under the ASME Nuclear Vessels Code or the USAS Nuclear Piping Code. Such a procedure might well result in bellows of adequate reliability. However, this is not the procedure presently in USAS B31.7. The difference is perhaps best illustrated by the following example.

Assume that a bellows joint is to be designed for 1000 displacement cycles; internal pressure is negligible and, for the sake of simplicity in the discussion, the temperature is 100 F. Now, from Anderson [14.8] Figure 12b for 1000 cycles, a value of $S_f$ = 340,000 psi (if $S_f$ is intended to be a stress range) or $S_f$ = 170,000 psi (if $S_f$ is intended to be a stress amplitude) can be obtained. Under USAS B31.7 (see Table 14.4), we are permitted an allowable stress of $S_f/1.75$ = 194,000 psi or 97,000 psi. The pertinent elastic stress in question is classifiable as a secondary bending stress and would be limited to a range of $3S_m$ = 60,000 psi for austenitic stainless steel at 100 F. The writer's best guess as to the intent, based on Anderson's report, is that both $S_f$ and S are intended to be stress ranges. If so, the B31.7 design procedure and Anderson's data would permit a secondary stress range of 9.7 $S_m$. For the B31.7 bellows design procedure to be in harmony with the Nuclear Vessel Code rules, it would be necessary to classify the calculated bending stress as a "peak" stress; a classification which does not fit the rules.

TABLE 14.4. USAS B31.7 (FEBRUARY, 1968) REQUIREMENTS FOR BELLOWS EXPANSION JOINTS, CLASS II PIPING

(page 1 of 8)

2-709.1 Expansion Joints

2-709.1.1 General

Expansion joints of the bellows, sliding, ball, or swivel types may be used to provide flexibility for Class II piping systems. The design of the piping systems and the design, material, inspection, and testing of the expansion joints shall conform to this Code, and shall comply with the following requirements:

(a) Piping system layout, anchorage, guiding, and support shall be such as to avoid the imposition of motions or forces on the expansion joints other than those for the absorption of which they are both suitable and intended. Bellows expansion joints are normally not designed for absorbing torsion (rotation about the axis). Sliding expansion joints are normally not designed for absorbing bending (angulation in the plane of the axis). In sliding and bellows expansion joints used for absorbing axial motion, the hydrostatic end force caused by fluid pressure and the forces caused by friction resistance and/or spring force must be resisted by rigid end anchors, cross-connections of the section ends, or other means. Where reaction to hydrostatic end force acts on pipe, guides must be provided to prevent buckling in any direction.

(b) The expansion joints shall be installed in such locations as to be accessible for scheduled inspection and maintenance, and for removal and replacement.

TABLE 14.4. (Continued)
(page 2 of 8)

(c) Expansion joints employing mechanical seals shall be sufficiently leak-tight to satisfy radiological safety requirements. The system designer shall specify the leak-tightness criteria for this purpose.

(d) Materials shall conform to the requirements of Chapter 1-III, except that no sheet material in the quench, age, or air-hardened condition shall be used for the flexible element of a bellows joint. If heat treatment is required, it shall be performed either after welding the element into a complete cylinder or after all forming of the bellows is completed, the only welding permissible after such treatment being that required to connect the element to pipe or end flanges.

(e) All welded joints shall comply with the requirements of Divisions 2-727 and 2-736.

2-709.1.2 Bellows-Type Expansion Joints

Bellows may be of the unreinforced or reinforced-convoluted type, toroidal type, or welded construction. The design shall conform to the following requirements.

(a) The membrane stresses due to pressure shall not exceed the allowable stress intensity value given in Table A.1 of Appendix A for the material at the design temperature.

(b) In unreinforced bellows, the sum of the membrane and bending stresses due to internal pressure shall not exceed 1.5 times the allowable stress intensity value for the material at design temperature.

TABLF 14.4 (Continued)

(page 3 of 8)

(c) The combination of membrane, bending, and torsional stresses (S) in the bellows due to internal pressure and deflection, multiplied by a safety factor of 1.75 shall not exceed the value defined by the following equation

$$1.75S = S_f \frac{S_c + S_h}{2S_t}$$

where $S$ = combined stress due to pressure and deflection, where the calculation of the individual stress components and their combination may be performed by any analytical method based on the elastic shell theory, provided that the same method is used for determining $S_f$. In case of angular deflection of the convolutions, the increase caused in bending stresses by the squirming effect of the internal pressure shall be included in calculating S;

$S_f$ = combined stress to failure at design cyclic life (number of cycles to failure) obtained from plots of stress versus cyclic life based on (previously available and/or new) data from fatigue tests of a series of bellows of the same design and basic material, at a given temperature (usually room temperature), evaluated by a best-fit continuous curve or series of curves;

$S_c$ = basic material allowable stress intensity value at minimum (cold) temperature from Table A.1 of Appendix A;

$S_h$ = basic material allowable stress intensity value at maximum (hot) temperature from Table A.1 of Appendix A;

$S_t$ = basic material allowable stress intensity value at test temperature from Table A.1 of Appendix A.

(d) The bellows manufacturer shall demonstrate by testing of prototype bellows the ability of the bellows to withstand a room-temperature pressure test to 2.25 times the equivalent cold-working pressure without squirm, and to 2.75 times the equivalent cold-working pressure without rupture. For joints used in axial or lateral motion, these tests

TABLE 14.4. (Continued)
(page 4 of 8)

shall be performed with the bellows fixed in the straight position at the maximum length expected in service; for rotation joints, the bellows shall be held at the maximum design rotation angle.

The equivalent cold-working pressure is defined as the design pressure multiplied by the ratio $S_c/S_h$ where $S_c$ and $S_h$ are as defined in Item (c) above, except that they refer specifically to room temperature and normal operating temperature.

For the specific bellows, this ability may be demonstrated by a single test on a duplicate bellows, except that the prototype bellows used for the test to rupture need not have more than three convolutions.

A consistent series of bellows of the same basic element and reinforcement design, class of material, and methods of manufacture may be qualified over a given size range by demonstrating predictability of squirm and rupture pressures by the theoretical formulas adequately correlated with test data. Correlation shall be considered adequate if the formulas conservatively predict average performance and not more than one out of 10 specimens squirm or rupture at less than 80 percent of the predicted pressure. The number of test specimens used in the correlation shall be such that each size and thickness is represented by at least one specimen for rupture, and two specimens for squirm, of a pitch diameter and thickness no less than two-thirds, nor more than three-halves of its own. Of the specimens used for squirm qualifications, one-half of the number shall have the maximum number of convolutions for

TABLE 14.4. (Continued)
(page 5 of 8)

which the size is to be qualified and the other half shall preferably have two fewer convolutions.

Squirm shall be considered to have occurred if, upon removal of pressure, the bellows axis is found to have deformed into a curve, resulting in lack of parallelity or uneven spacing of adjacent convolutions. This deformation shall not be construed as evidence of squirm, unless the permanent change of pitch of any convolution or deviation from parallel between adjacent convolutions exceeds the value of

$$\frac{0.0003\ (d_c - d_r)^2}{t}$$

where $d_c$ and $d_r$ = normal bellows diameters at convolution crest and root, respectively,

$t$ = nominal bellows element thickness.

(e) Where necessary to carry the pressure, the cylindrical ends shall be reinforced by suitable collars. The design of the attachment between pipe and bellows element and/or reinforcement shall assure that no detrimental stresses will be generated that may cause the failure of the bellows material or weldment. The distance between the bellows attachment weld line and the tangent line with the root of the end convolution shall be equal to or greater than the smaller of $\sqrt{d_r t_r}$ or 2-1/2 inch where $d_r$ and $t_r$ are the diameter and bellows thickness of the convolution root or cylinder end. Attachment welds may be fillet or butt-type welds.

TABLE 14.4 (Continued)
(page 6 of 8)

(f) The natural frequency of the expansion joint assembly shall not be near the frequency of any vibrations occurring in the piping system as specified by the piping designer.

(g) The inspection and testing of bellows-type expansion joints shall be performed in accordance with the provisions of Subdivisions 2-736.6 and 2-737.5 of this Code.

2.736.6 Inspection of Bellows-Type Expansion Joints

The following examinations are required to qualify bellows-type expansion joints for installation in nuclear piping systems.

(a) The bellows material shall be determined to be free of injurious defects by definitive inspection methods prior to forming.

(b) After forming, the bellows shall be determined to be free from injurious defects by definitive inspection methods consistent with and at least as sensitive as the inspection methods applied to the piping system within which the joint is to be installed.

(c) All welds of the bellows element shall be radiographed after forming or fabrication, except that radiography of the longitudinal seam welds in rolled form, hydraulically or bulge-formed bellows may be performed on the tubing prior to forming.

(d) All welds in the expansion joint assembly shall be radiographed except that where radiography is not practical or meaningful (e.g., for bellows attachment welds to pipe flange), liquid penetrant examination may be substituted.

TABLE 14.4. (Continued)
(page 7 of 8)

(e) The completed expansion joint assembly shall be examined visually. Variances of fabrication, such as notches, crevices, nonuniform or excessive material thinning, buildup or upsetting which may serve as points of local stress concentration are not allowed.

(f) Unreinforced or reinforced convoluted bellows elements shall meet the following tolerances:

(1) The variation of the root of cylindrical end thickness, $t_r$, from the nominal or specified thickness, t, shall not exceed the values given in Table III of ASTM Specification A-240.

(2) The ratio $t_c/t_r$ of crest-to-root thickness shall not be less than given by the formula

$$t_c/t_r = \sqrt{d_r/d_c} - 0.04$$

where $d_r$ and $d_c$ = the nominal bellows diameters at convolution root and crest, respectively.

(3) The depth, w, of convolution (one-half the difference between outside crest and outside root diameter), as measured with the bellows at nominal length, shall not vary by more than $\pm(w + 2)/50$ from the nominal or specified dimension.

(4) The outside crest diameter, $d_c$, as measured with the bellows at nominal length, shall not vary by more than $\pm(w + 2)/32$ from the nominal or specified dimensions.

(5) Tolerances on the root diameter of an unreinforced bellows shall not exceed ±1/8 inch and on a reinforced bellows shall

TABLE 14.4. (Continued)
(page 8 of 8)

not exceed ±3/32 inch in neutral position. On a reinforced bellows, the reinforcing ring shall be in intimate contact with the bellows material at the root of the convolution when the bellows is in relaxed position.

(6) The outer meridional radius of convolution at root or crest, as measured over a 90-degree arc, shall not be less than 5t for single-ply bellows, nor less than (4 + n)t for bellows of n plies. Tolerances of the root and crest radii shall not exceed ±15 percent of the nominal radius.

(7) The pitch of convolutions or center-to-center distance from crest to crest of the convolution shall not deviate more than ±8 percent from the nominal pitch.

2-737 LEAK TESTS

The requirements for leak test shall be the same as stated in Division 1-737.

2-737.5 Testing of Bellows-Type Expansion Joints Prior to Installation

This paragraph covers testing requirements for qualifying bellows-type expansion joints in nuclear piping systems.

(a) The completed expansion joint shall be leak tested to a sensitivity equal to or better than any other pressure part in the piping system within which the joint is to be installed.

(b) The completed expansion joint shall be subjected to a hydrostatic test in accordance with the applicable provisions of paragraph 1-737.4.1 while in design deflection position. Bellows showing visually detectable squirm during this test are not acceptable.

## 14.2 Slip Joints

A slip joint provides the same function as a bellows joint used in axial displacement. It shares the bellows requirements for anchoring and guiding but is usually immune from fatigue failure or stress-corrosion cracking. The problem with packed joints is that of maintaining the packing to restrict leakage to a tolerable amount. For nuclear piping applications, the tolerable leakage usually is so small that this type of joint is not acceptable.

General discussions of the types of packed joints and their maintenance problems are given by York[14.25], Hannah[14.26], and by Brock[14.27].

No quantitative data on design or performance characteristics of packed joints are known to the writer. Presumably, the body is designed for internal pressure by normal commercial methods. The force-deflection characteristics (which might be a significant function of the packing, packing condition, and gland tightness) are not known nor is the capacity of such joints to absorb offset forces or moments.

## 14.3 Swivel Joints and Ball Joints

Swivel joints permit rotation in one plane while ball joints permit rotation in all planes. These kinds of joints are fairly common in some small-size, low-pressure, noncritical piping, but have not, up to the present time, found much application in larger sizes and higher pressures. The swivel joint is used for connecting piping to a rotating machine such as a dryer drum. A general discussion of available types, sizes, and pressure capacities of such joints is given by York[14.25] and by Brock[14.27]. No quantitative data on design or performance characteristics are available to the writer. Ball joints are used in at least one heating-air conditioning piping system with satisfactory service experience according to Liles[14.28].

## 14.4 Summary and Recommendations

### 14.41 Summary

(1) The data presented herein indicate that, aside from instability or squirm, the elastic stresses in bellows expansion joints can be calculated with an accuracy equal to or exceeding the accuracy of stress calculations for a number of other piping components.

(2) Existing practice in rating typical pipeline bellows is such that the bellows may be used at calculated stresses far exceeding the elastic range. The design procedure in B31.7 for bellows in Class II piping systems continues this practice.

(3) One of the principal sources of unreliability of bellows is caused by stress-corrosion cracking or corrosion accelerated fatigue.

(4) Correlation of fatigue test data with calculated elastic stresses has been reasonably successful, even where the calculated stresses are far above the elastic range.

(5) Instability (squirm) and vibration of bellows are recognized problems.

(6) Essentially no quantitative data are available on performance characteristics of "packed" expansion joints.

### 14.42 Recommendations

From the standpoint of B31.7, perhaps the most immediate need is a careful review of the present B31.7 rules for bellows in Class II piping and available data to:

(a) Clarify the meaning of $S$ and $S_f$ (stress range or stress amplitude?)

(b) Determine if the parameter $(S_c + S_h)$ is appropriate for extrapolating room-temperature tests to elevated-temperature service

(c) Standardize methods for calculating stresses due to pressure and deflection; and for combining those stresses for fatigue evaluation

(d) Review the criteria to be applied to bellows design; i.e.,

1) Safety factor on burst
2) Safety factor on gross deformation
3) Safety factor on squirm
4) Safety factor on fatigue
5) Safety factor on hardware

(The last item refers to design of hinges, gimbals, etc.).

## REFERENCES


(14.1) Standards of the Expansion Joint Manufacturers Association, Third Edition, 1969, Expansion Joint Manufacturers Association, 53 Park Place, New York, New York.

(14.2) Salzmann, F., "Ueber die Nachgiebigkeit von Wellrohrexpansionen", Schweizerische Bauzeitung, Vol. 127, No. 11, pp 127-130 (1946)

(14.3) Clark, R. A., "On the Theory of Thin Elastic Toroidal Shells", Journal of Mathematics and Physics, Vol. 29, pp 146-178 (1950).

(14.4) Dahl, N. C., "Toroidal-Shell Expansion Joints", Journal of Applied Mechanics, Vol. 20, Trans. ASME, Vol. 75, pp 497-503 (1953).

(14.5) Turner, C. E., "Stress and Deflection Study of Flat-Plate and Toroidal Expansions Bellows, Subjected to Axial, Eccentric or Internal Pressure Loading", Journal of Mechanical Engineering Science, Vol. 1, No. 2, pp 130-143 (1959)

(14.6) Laupa, A. and Weil, N. A., "Analysis of U-Shaped Expansion Joints", Journal of Applied Mechanics, Vol. 29, Trans. ASME, Vol. 84, Series E, pp 115-123 (1962).

(14.7) McKeon, J. T., "Primary Piping Flexibility Analysis for Liquid-Metal-Cooled Fast Breeder Reactors", Nuclear Engineering and Design, Vol. 7, pp 427-441 (1968).

(14.8) Anderson, W. F., "Analysis of Stresses in Bellows, Part I, Design Criteria and Test Results, and Part II, Mathematical", AEC Research and Development Report NAA-SR-4527 (1964-1965).

(14.9) MOLSA, See "Analysis of Shells of Revolution Subjected to Symmetrical and Nonsymmetrical Loads", by A. Kalnins, ASME J. of Applied Mechanics, Sept. 1964.

(14.10) NONLIN, See "On Nonlinear Analysis of Elastic Shells of Revolution", by A. Kalnins and J. F. Lestingi, J. Applied Mechanics, Trans. ASME, Series E, Vol. 34, No. 1, pp 59-64 (March, 1967).

(14.11) Trainer, Hulbert, Lestingi, and Keith, "Final Report on the Development of Analytical Techniques for Bellows and Diaphragm Design", Technical Report No. AFRPL-TR-68-22, March, 1968. Battelle Memorial Institute (Columbus, Ohio) for Air Force Rocket Propulsion Laboratory, Edwards Air Force Base, California

(14.12) FEELAP, See "Elastic-Plastic Analysis of Two-Dimensional Stress Systems by the Finite Element Method", by P.V. Marcal and I. P. King, Int. J. Mech. Science, Vol. 9, pp 143-155 (1967).

(14.13) NONLEP, See, "On Nonlinear Elastic-Plastic Analysis of Shells of Revolution", by J. C. Gerdeen, Battelle-Columbus Special Report, August 1968.

REFERENCES (contd)

(14.14) Haringx, J. A., "Instability of Bellows Subjected to Internal Pressure", Philips Research Reports, Vol. 7, pp 189-196 (1952).

(14.15) Marcal, P. V. and Turner, C. E., "Elastic Solution to the Limit Analysis of Shells of Revolution with Special Reference to Expansion Bellows", J. Mech. Eng. Science, Vol. 3, pp 252-257 (1961).

(14.16) Newland, D. E., "Buckling of Double Bellows Expansion Joints Under Internal Pressure", J. Mech. Eng. Science, Vol. 6, No. 3 (1964).

(14.17) Daniels, V. R.,"Dynamic Aspects of Metal Bellows", The Shock and Vibration Bulletin (January, 1966), Bulletin 35, Part 3, USNRL, Washington, D. C.

(14.18) SHOREF, See "On Free and Forced Vibration of Rotationally Symmetric Layered Shells", by A. Kalnins, J. Applied Mechanics, Vol. 32, pp 941-943 (1965).

(14.19) Feely, F. J. and Goryl, W. M., "Stress Studies on Piping Expansion Bellows", ASME J. of Applied Mechanics, June, 1950.

(14.20) Turner, C. E. and Ford, H., "Stress and Deflection Studies of Pipeline Expansion Bellows", Proc. Instn. Mech. Engrs., Vol. 171, pp 526-562 (1957).

(14.21) Bowden, A. T. and Drumm, J. C., "Design and Testing of Large Gas Ducts", Proc. Instn. Mech. Engrs., Vol. 174, pp 119-157 (1960).

(14.22) Winborne, R. A., "Stress and Elevated Temperature Fatigue Characteristics of Large Bellows", Atomics International Report No. NAA-SR-9762, Sept. 15, 1964.

(14.23) Winborne, R. A., "Simplified Formulas and Curves for Bellows Analysis", Atomics International Report No. NAA-SR-9848, August 1, 1964.

(14.24) Samans, W. and Blumberg, L., "Endurance Testing of Expansion Joints", ASME Paper No. 54-A-103.

(14.25) York, J. E., "Joints to Permit Movement", Heating and Ventilating, January p. 85, February p 93, and March p 87, 1949.

(14.26) Hannah, M. J., "Packed Slip or Packless Bellows Expansion Joints", Heating, Piping and Air Conditioning, May 1968.

(14.27) Brook, J. E., "Expansion and Flexibility", Chapter 4 of Piping Handbook, 5th Edition, 1967, McGraw-Hill Book Co.

(14.28) Liles, R. M., "Ball Joints Accomodate Expansion in New Piping at O'Hare Field", Heating, Piping and Air Conditioning, pp 116-117, February, 1967.

# CHAPTER 15

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# 15. PIPING SYSTEM SUPPORTING ELEMENTS

The control of the motion of piping systems is an important consideration in design. The term "supporting elements" is used in the USAS Piping Code, and herein, as including any device which prevents, resists or limits the movement of the piping. The following terminology* is used herein:

Brace. A device primarily intended to resist displacement of the piping due to the action of any forces other than those due to thermal expansion or to gravity. Note that with this definition, a damping device is classified as a kind of brace.

Anchor. A rigid restraint providing substantially full fixation (i.e., encastre; ideally permitting neither translatory nor rotational displacement of the pipe on any of the three reference axes). It is employed for purposes of restraint but usually serves equally well as restraint, support, or brace.

Stop. A device which permits rotation but prevents translatory movement in at least one direction along any desired axis. If translation is prevented in both directions along the same axis, the term double-acting stop is preferably applied.

Two-axis Stop. A device which prevents translatory movement in one direction along each of two axes. A two-axis double-acting stop prevents translatory movement in the plane of the axes while allowing such movement normal to the plane.

Limit Stop. A device which restricts translatory movement to a limited amount in one direction along any single axis. Paralleling the various stops there may also be: double-acting limit stops, two-axis limit stops, etc.

---

* Terminology is taken from USAS B31.3-1966 (15.1) and the M. W. Kellogg book, "Design of Piping Systems" (15.2).

Guide. A device preventing translatory displacements except along the pipe axis.

Rotational Guide. A device preventing rotational displacements about one or more axes.

Hanger. A support by which piping is suspended from a structure, etc., and which functions by carrying the piping load in tension.

Resting or Sliding Support. A device providing support from beneath the piping but offering no resistance other than frictional to horizontal motion.

Inextensible Support. A support providing stiffness in at least one direction, comparable to that of the pipe.

Resilient Support. A support which includes one or more largely elastic members (e.g., spring).

Constant Support. A support which is capable of applying a relatively constant force at any displacement within its useful operating range (e.g., counterweight or compensating spring device).

Damping Device. A dashpot or other frictional device which increases the damping of a system offering high resistance against rapid displacement caused by dynamic loads, while permitting essentially free movement under gradually applied displacements.

The design of restraints is intimately connected with the piping flexibility analysis (see Chapter 3, Section 3.14). In fact, for an accurate flexibility analysis it is necessary to establish, and use as input, the locations and functions of the various restraints. The flexibility analysis then indicates what loads must be sustained by the supporting elements.

Briefly, the objective of the layout of the piping and its supporting elements is to prevent the following:

(1) Excessive forces or moments on connected equipment (such as pumps and turbines)

(2) Excessive strains in the piping components

(3) Leakage at flanged joints

(4) Resonance with imposed vibrations

(5) Unintentional disengagement of piping from its supports

(6) Excessive sag in piping requiring drainage

(7) Excessive strains in the support elements.

## 15.1 Design of Supporting Elements

The design of supporting elements is discussed in References (15.2) and (15.3). The various sections of the USAS Piping Code[(15.4)] each contain a chapter on the loadings and design of supporting elements. MSS SP-48, "Pipe Hangers and Supports"[(15.5)] was developed as a cooperative effort of representatives of pipe hanger manufacturers. It includes basic design criteria for many types of supporting elements. These references fairly well summarize the state-of-the-art of supporting element design. Reference (15.2), in particular, gives a good discussion of the subject.

Several aspects which merit additional comments or emphasis are discussed in the following.

### 15.11 Supporting Structures

It should be recognized that the loads developed on some supporting elements, particular anchors, may for some piping be very large. In the piping flexibility analysis, it is assumed that an anchor prevents all motions from occurring. It is, of course, not possible to construct an actual anchor which, with non-zero loads, is completely rigid. However, the actual anchor should have negligible small motions under the imposed loads if the flexibility analysis is to be significant*. Design of adequate anchors for large loads may involve large, reinforced concrete foundation blocks and the soil mechanics associated with the design of such foundations.

Many piping systems are supported above ground, either from framework constructed for this purpose or from building frames. The flexibility analysis assumes control of motion at supporting elements; accordingly, displacements of the supporting framework may have to be considered in some cases.

## 15.12 Expansion Joints

Some piping systems employ expansion joints for absorbing temperature displacements or end displacements. These may either be the slip-joint type or bellows type. Such joints must be very carefully guided and anchored. Manufacturer's catalogs(15.6, 15.7) give guidance in this respect. A paper by Hannah(15.8) also emphasizes this aspect. Further discussion is given in Chapter 14.

## 15.13 Vibration

One of the objectives listed for supporting elements is to prevent resonance with imposed vibrations. Vibration of piping systems is discussed in Chapter 16. In the present state-of-the-art, about all that practically can be

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* If an end anchor does move in the direction of applied loads, the flexibility analysis will be conservative. However, in some piping systems an "intermediate anchor" is used. Movement of this anchor may make the analysis for one subsystem conservative but make the analysis for the other subsystem unconservative.

done in this area is to space supports so that the frequencies of known major imposed vibrations will not coincide with fundamental frequencies of the parts of the piping system. Actually, the piping system as installed may be exposed to a broad spectrum of imposed vibrations and, even if fundamental mode vibrations are not prevalent, higher order frequencies may be. It is common practice during early stages of operation of the piping system to inspect for vibration and, if necessary, to add braces or damping devices.

## 15.14 On-Site Inspection

As implied above, the control of vibration in a piping system is difficult to establish in the design stage and on-site inspection is desirable for critical piping systems. This step is a requirement in USAS B31.7 (see Par. 1-701.5.4). In addition to vibration, it should be noted that final adjustments of hangers and supports are usually necessary during start-up. The following quote from Reference (15.2) is pertinent.

> "For critical piping it is desirable to define clearly the installation and subsequent adjustment requirements, and where at all possible to send a design engineer thoroughly familiar with the basic and installation requirements, to assist with and observe the adequacy of the installation. This is particularly important on stiff or large high-temperature piping or where critical materials are involved. In particular, measures for prestress should be properly executed, and the adjustment of special support and restraint fixtures properly accomplished."

## 15.2 Attachment of Supporting Elements to Pipe

The preceding discussion has been concerned with the design of the supporting elements. An equally important aspect concerns the attachment of the elements to the pipe. In some elements, the pipe may simply rest on the element (e.g. supports with rollers) or a clamp may attach the pipe to a hanger. More commonly, the pipe is either above or below atmospheric temperature and insulation is desirable to reduce heat transfer. Because insulating materials are normally not very strong, it is often necessary to transfer the load through the insulation to the pipe by appropriate metal structures. These metal structures are usually welded to the pipe; the transfer of loads to the pipe lead to localized stresses in the pipe. Some of the piping code[15.4] sections (B31.1, B31.3) simply state that : "Consideration shall be given to the localized stresses induced in the piping by the integral attachment". USAS B31.7 uses the same sentence; however, in view of the stress criteria established in Appendix F of B31.7, it would appear that such stresses must remain below certain prescribed limits.

There are a variety of ways of transferring load from supporting elements to the pipe. Some typical designs are shown in Figures 15.1 and 15.2. These are called integral connections because the load transfer member (lugs, brackets, rings, etc.) are welded to the pipe.

The status of stresses at local loads on straight pipe is discussed in Chapter 6. As illustrated by Figure 15.1, attachments may be made at elbows or curved pipe. Attachments may also be made on tees, reducers, caps, etc. Theory*

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* Some of the finite-element computer programs discussed in Chapter 3 could, in principal, be used to calculate stresses at such attachments.

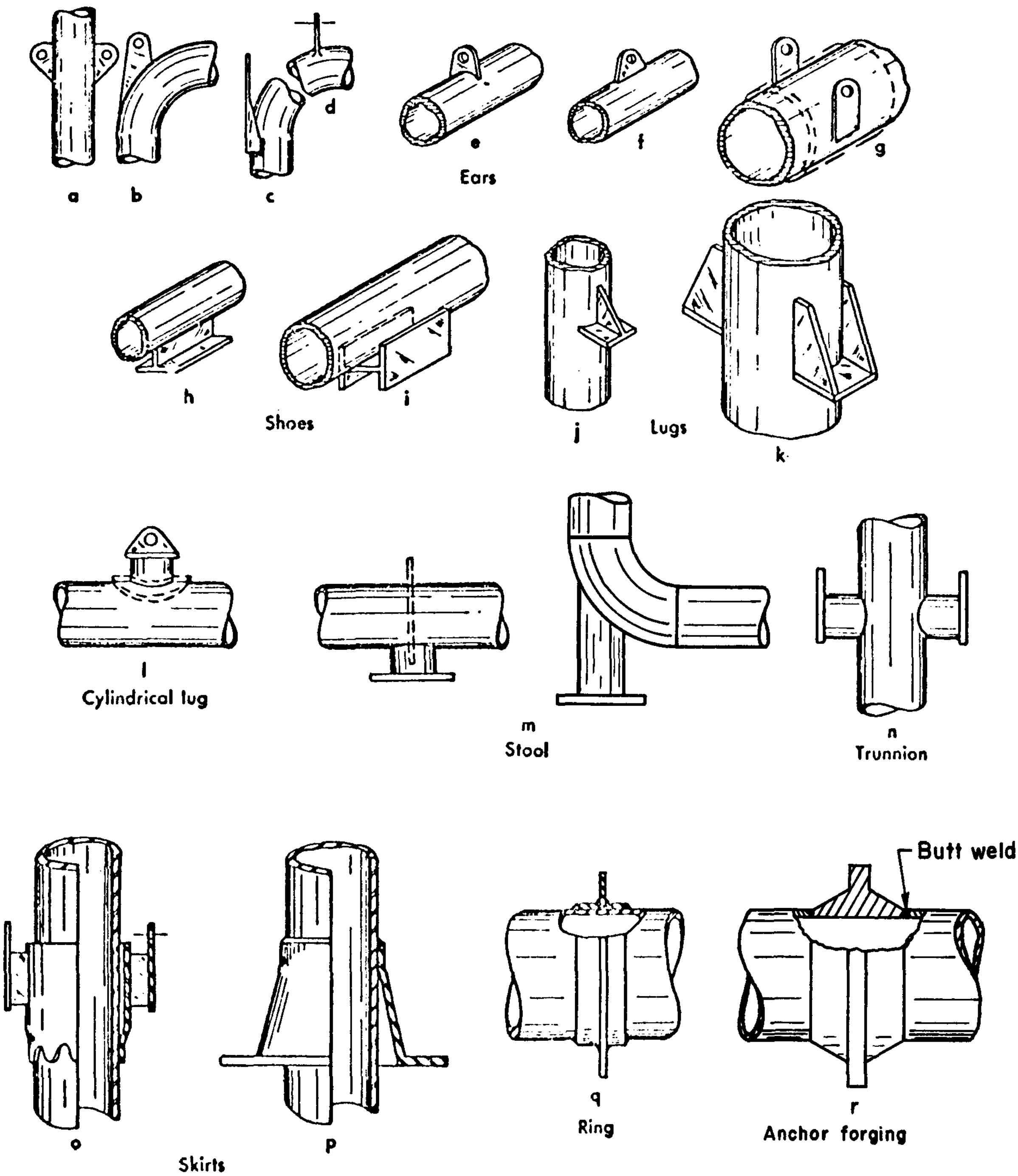
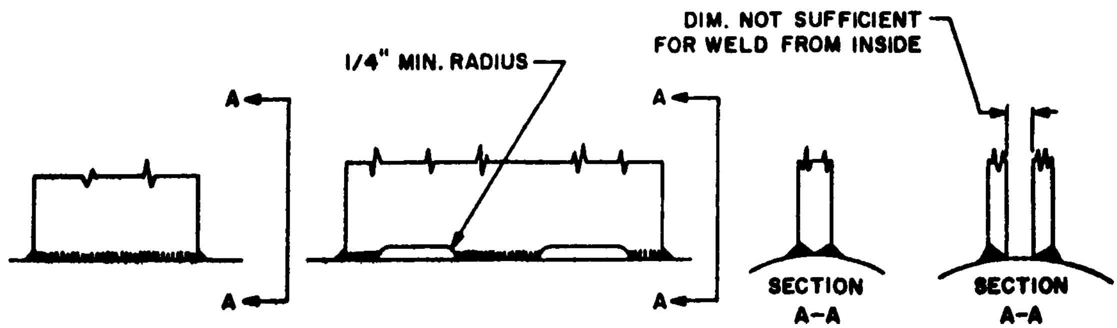
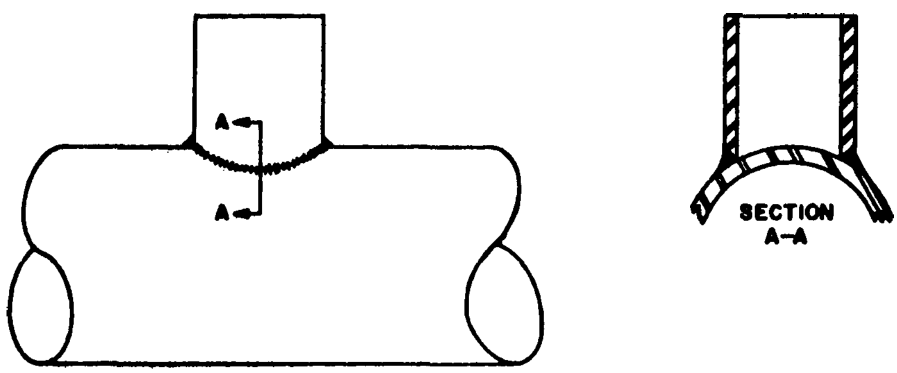


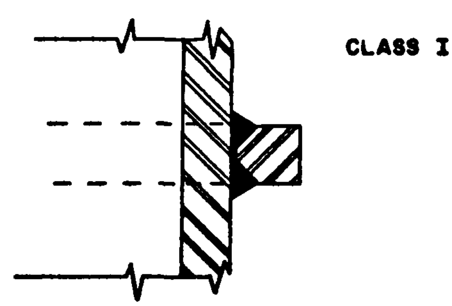

FIGURE 15.1 TYPICAL SUPPORT ATTACHMENTS TO PIPE AND ELBOWS




ATTACHMENT OF LUGS
SHOES, PIPE SADDLES, & BRACKETS




ATTACHMENT OF TRUNNIONS




ATTACHMENT OF RINGS

FIGURE 15.2 TYPICAL SUPPORT ATTACHMENTS TO PIPE, TAKEN FROM USAS B31.7

or test data for connections to such components has not been located by the author and probably very little such data exists. Analysis of such attachments, except perhaps using finite-element computer programs, will necessarily involve crude (but preferably conservative) assumptions.

# 15. REFERENCES


15.1 Petroleum Refinery Piping, USAS B31.3-1966, Published by the ASME, 345 E. 47th Street, New York, N. Y. 10017.

15.2 Design of Piping Systems, 1956, The M. W. Kellogg Company, Published by John Wiley & Sons, New York, N. Y.

15.3 Piping Handbook, 1967, Published by McGraw-Hill Book Company, New York, N. Y.

15.4 American Standard Code for Pressure Piping, Sections 1 (Power Piping), 2 (Industrial Fuel Gas Piping), 3 (Petroleum Refinery Piping), 4 (Oil Transportation Piping), 5 (Refrigeration Piping), 6 (Chemical Industry Process Piping), 7 (Nuclear Power Piping), and 8 (Gas Transmission and Distribution Piping). Published by ASME, 345 E. 47th Street, New York, N. Y. 10017.

15.5 Pipe Hangers and Supports, MSS SP-58(1963), published by Manufacturers Standardization Society, 420 Lexington Avenue, New York 17, N. Y.

15.6 Zallea Expansion Joints, Catalog 56, Zallea Brothers, Willmington, Delaware.

15.7 Tube-Turn Bellows Expansion Joints, Tube Turns, Louisville, Kentucky 40201.

15.8 Hannah, M. J., "Packed Slip or Packless Bellows Expansion Joints", Heating, Piping and Air Conditioning, May, 1968.

CHAPTER 16

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# 16. THERMAL STRESSES IN PIPING COMPONENTS

## 16.1 Theory

The theoretical analysis of stresses due to thermal gradients in piping components can be divided into two steps:

1) Calculation of the temperature distribution as a function of location on the component surface and through the wall thickness of the component.

2) Calculation of the stresses due to the temperature distribution found in step 1).

Step 1 is dependent upon a heat transfer analysis involving conduction, convection, radiation, and heat storage. Step 2 involves a stress analysis by methods basically the same as that used for other loadings such as internal pressure. While the two steps can be combined in some relatively simple cases, it seems desirable to discuss them separately herein.

## 16.11 Calculated Temperature Distributions

Calculation of temperature distributions in the walls of fluid-containing structures is a subdivision of the general field of heat transfer. The theory of heat transfer is discussed in numerous texts; e.g., McAdams,[16.1] Jakob,[16.2] and Schneider.[16.3] The present brief discussion gives some pertinent nomenclature and correlation parameters and some simple illustrations of their application. As in other fields, computer programs are coming into widespread use in the calculation of temperature distributions. Accordingly, a discussion of some known available programs is included.

## 16.111 Steady-State Radial Temperature Gradient

Many of the correlation parameters involved in heat transfer as applied to piping components can be illustrated by the simple case of radial flow of heat through a composite cylindrical structure as illustrated in Figure 16.1.

The successive temperature differences are:*

- across the inside convection film:

$$T_a - T_b = \frac{q}{2\pi}\left[\frac{1}{h_1 R_1}\right] \tag{16.1}$$

- across material 2:

$$T_b - T_c = \frac{q}{2\pi}\left[\frac{\ln (R_2/R_1)}{k_2}\right] \tag{16.2}$$

- across the interface between materials 2 and 4:

$$T_c - T_d = \frac{q}{2\pi}\left[\frac{1}{h_{c_3} R_2}\right] \tag{16.3}$$

- across material 4:

$$T_d - T_e = \frac{q}{2\pi}\left[\frac{\ln (R_4/R_2)}{k_4}\right] \tag{16.4}$$

- across the outside convection film:

$$T_e - T_f = \frac{q}{2\pi}\left[\frac{1}{h_5 R_4}\right] \tag{16.5}$$

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* Definitions given on pages 16.5 - 16.7.

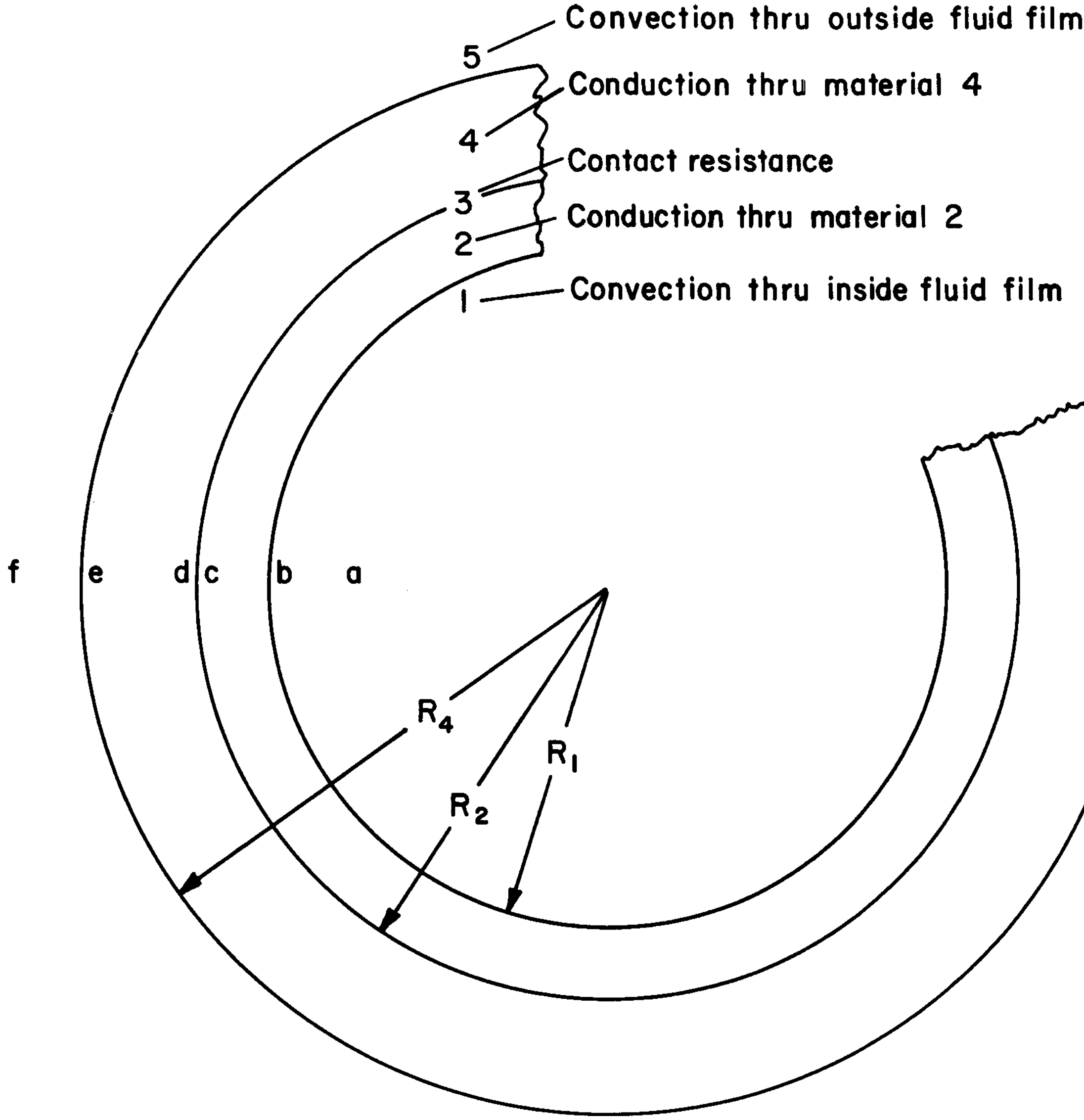



FIGURE 16.1. ILLUSTRATION OF HEAT TRANSFER PROBLEM IN STEADY-STATE RADIAL HEAT FLOW.

Since the total temperature drop is

$$T_a - T_f = (T_a - T_b) + (T_b - T_c) + (T_c - T_d) + (T_d - T_e) + (T_e - T_f) \qquad (16.6)$$

the temperature drop across any element can be determined by proportion. For instance, the temperature drop across material 2 is:

$$T_c - T_b = \frac{(T_a - T_f)\,\frac{1}{k_2}\ln(R_2/R_1)}{\frac{1}{h_1R_1} + \frac{1}{k_2}\ln(R_2/R_1) + \frac{1}{h_{c_3}R_2} + \frac{1}{k_4}\ln(R_4/R_2) + \frac{1}{h_5R_4}} \qquad (16.7)$$

and the exact distribution within material 2 would be:

$$T - T_b = (T_c - T_b)\,\frac{\ln(R/R_1)}{\ln(R_2/R_1)} \qquad (16.8)$$

Although, where $\frac{R_2 - R_1}{R}$ is small, a linear temperature distribution can be assumed as a reasonable first approximation; so:

$$T - T_b = \left(\frac{R - R_1}{R_2 - R_1}\right)(T_c - T_b) \qquad (16.8a)$$

For thermal stress analysis, the temperature distribution in the wall of the pressure retaining structure is of interest. In Figure 16.1, for example, if material 2 is a steel pipe, and material 4 is insulation, the thermal stress analysis requires the temperatures as given by equations 16.7 and 16.8. The other temperatures are not significant in the thermal stress analysis but are involved in the heat transfer analysis.

## Definition of Variables

$T_a$, $T_b$, .... These are the temperatures of any point designated by the subscript.

$R_1$, $R_2$, .... These are the radii at points designated by subscript.

$q$ = heat flow

$h_1$ = convection film on inside of the pipe. These are generally determined from a correlation by Nusselt Number (Nu), Reynolds number (Re), and Prandtl number (Pr) of the form

$$Nu = C_1 \, (Re)^a \, (Pr)^b \qquad (16.9)^*$$

where

$$Nu = \frac{hD}{k} \qquad (16.10)$$

$$Re = \frac{DV\rho}{\mu} \qquad (16.11)$$

$$Pr = \frac{c\mu}{k} \qquad (16.12)$$

$h$ = film coefficient, $h_1$

$D$ = significant dimension, = 2 $R_1$

$k$ = thermal conductivity of fluid flowing inside pipe

$\rho$ = density of fluid

$\mu$ = absolute viscosity of fluid

$c$ = specific heat of fluid

$C_1$ = coefficient of correlation = 0.23 for most cases*

$a$ = correlation exponent = 0.8 for most cases*

$b$ = correlation exponent = 0.4 for most cases*

---

*Reference to a standard work such as Reference 16.1 is recommended. For fluids with very low Prandtl numbers, such as liquid metals, a correlation of the form: $Nu = C_1 + C_2 \, (Pe)^a$, where $Pe = Re \times Pr$, may be better.

$k_2$, $k_4$ are the thermal conductivities of materials 2 and 4, respectively, estimated at the average temperature for the materials. If this normal assumption of constant thermal conductivity at the mean temperature cannot be made and a temperature sensitivity must be considered and if the form is:

$$k_2 = k_0 + \alpha_1 T + \alpha_2 T^2, \tag{16.13}$$

equation (16.2) would become:

$$(T_b - T_c) = \frac{q}{2\pi} \frac{\ln(R_2/R_1)}{\underbrace{\left[k_o + \alpha_1\left(\frac{T_b+T_c}{2}\right)\right]}_{\text{(mean conductivity)}} + \underbrace{\frac{\alpha_2}{3}\frac{T_b^3 - T_c^3}{T_b - T_c}}_{\text{(correction)}}} \tag{16.2a}$$

where the last term in the denominator is the correction introduced by not assuming a constant thermal conductivity at the mean temperature. Equation (16.2a) is not explicit in temperature, therefore iterations to find an adequately accurate mean temperature could be necessary.

$h_{c_3}$ is the contact conductance at the interface between materials. This induces a temperature discontinuity at the interface since effective contact can be across a significantly reduced area (unless bonding agents are used) because of the microsurface form. Some typical values under moderate loads are listed as follows:(16.4)

| Materials | $h_{c_3}$, Btu/hr·ft$^2$·°F |
|---|---|
| ceramic/metal | 250 - 1500 |
| steel/steel | 300 - 1500 |
| aluminum/aluminum | 500 - 5000 |
| metal/metal with a soft metal foil or grease joint filler | 5000 - 15,000 |

The contact resistance may change if thermal expansion significantly modifies the contact pressure at the interface and the stress calculation and the temperature calculations become coupled.

$h_5$ is the outside film coefficient. If forced convection occurs, a form of equation (16.9) would apply. If free convection exists the correlation is between Nusselt number (Nu) and the Rayleigh number (Ra).*

$$Nu = C_2 \, (Ra)^d \tag{16.9a}$$

$Ra = \dfrac{g\, L^3 \beta \Delta T\, \rho^2 c}{\mu\, k}$ and data are often tabulated so that

$Ra = a\, L^3\, \Delta T$

where $a = \dfrac{2\beta\, \rho^2 c}{\mu\, k}$

$g$ = gravitational constant

$\beta$ = the expansion coefficient for fluid

$C_2$ and d are the correlation coefficients generally treated as two sets:

| $a\,L^3\Delta T$ | $C_2$** | d** |
|---|---|---|
| $10^3$-$10^9$ | 0.45 | 1/4 |
| $> 10^9$ | 0.11 | 1/3 |

* Rayleigh number (Ra) is equivalent to Grashof number times Prandtl number.

** These values are for long horizontal cylinders, reference to a standard text such as Reference (16.1) is recommended.

Radiation

If radiation from the outside surface is significant, equation (16.5) should be modified:

$$T_e - T_f = \frac{q}{2\pi}\left[\frac{1}{R_r(h_5 + h_r)}\right] \qquad (16.5a)$$

where: $h_r$ = radiation-heat-transfer coefficient,

$$h_r = [(T_e + 460)^2 + (T_f + 460)^2]\,[T_e + F_f + 920]\,\sigma\, F_A F_E \qquad (16.14)$$

$\sigma$ = Boltsman constant = $0.173 \times 10^{-8}$

$F_A$ = an area factor*, generally = 1 for a small body relative to its enclosure

$F_E$ = an emissivity factor*, generally = $\epsilon_1$ for a small completely enclosed body

$\epsilon_1$ = emissivity of the outer surface of the inner body.

16.112 Steady-State Axial Gradient

A long pipe in which the fluid experiences a temperature drop as it flows is an extension of the previous case that can be readily handled within the following general limitations:

- the axial temperature gradient in the fluid is small enough so that axial conduction in the pipe walls can be ignored.
- the film coefficients can be assumed essentially constant over the pipe length
- the entering and leaving temperatures of the fluid are known
- the ambient surrounding temperature, $T_f$, is constant.

---

* Reference to a standard text such as Reference (16.1) is recommended.

The temperature of an element at any point along the length of the pipe can be determined by the following equation:

$$T_x - T_f = (T_{x'} - T_f) \exp (m/L)(\ln\frac{T_{x''} - T_f}{T_{x'} - T_f}) \qquad (16.15)$$

where

$T_x$ = Temperature of an element at a point along the pipe length.

$T_{x'}$ = Temperature of the element at the cooler end of the pipe.

$T_{x''}$ = Temperature of the element at the hot end of the pipe.

$T_f$ = Fluid temperature.

$(m/L)$ = Location of the point under consideration expressed as a fraction of the total pipe length.

End temperatures ($T_{x'}$ and $T_{x''}$) can be determined by proportion in a manner similar to that presented in example equation 16.7.

### 16.113 More General Steady-State Cases

The solution of the previous problem for more general situations would require a more complex calculation for which a computer program would be appropriate (see Par. 6.115). These programs are generally based on finite difference solution of the Fourier conduction equation

$$0 = \frac{k}{\rho c}\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) \qquad (16.16)$$

with internal boundaries defining heat flows across contact resistances and other boundaries considering convective and radiation heat flows. This procedure involves subdividing the structure into small elements

Several entire books are devoted to the solution of the heat conduction equation. Schneider (16.7) gives temperature response charts in which both exact and approximate solutions are presented in non-dimensional form. Time is expressed by the dimensionless group called the Fourier number

$$Fo = \frac{\alpha\theta}{R^2} \tag{16.18}$$

where $\alpha$ = thermal diffusivity of the solid = $\frac{k}{\rho c}$

$\theta$ = time

R = significant dimension.

The convection boundary condition is expressed by the Biot number

$$Bi = \frac{hR}{k} \tag{16.19}$$

where h = convective heat transfer coefficient

R = significant dimension

k = thermal conductivity of the solid.

Although this number is the same form as the Nusselt number, it is different since conductivity is that of the solid boundary rather than that of the flowing fluid. The charts all represent one-dimensional solutions for the constant property case. For the cylindrical shell the following solutions are presented, with all solutions having a constant temperature initial condition except where noted:

I. Constant inside wall temperature. Step change is outside wall temperature.

II. Insulated at inside surface. Sudden exposure to a constant temperature convective environment at outside surface.

III. Sudden exposure to a constant temperature convective environment at inside surface. Insulated at outside surface.

If the curvature effect can be neglected so that the problem reduces to a finite plate the following additional cases are presented. (In this case either surface A or B can be considered the inside or outside.)

IV. (A) Insulated surface.

(B) Surface temperature varying linearly with time.

V. (A) Constant surface temperature.

(B) Sudden exposure to a constant temperature convective environment.

VI. (A) Sudden exposure to a constant temperature convective environment.

(B) Surface temperature varying linearly with time.

VII. (A) Insulated surface.

(B) Sudden exposure to a heat input increasing or decreasing linearly with time.

VIII. Using a solution to VII as the initial conditions. Both surface insulated.

IX. (A) Insulated surface.

(B) Heat input varying as a cosine pulse with time.

X. (A) Insulated surface.

(B) Sudden exposure to a constant temperature radiation heat sink.

For a two-layer finite plate (which could approximate a pipe wall with insulation where curvature can be ignored) the following pertinent case is presented.

XI. (A) Insulated surface.

(B) Sudden exposure to a constant convective environment.

In many of these cases, results are given at various locations within the body although some only present the surface temperature response with time.

When the material properties are assumed constant, the heat conduction equation is linear and, therefore, various solutions can be added to obtain a solution to rather complicated problems by judicious combination of simpler solutions. This addition of solutions is also the basis for a very powerful tool in heat condiction, the infinite series solutions. Carslaw and Jaeger[16.8] outlined this method and others in great detail. In most cases, however, they present the form of the series solution and the user must do the calculations and, even though many of the series can be truncated after a reasonable number of terms, this can be a very tedious undertaking.

Another powerful tool in solving heat conduction problems is Duhamel's theorem discussed in Carslaw and Jaeger,[16.8] Schneider,[16.3] and Eckert and Drake.[16.9] By applying this technique the solution for transient boundary conditions can be found if the solution for the particular body for a constant boundary condition of the same type is available. Even if the known solution is in tabular form the more general solution can be found by graphical methods.

## <u>16.115 Computer Programs</u>

A very common method in solving heat flow problem today is numerical analysis. Originally this was done by hand calculation or even graphical techniques but, with the advent of modern computers, is now done almost exclusively by machine. The general procedure involves the finite difference solution of the heat conduction equations as has been discussed previously. This method is also discussed by Dusinberre.[16.10] It is possible to actually program this technique for every specific problem in heat flow that arises. However, in many cases it is quite advantageous to benefit from previous work and experience and adapt the problem to an existing more general program.

A number of computer programs have been developed to handle a wide variety of heat transfer problems by finite difference techniques. Some of these programs are quite general and can handle one-, two-, and even three-dimensional conduction together with convection and/or radiation at the boundaries for either the steady state or transient case. In some cases radiation between internal nodes as well as internal convection passages can also be handled.

These programs have been developed by various companies mainly for their own use but in most cases government funds have been used in the development. Most of these companies, therefore, will contract to run a particular problem using their program on their own computer or supply consulting assistance to implement their program for use by someone else. It should be recognized that a considerable internal investment will probably be required before one becomes proficient in the use of these complex programs. In fact, the use of these programs remains basically an art particularly in such areas as the choice of nodal network.

Some of the available general programs (in Fortran IV) are Jet Propulsion Laboratories' TAS[16.11], General Electric Company's THT[16.12], TIGER II[16.13], Chrysler Space Division's CINDA[16.14], Union Carbide's TOSS[16.15], and Applied Physics Laboratories' NETHAN (Network Thermal Analyzer). Each program has its own advantages and limitations or drawbacks. Therefore, some laboratories have a number of these programs available so that the best one can be chosen for a particular problem.

The programs listed above are quite general and can handle nearly any geometry with limitations only as to the maximum number of nodes. The input to these general programs can become extensive and complicated however so that in some cases a more specialized program may be desirable. One example of a more specialized program is HECTIC II (a Fortran Computer Program for Heat Transfer Analysis of Gas or Liquid Cooled Reactor Passages) which was originated by Aerojet General and modified by Argonne-Idaho[16.16]. Another example of a specialization that can provide considerable simplification is limiting the geometry to axisymmetric shapes. This technique is employed in Battelle's TAC (Thermal Analysis Code in Fortran IV). Non-axisymmetric boundary conditions are allowed however so that three dimensional effects can be studied. A list of specialized computer programs for determining both steady state and transient temperature distributions is included under the category of engineering in compilations prepared by Nather and Sangren[16.17] and Roos and Sangren[16.18]. Some of these programs were originally written in machine language or earlier Fortran and may not have been updated to Fortran IV.

Analog methods have also been used to determine temperature distributions. The active or direct method uses another equivalent system to simulate the thermal system with one-to-one correspondence between parameters. The electrical analog system in which voltage represents temperature and current represents heat flow is most commonly used. Other systems that have been used include the elastic sheet system and the hydraulic system. The active analog system is most useful in cases in which a specific configuration will be studied extensively since considerable effort is required in setting up a given problem and obtaining the proper scaling between the variables of the two systems. Some of the digital computer programs are actually inputed on a thermal network basis so that they are actually quite similar to the active analog system. In addition a general digital computer program called MIMIC[16.19] is available which simulates the analog computer directly.

In the passive or indirect analog method the mathematical description of the problem is formulated and the computer components simply perform one or more mathematical operation. This has the theoretical advantage that integration in one variable (usually time) can be done directly rather than by a finite difference technique. Additional variables must still be represented by difference techniques; however, this can easily require a very large amount of equipment. Thus, this method is practically limited to rather simple problems especially when compared with current finite differencing techniques on present day digital computers. For an extensive discussion of analog techniques the book by Schneider[16.3] is recommended.

## 16.12 Theory of Elastic Thermal Stresses

The theory of elastic thermal stresses is basically a part of the theory of elasticity and constitutes a specialized part thereof only in that strains $\alpha T$ are specifically considered ($\alpha$ = coefficient of thermal expansion, T = temperature). Thermal stresses are treated in several specialized texts, among which are those by Gatewood [16.20], Boley and Weimer [16.21], Nowacki [16.22], Benham and Hoyle [16.23] and Zudans, Yen and Steigelman [16.24].

As applied to piping components, recent advances in calculating thermal stresses are contained in computer programs. Most of the computer programs discussed in Chapter 3 (Par. 3.11) include temperature gradients as a loading condition and can be used to calculate thermal stresses due to such temperature gradients. Accordingly, for axisymmetric components with axisymmetric temperature gradients, calculation of thermal stresses is relatively routine using either shell or finite element type programs. For shells-of-revolution, at least one computer program [16.25] and probably several others can handle a non-symmetric temperature distribution, provided that the distribution can be described by a Fourier series; and practically a Fourier series of a reasonable number of terms.

For general structures, the several finite element programs now being developed (See Chapter 3, Par. 3.12) will presumably include temperature gradients as a loading condition. Until such time as these programs are developed for practical application to piping components, approximations must be selected with engineering judgement; hopefully these approximations will be conservative.

One non-symmetric structure of general interest consists of branch connections or nozzles in cylindrical shells. A computer program exists for such a structure with internal pressure loading [16.26]. It would seem that this computer program could be modified relatively easily to cover the thermal gradient case in which the nozzle is at some uniform temperature $T_n$ while the cylindrical shell is at some uniform temperature, $T_c$. This would then afford some guidance for design of such branch connections or nozzles for temperature gradients of this type.

There are useful compilations of equations for thermal stresses in simple structures subjected to specified temperature distributions. Some examples are (a) a semi-infinite body subjected to a surface temperature change; (b) a thin-wall tube subjected to a radial (thru-the-wall) temperature gradient, (c) a thick-wall tube subject to an arbitrary radial temperature. Such equations are useful in establishing bounds on thermal stresses. Compilations of this type are given by Goodier [16.27] and Roark [16.28].

## 16.13 USAS B31.7 Thermal Stresses

Equations (10) and (11) of USAS B31.7 [16.29] are shown herein as Table 16.1. The third and fourth terms of equation (10) and the third, fourth and fifth terms of equation (11) give thermal stresses.

The third term in Equation (10) of B31.7 is taken from the equation for hoop and axial stress at the surface of a thin wall pipe with a radial (thru-the-wall), linear thermal gradient of $\Delta T$. If the outside surface is hotter than the inside surface, then the outside surface stresses are tensile; the stresses vary linearily thru the wall thickness to equal magnitude compressive stresses on the inside surface. For thick wall pipe

TABLE 16.1: EQUATIONS FROM USAS B31.7
(1 of 3)

Equation (10) of B31.7:

$$S_n = C_1 \frac{PD_o}{2t} + C_2 \frac{D_o}{2I} M_i + \underbrace{\frac{E\alpha|\Delta T_1|}{2(1-\nu)} + C_3 E_{ab}|\alpha_a T_a - \alpha_b T_b|}_{\text{Thermal stress terms}}$$

For the thermal stress terms:

$E\alpha$ = modulus of elasticity (E) times the mean coefficient of thermal expansion ($\alpha$), psi/F

$|\Delta T_1|$ = absolute value of the temperature difference between the temperature of the outside surface ($T_o$) and the temperature of the inside surface ($T_i$) of the component assuming moment-generating equivalent linear temperature distribution. See figure at bottom of Sheet 2 of this table.

$\nu$ = Poisson's ratio = 0.3

$E_{ab}$ = the average modulus of elasticity of the two parts of the gross discontinuity

$\alpha_a$ = mean coefficient of expansion on side "a" of a gross discontinuity such as a branch-to-run or flange-to-pipe or socket fitting-to-pipe gross discontinuity

$T_a$ = average temperature minus the room temperature on side "a" of a gross discontinuity

$\alpha_b$ = mean coefficient of expansion on side "b" of a gross discontinuity

$T_b$ = average temperature minus the room temperature one side "b" of a gross discontinuity

$C_3$ = secondary stress index, see Sheet 3 of this table.

continued on next page

TABLE 16.1 EQUATIONS FROM USAS B31.7, (2 OF 3)

Equation (11) of B31.7

$$S_p = K_1 C_1 \frac{PD_o}{2t} + K_2 C_2 \frac{D_o}{2I} M_i +$$

$$\underbrace{\frac{K_3 E \alpha \left|\Delta T_1\right|}{2(1-\nu)} + K_3 C_3 E_{ab} \left|\alpha_a T_a - \alpha_b T_b\right| + \frac{E \alpha \left|\Delta T_2\right|}{1-\nu}}_{\text{Thermal stress terms}}$$

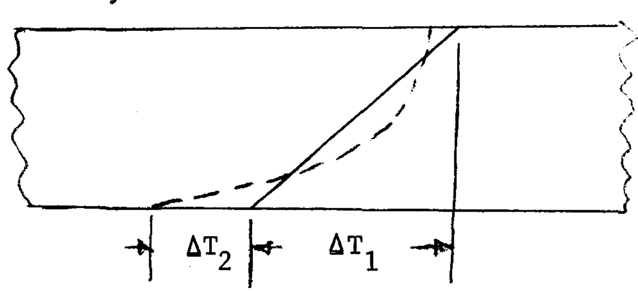
where, in addition to the definitions under Equation (10):

$K_3$ = local stress index, see Sheet 3 of this Table

$\Delta T_2$ = absolute value for that portion of the nonlinear thermal gradient through the wall thickness not included in $\Delta T_1$, of Equation 10, as shown below.

--- Actual

—— Equivalent Linear




( $\Delta T_2$ is produced by a rapid change in fluid temperature.)

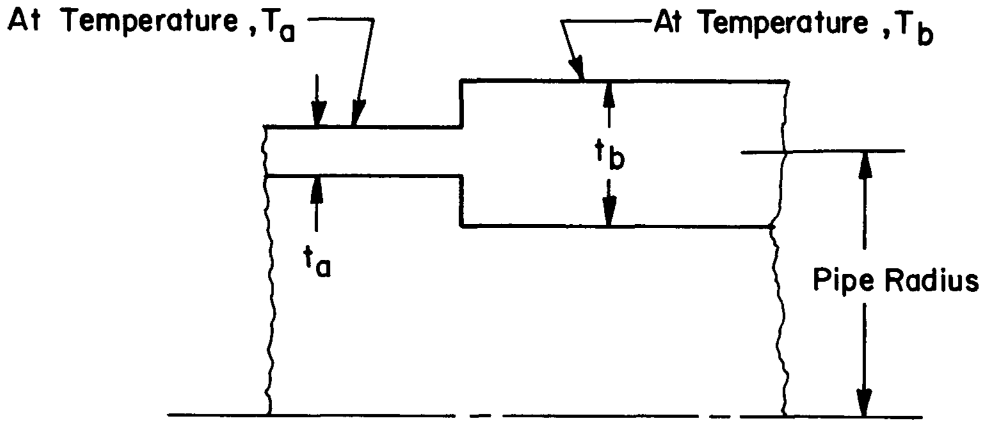
TABLE 16.1: EQUATIONS FROM USAS B31.7
(3 of 3)

| Component | Stress Indices for Thermal Loading* | |
|---|---|---|
| | $C_3$ | $K_3$ |
| Straight pipe, remote from welds or other discontinuities | 1.0 | 1.0 |
| Girth butt weld between straight pipe or between straight pipe and butt-welding components | | |
| (a) flush | 1.0 | 1.1 |
| (b) as welded | 1.0 | 1.7 |
| Girth fillet weld to socket weld fittings, slip-on flanges or socket-welding flanges | 1.8 | 3.0 |
| Longitudinal butt welds in straight pipe | | |
| (a) flush | 1.0 | 1.1 |
| (b) as welded | 1.0 | 1.2 |
| Tapered transition joints | 1.0 | 1.5 |
| Branch connections | 1.8 | 1.7 |
| Curved pipe or welding elbows | 1.0 | 1.0 |
| Butt welding tees | 1.0 | 1.0 |
| Butt welding reducers | 1.0 | 1.0 |

* From USAS B31.7. appendix D.

this term is slightly unconservative; for example, for a diameter-to-thickness ratio (D/T) of 12, the maximum thermal stress is 3 percent higher than given by the term. Because in nuclear piping a D/T less than 12 is seldom used, this slight unconservatism was considered acceptable.

The fourth term in Equation (10) of B 31.7 represents thermal stresses due to an axial discontinuity in structure and temperature, such as may occur between a pipe and a socket-welded fitting. The $C_3$ factor of 1.8 shown in Appendix D of B 31.7 is based on the assumption that the fitting is rigid. The relative displacement between the (thin-wall) pipe and the fitting is then given by $r(\alpha_a T_a - \alpha_b T_b)$, where r = pipe radius. The separate values of $\alpha_a$ and $\alpha_b$ provide for the case where the pipe is made of a material with a coefficient of thermal expansion different than that of the material used for the fitting. An average value of the modulus-of-elasticity ($E_{ab}$) is used and properly this should be averaged over the temperatures involved; however, for the materials and temperatures covered by B 31.7, this is a relatively minor consideration. The $C_3$ factor of 1.8 gives the maximum bending stress, which is in the axial direction at the pipe-rigid structure juncture. Strictly speaking, the stress intensity is some 25% higher; however, the rigid-structure assumption is very conservative for typical fittings. If, for example, the fitting is effectively 4 times as thick as the pipe, then the maximum bending stress is only 65% of that indicated by a $C_3$ factor of 1.8. A graph showing how this stress varies as a function of the thickness ratio is shown in Figure 16.2. For fabricated branch connections (diameter ratio less than one-half), a $C_3$ factor of 1.8 is also used. For B 16.9 or similar tees,




$\sigma_{max}$ = $C_3$ E ($\alpha_a T_a$ - $\alpha_b T_b$)

E = modulus of elasticity (assumed to be same for both thin & thick wall pipes)

$\alpha_a$ = coeff. of thermal expansion, thin pipe

$\alpha_b$ = coeff. of thermal expansion, thick pipe

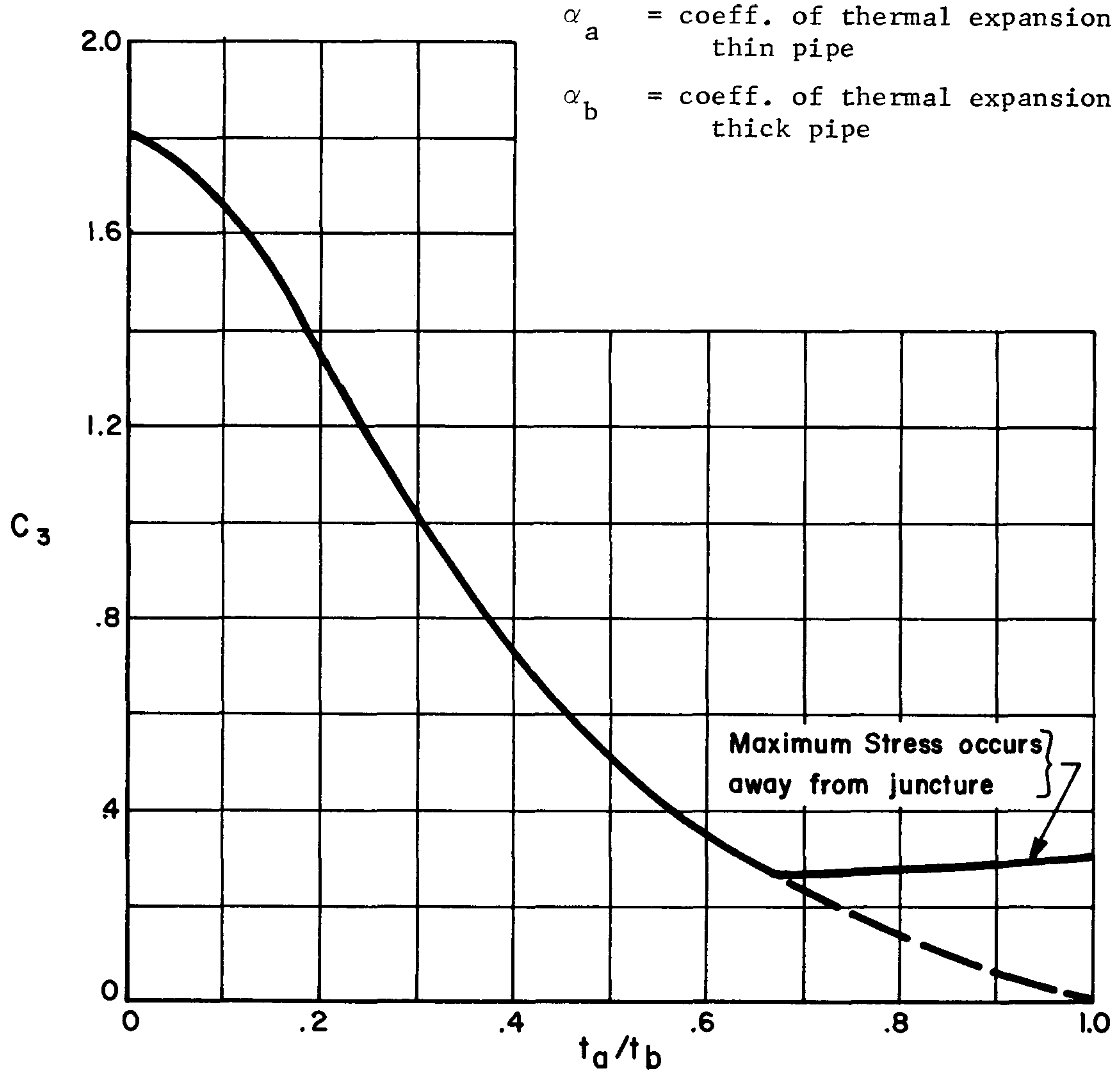



FIGURE 16.2 AXIAL BENDING STRESS AT A DISCONTINUITY IN WALL THICKNESS AND TEMPERATURE

a $C_3$ factor of 1.0 is used. This is equivalent to assuming a discontinuity thickness ratio of about three. For branch connections and tees, $T_a$ is to be considered as the temperature of the branch pipe, $T_b$ the temperature of the run pipe.

The third and fourth terms of Equation (11) are the same as those of Equation (10), except for the $K_3$ factor. Equation (11) is used to indicate the magnitude of peak stresses for fatigue evaluation. As discussed in the subsequent section on "Test Data", there is very little quantitative information of the fatigue strength of piping components with cyclic thermal loading. The same $K_3$ factor is used for both the radial thermal gradient (third term) and axial discontinuity gradient. The $K_3$ factors shown for various components are generally similar in magnitude to $K_2$ factors for moment loadings. The $K_3$ factors are believed to be conservative and may be ultra conservative. Some test data in this area are highly desirable.

The fifth term in Equation (11) of B 31.7 represents that part of the thru-the-wall thermal gradient which is in excess of the linear-equivalent gradient. This excess gradient is considered as a surface temperature change. If $\Delta T_2$ is positive (the surface of the pipe is hotter than the remainder of the pipe wall), then that surface is subjected to a biaxial compression stress as given by the fifth term. It could be contended that this term is also subject to a local stress factor if, for example, the stress occurred at a weld. Other conservative aspects appear to compensate for this omission.

## 16.2 Test Data

### 16.21 Measured Thermal Stresses

In many significant cases of elastic thermal stresses, the stresses arise due to a suppression of the "free" thermal expansion. Accordingly, measurement of surface strains (in analogy to strain gage tests for mechanical loadings) is not always informative. However, in the past few years progress has been made in developing and applying techniques for measuring thermal stresses. References (16.30) through (16.37) are a few examples of papers on such techniques and results obtained thereby.

Two aspects of design involving thermal stresses are:

(1) Progressive distortion due to (usually) a combination of mechanical loads and thermal stresses.

(2) Fatigue failure caused by cyclic thermal stress.

Neither of these problems is strictly one of elastic thermal stress and both can be complicated by creep or relaxation. However, in the present state of the art, thermal stresses calculated on an elastic basis are used for design purposes.

### 16.22 Progressive Distortion or Racheting

Miller[16.38] cites two references, (16.39) and (16.40) herein, in which observations of progressive distortion of pressure vessels subjected to repeated thermal stresses were reported. Miller[16.41] has compiled a bibliography on ratcheting which is included herein as References (16.42) thru (16.71). These references cover the theory as well as, in some cases, test data. It should be remarked that ratcheting is not restricted to combinations of cyclic thermal stress with a mechanical load mean stress but may also arise with any strain-controlled cyclic strain in the presence of a mean load stress. Observation of ratcheting or progressive incremental straining occurs quite often in the literature in conjunction with fatigue tests which involve a mean load plus a cyclic strain; e.g., References (16.53) and (16.69). Most of the cited test data is concerned with test coupons or hollow cylinders. A notable exception is the paper by Weil and Rapasky[16.39]; who described service experience on observed incremental growth in cyclindrical shells, flanged manholes and conical heads of pressure vessels used for delayed-coking units in a petroleum refinery. Edmunds and Beer[16.49] give data on incremental deformation of elbows with internal pressure and in-plane bending deflection. This is an example of ratcheting without thermal stress.

## 16.23 Fatigue Failure - Cyclic Thermal Strains

Test data on fatigue due to cyclic thermal strains are relatively scarce. The data known to the writer are almost entirely limited to low cycles; i.e., up to about $10^5$ cycles. One of the earliest investigations is reported by Coffin[16.72]. This paper covers an extensive series of tests on 347 materials at cyclic temperatures between 200 and 500 C with hold times from about 8 to 200 seconds. Subsequently, many additional papers have been published. The book by Manson[16.73] contains an extensive discussion and numerous references on the subject. Coffin[16.74] gives a brief resume of the status of high-temperature, low-cycle fatigue while Benham[16.75] gives a survey of current work in Britain.

The above mentioned references are essentially limited to "coupon" tests of the material. There is a considerable jump between these tests and the fatigue behavior of piping components as actually fabricated into a piping system. There are at least a few references which give some indications of component response to cyclic thermal stresses. Stewart and Schreitz[16.76] give results on thermal shock tests on 6" Sch 80 and Sch 160 pipe and valves and welds therein. Both ferretic and austenitic materials were used. Testing consisted of heating the piping section with steam flow at 1050 F, followed by water flow at 500 to 600 F. From 100 to 125 cycles were applied to each of the four test assemblies (two schedules X two materials). Examination of the assemblies and sections cut therefrom after test indicated no significant damage due to the 100 to 125 thermal stress cycles. There were some indications that small surface cracks in the welds may have been caused (or, at least opened-up) by the thermal cycles.

Weisberg and Soldan[16.77] give results on tests on pipe and welds therein. Tests were run on 12" X 2.25" wall pipe made of ferritic or austenitic material. Thermal cycles were applied by flowing steam at 1100 F, followed by a water flow (water at $\sim$ 600 F) and subsequent cooling to about 150 F. A total of 100 cycles of thermal stresses were applied.

The assemblies were then inspected for signs of damage. No cracking was found in any of the test pieces which could be attributed to thermal cycling.

Tidball and Shrut[16.78] give results on tests of austenitic steel pipe and welds therein. The pipe was 8" Sch 40; some of the welds were made with backing rings. Tests consisted of flowing sodium at 850 F thru the specimen, followed by flowing sodium at 580 F. 2500 cycles were applied at a rate of 4 cycles per hour. Metallographic examination of the unshocked duplicate test specimen indicated that failure to remove the backing ring after welding had permitted cracks several microinches in length to remain in the root pass. Inspection of the test section after 2500 shocks revealed that these cracks had increased to twice the original size (based on examination of the unshocked specimen). However, no evidence was present which showed that any new cracks had been formed during the thermal-shock cycling. On similar test specimens, the initial root-pass cracks were eliminated by machining out the backing rings. Inspection of the shocked test piece again indicated that no new cracks were formed. Each test specimen also was checked for possible distortion due to thermal shocking. Measurements for the outer diameter of the 8" test section were made before and after testing using a pair of micrometer calipers. Although these measurements indicated possible distortion, the magnitude was so small that the results are not conclusive.

In contrast to the preceding three references, in which the results were mostly negative, Gysel, Werner and Gut[16.79] ran tests in which thermal fatigue cracks were obtained. The tests were run on hollow cylinders 6" long, 2" O.D. and 1/2" I.D. Test specimens were girth-butt-welded at the center of the length. Circumferential notches (grooves) were placed in the bore, including a groove in the root of the weld. Specimens were heated in a furnace to various temperatures from 450 to 500 C; the bore was then quenched by running water thru it. This was repeated 1000 times for each specimen. The specimens were then sectioned and examined for cracks. Eight different types of cast steels were tested, ranging from a plain carbon cast steel to a 17% Cr-4% Ni cast alloy steel. The tests were run to assign relative thermal shock resistance values to these eight kinds of cast steels. The tests were sufficient to produce cracks in all specimens. Except at the notches, the cracks were shallow; at the notches the cracks extended radially up to some 1/3 of the wall thickness. The welds responded about the same as the base metal to these tests.

Estimated maximum thermal stresses are given in References (16.76) and (16.78). Maximum thermal stresses in Reference (16.77) are estimated to be about 30,000 psi. For Reference (16.79), no flow rate of the cooling water is given. Assuming that the water produced a very rapid drop in bore temperature; the skin stresses would be given by

$$\sigma_{max} = \frac{E\alpha(T_h - T_w)}{1 - \nu} \tag{16.20}$$

where $E$ = modulus of elasticity

$\alpha$ = coefficient of thermal expansion

$\nu$ = Poisson's ratio

$T_h$ = hot (test) temperature

$T_w$ = water temperature

Assuming $E = 3 \times 10^7$, $\alpha = 6 \times 10^{-6}$, $\nu = 0.3$, $T_w = 70$ F

$$\sigma_{max} = 257 \ (T_h - 70) \ .$$

This value represents an upper bound to the thermal stress applied in the tests.

It is pertinent to compare the results of tests in References (16.76) thru (16.79) with the proposed* high-temperature code case of USAS B 31.7. Table 16.2 shows these comparisons. The design method indicates that no cracks would appear in tests of References (16.76), (16.77) and (16.78) and apparently there were none. The design method indicates cracks would occur in Reference (16.79) tests and they did. With respect to comparison with Reference (16.79), two questions arise. First, are the code case graphs supposed to be for crack initiations or for cracks thru the wall? Second, at what number of cycles (less than 1000) did cracks initiate in Reference (16.79) tests? The writer is unable to answer either of these questions.

* The S-N graphs of the proposed case are the same as those shown in ASME Boiler Code Case 1331-4.

TABLE 16.2: SOME COMPARISONS OF TEST DATA WITH USAS B 31.7 HIGH TEMPERATURE CODE CASE

| Reference Number | Test Maximum Temperature | Range of Thermal Stresses | Cycles Applied in Test | Design Cycles-to-Failure (b) | | Estimated Cycles-to-Failure (c) | |
|---|---|---|---|---|---|---|---|
| | | | | Ferritic | Austenitic | Ferritic | Austenitic |
| 16.76 | 1050 | 32,000 | 100/125 | 7000 | 35000 | $\sim 10^6$ | $>10^6$ |
| 16.77 | 1100 | ~30,000 | 100 | 6000 | 35000 | $\sim 10^6$ | $>10^6$ |
| 16.78 | 850 | 92,000 | 2500 | -- | 7000 | -- | 800,000 |
| 16.79 | 842 | 200,000(a) | 1000 | 90 | -- | 900 | -- |
| | 932 | 220,000 | | 20 | -- | 250 | -- |
| | 1022 | 240,000 | | <10 | -- | 80 | -- |
| | 1112 | 270,000 | | <10 | -- | 30 | -- |

(a) Based on Equation (16.20).

(b) Obtained from ASME Code Case 1331-4 by entering S-N graphs with one-half of the indicated "Range of Thermal Stress" and reading off number of cycles. The curve used corresponded to the "Test Max Temp" indicated.

(c) As in (b), except entering with one-quarter of the indicated "Range of Thermal Stress". This estimate is based on the assumption that the design graphs have a factor of safety on stress of two.

## 16.24 Mechanical Strain vs Thermal Strain Fatigue

The 1955 ASA Code for Pressure Piping introduced the criterion of fatigue failure in piping systems under restrained cyclic thermal expansion. However, the stress intensification factors used were based on Markl's[16.84] test data from mechanically imposed displacements. It was realized that behavior under thermal cycling, i.e., the case where strains are induced by restrained thermal expansion, would probably not be entirely the same. An investigation was instituted to check the possible differences. The results of the investigation are given by Coffin[16.80] and a discussion by Markl of Coffin's paper. Markl showed that, under comparable conditions:

$SN^{0.2}$ = 367,000 mechanical cycling

$SN^{0.2}$ = 560,000 fully restrained thermal cycling.

The above equations are for Type 347 stainless steel. For mechanical cycling, the temperature was 1050 F. For thermal cycling, the temperature was varied from 212 to 1112 F.

Since Coffin's paper (1957), much additional work has been done on low-cycle high-temperature fatigue and correlations between mechanical cycling and thermal cycling. The significance of hold-time and, in addition, the exact characteristics of the cycle have become more appreciated. Carden, Vogel and Kyzer[16.81] present a good discussion of some types of cycles that can be applied. Carden and Sodergren give some recent data on correlations of thermal cycling with mechanical (iso-thermal) cycling for type 304 stainless steel.

## 16.3 Service Experience

Field failures are discussed in Chapter 4 ; however, there are some aspects of service experience of direct relevance to some of the theory on thermal stresses presented in the foregoing. Service failures ascribable to cyclic thermal stresses are not uncommon in piping systems. Thielsch(16.83) describes a number of such failures. In some cases, the severity of thermal stresses and their cycle frequency could have been predicted. However, in many if not most field failures, the prediction of cyclic thermal conditions would have been difficult. For example, several failures have been reported at small drain lines in high temperature steam lines. What apparently happens is that the small drain line partially fills with relatively cold condensate. Changes in flow rate and/or pressure in the main steam line then periodically draw this condensate back into the hot main steam line with resulting thermal stress fatigue cracks at the branch juncture. Desuperheaters are another component where, usually due to unanticipated flow conditions, thermal stress fatigue is common. A comparable condition may occur in a so-called mixing tee.* Here, for example, hot fluid comes in through the branch to mix with colder fluid flowing through the run. Under certain flow conditions the hot and cold fluid may intermingle in discrete layers. These layers then rotate so that the metal walls are subjected to rapid cycles of thermal stress due to alternate contact with hot and cold fluid. This is a subject on which little theoretical guidance is available for design purposes.

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* A failure of this type is discussed in Chapter 4.

# 16. REFERENCES


16.1 McAdams, W. H., Heat Transmission, Third Edition, McGraw-Hill, New York (1954).

16.2 Jakob, M., Heat Transfer, John Wiley and Sons, Inc., Vol 1, New York (1962).

16.3 Schneider, P. J., Conduction Heat Transfer, Addision-Wesley Publishing Co., Inc., Reading, Massachusetts (1957).

16.4 Minges, M. L., "Thermal Contact Resistance Volume I -- A Review of the Literature", Report from Air Force Materials Laboratory, Wright-Patterson Air Force Base, AFML-TR-65-375 (April 1966) AD No. 482633, p 7.

16.5 Csaba, J., Leggett, A. D., and Horn, G., "The Temperature Distribution in the Wall of a Tube with Nonuniform External Heating and Internal Cooling", International Journal of Heat and Mass Transfer, Vol 9, No. 4, April 1966, pp 325-336.

16.6 Brown, A. I., and Marco, S. M., Introduction to Heat Transfer, Third Edition, McGraw-Hill, New York (1958).

16.7 Schneider, P. J., Temperature Response Charts, John Wiley and Sons, Inc., New York (1963).

16.8 Carslaw, H. S., and Jaeger, J. C., Conduction of Heat in Solids, Second Edition, Oxford University Press, London (1959).

16.9 Eckert, E. R. G., and Drake, R. M., Heat and Mass Transfer, Second Edition, McGraw-Hill Book Co., Inc., New York (1959), pp 178-183.

16.10 Dusinberre, G. M., Numerical Analysis of Heat Flow, First Edition, McGraw-Hill, New York (1949).

16.11 TAS, See "Thermal Analyzer System I", by J. A. Hultberg, Space Propulsion Systems, Jet Propulsion Laboratory, Vol IV, January-February 1967.

16.12 THT (Transient Heat Transfer), See Paper by Campbell and Vollemweider, Proceedings of Eastern Joint Computer Conference, Vol 16, p 143 (1959).

16.13 TIGER-II, See "TIGER-II, an IBM-704 Digital Computer Program: "Temperatures from Internal Generation Rates" by A. P. Bray and S. J. MacCraken, KAPL-2044, May 29, 1959.

16.14 CINDA, See "Chrysler Improved Numerical Differencing Analyzer, Computer Program CO-0045" by J. D. Gaski, Technical Note TN-AP-66-15, Chrysler Corporation, Space Division, April 1966.

16.15 TOSS, See "TOSS: An IBM-7090 Code for Computing Transient or Steady-State Temperature Distributions" by D. Bagwell, AEC Report K-1494, Union Carbide Nuclear Co., December 1961.

16.16 HECTIC-II, See "Army Gas-Cooled Reactor Systems Program HECTIC-II - An IBM-7090 Fortran Computer Program for Heat Transfer Analysis of Gas or Liquid Cooled Reactor Passages" by N. Kattchee and W. C. Reynolds, AEC Report IDO-28595 Rev., December 1965.

16.17 Nather, V. and Sangren, W., "Codes for Reactor Computations", Nucleonics, Vol 19, No. 11, pp 154-158 (November 1961).

16.18 Roos, B. W. and Sangren, W., "Codes for Reactor Computations", Nucleonics, Vol 20, No. 8, pp 132-133 (August 1962).

16.19 MIMIC, See "Mimic Programming Manual" by F. J. Sansom and H. E. Petersen, Wright-Patterson Air Force Base, Ohio, AD-656301, July 1967.

16.20 Gatewood, B. E., Thermal Stresses, McGraw-Hill, 1957.

16.21 Boley, B. A. and Weimer, J. H., Theory of Thermal Stresses, John Wiley and Sons, 1960.

16.22 Nowacki, W., Thermoelasticity, Pergammon Press, 1962

16.23 Benham, P. P. and Hoyle, R., Thermal Stress, Sir Isaac Pitman and Sons, 1964.

16.24 Zudans, Z., Yen, T. C., and Steigelmann, W. H., Thermal Stress Techniques in the Nuclear Industry, American Elsevier Publishing Company, 1965.

16.25 MOLSA, See "Analysis of Shells of Revolution Subjected to Symmetrical and Nonsymmetrical Loads", by A. Kalnins, ASME Journal of Applied Mechanics, September 1964.

16.26 Eringen, Naghdi, Mahmood, Thiel and Ariman, "Stress Concentrations in Two Normally Intersecting Cylindrical Shells Subject to Internal Pressure", General Technology Corp., Technical Report No. 3-9, January 1967.

16.27 Goodier, J. N., "Thermal Stress and Deformation", Trans. ASME, Journal of Applied Mechanics, pp 467-474, September 1957.

16.28 Roark, R. J., "Formulas for Stress and Strain", McGraw-Hill, 1954.

16.29 USAS B 31.7, American Standard Code for Pressure Piping, Nuclear Power Piping, Issued February 1968, for Trial Use and Comment, Published by the American Society of Mechanical Engineers, 345 East 47th Street, New York, New York 10017.

16.30 Avery, L. R., Carayanis, G. S., and Michky, G. L., "Thermal-Fatigue Tests of Restrained Combuster-Cooling Tubes", Experimental Mechanics, June 1967, pp 256-264.

16.31 Frisch, J. and Morris, J. E., "Strain Measurements in Tubes During Rapid Transient Heating," Experimental Mechanics, August 1967, pp 353-358.

16.32 Sciammarella, C. A. and Sturgeon, D., "Thermal Stresses at High Temperature in Stainless Steel Rings by the Moire Method", Experimental Mechanics, May 1966, pp 235-243.

16.33 Emery, A. F., Barrett, C. F. and Kobayashi, A. S., "Temperature Distributions and Thermal Stresses in a Partially Filled Annulus", Experimental Mechanics, December 1966, pp 602-608.

16.34 Rothstein, R. J. and Kirkwood, W. F., "Photothermoelastic Analysis of Stresses in Multiconnected Flat Circular Rings", Experimental Mechanics, August 1964, pp 237-243.

16.35 Sciammarella, C. A. and Ross, B. E., "Thermal Stresses in Cylinders by the Moire Method", Experimental Mechanics, October 1964, pp 289-296.

16.36 Frisch, J. and Arne, C. L., "Optical Strain Determination at Transient High Temperature in Stainless Steel", Experimental Mechanics, November 1964, pp 320-328.

16.37 Leven, M. M. and Johnson, R. L., "Thermal Stresses on the Surface of Tube-Sheet Plates of 10 and 33-1/3 Percent Ligament Efficiency", Experimental Mechanics, December 1964, pp 356-365.

16.38 Miller, D. R., "Thermal-Stress Ratchet Mechanism in Pressure Vessels", Trans. ASME, Vol 81, Series D, pp 190-196, 1959.

16.39 Weil, N. A., and Rapasky, F. S., "Experience With Vessels of Delayed-Coking Units", Preprint of Papers Submitted to a Session on Pressure Vessels, American Petroleum Institute, Division of Refining, Los Angeles, California, May 13, 1958.

16.40 Coffin, L. F., "The Resistance of Materials to Cyclic Thermal Strains", ASME Paper No. 57-A-286.

16.41 Miller, D. R., Private Communication to E. C. Rodabaugh, September 9, 1968.

16.42 Swift, H. W., "Plastic Bending Under Tension", Engineering, Vol 166, pp 333-335 and 357-359, 1948.

16.43 Gatewood, B. E., "Thermal Stresses", McGraw-Hill, p 138, 1957.

16.44 Parkes, E. W., "A Design Philosophy for Repeated Thermal Loading", AGARD-213, 1958.

16.45 Gatewood, B. E., "The Problem of Strain Accumulation Under Thermal Cycling", Journal of Aerospace Sciences, Vol 27, No. 6, pp 461,2, 1960.

16.46 Bochvar, A. A., et al, "The Deformation of Uranium Under the Influence of Thermal Cycles During the Action at the Same Time of a Tension Load", (In Russian), Atomn. Energ., Vol 8, No. 2, 1960, pp 112-116; Ref. Zh. Mekh. No. 4, 1961, Rev. 4V489.

16.47 "Plastic Strain Induced by Thermal Cycling of Zirconium", NP-14268, September 1960 (AEC unclassified nuclear abstract No. 18-39905).

16.48 Coffin, L. F., Jr., "The Stability of Metals Under Cyclic Plastic Strain", Trans. ASME, Journal of Basic Engineering, pp 671-682, September 1960.

16.49 Edmunds, H. G. and Beer, F. J., "Notes on Incremental Collapse in Pressure Vessels", Journal Mechanical Engineering Science, Vol 3, No. 3, pp 187-199, 1961.

16.50 Gatewood, B. E., et al, "Experimental Data on Strain Accumulation Under Equivalent Thermal Cycling", Journal Aerospace Sciences, Vol 28, pp 502-3, 1961.

16.51 Tilly, G. P. and Benham, P. P., "Load Cycling in the Low Endurance Range in Relation to Brittle Fracture of Mild Steel", Journal of the Iron and Steel Institute, pp 216-223, March 1962 (specimens subjected to high levels of pulsating tension or compression deformed progressively).

16.52. Schwiebert, P. D. and Moyar, G. J., "An Application of Linear Hardening Plasticity Theory to Cycle and Path Dependent Strain Accumulation", T & AM Report No. 212, University of Illinois, 1962. See also Brief Note on above subject, Trans. ASME, Vol 30, Series E, No. 2, pp 296-298, 1963.

16.53 Moyar, G. J. and Sinclair, G. M., "Cyclic Strain Accumulation Under Complex Multiaxial Loading", AD 427919, RTD-TDR-63-4120, December, 1963. Paper of same title published in Proc. Joint International Conference on Creep, 1963, Institute of Mechanical Engineers, London, pp 2-47 to 2-57 incl.

16.54 Hoger, R., "Thermal Stresses in Cylindrical Structures and Allowable Speed of Temperature Changes" (In German), Allgemeine Warmetechnik, Vol 12, No. 1, pp 10-19, 1963.

16.55 Jenkins, G. M. and Williamson, G. K., "Deformation of Graphite by Thermal Cycling", Journal of Applied Physics, Vol 34, No. 9, pp 2837-2841, 1963.

16.56 Raymond, M. H. and Coffin, L. F., Jr., "Geometrical Effects in Strain Cycled Aluminum", Trans. ASME, Vol 85, Series D, p 548, 1963.

16.57 Taira, S. and Ohnami, M., "Fracture and Deformation of Metals Subjected to Thermal Cycling Combined With Mechanical Stress", Proceedings Joint International Conference on Creep, 1963, Institute of Mechanical Engineers, London, pp 3-57 to 3-62 incl.

16.58 Moyar, G. J. and Sinclair, G. M., "Cumulative Plastic Deformation in Rolling Contact", Transactions ASME, Vol 85, Series D, No. 1, pp 105-115, 1963.

16.59 Bree, J., "Elastic-Plastic Deformation of a Long Hollow Cylinder Under Thermal Cycling and Internal Pressure", TRG Report 790(D), 1964.

16.60 Oelschlagel, D. and Weiss, V., "Superplasticity of Steels During the Ferrite-Austenite Transformation", ASM Transactions Quarterly, June 1964. (Digested in Metal Progress, Vol 90, No. 5, p 148)

16.61 Burgreen, D., "Ratcheting Growth of an Element Subjected to Parabolic Thermal Cycling", Trans. ANS, Vol 7, No. 2, pp 436-7, 1964.

16.62 Coffin, L. F., Jr., "The Influence of Mean Stress on the Mechanical Hysteresis Loop Shift of 1100 Aluminum", Trans. ASME, Journal of Basic Engineering, pp 673-680, December 1964.

16.63 McConnelee, J. E., "Thermal Stress Ratchet Mechanisms", (General Electric Company, Nuclear Materials and Propulsion Operation, Cincinnati, Ohio, 45215) GE-TM 65-5-31, 1965.

16.64 Martin, W. R., "Mechanical Cladding-Fuel Interactions During Thermal Cycling of Metal Clad Fuel Elements", ORNL 3514, 1965. (See also ORNL-3619, pp 123-146 and ORNL-3670, pp 174-200 on same subject.)

16.65 Nichiporchik, S. N., "Determination of the Residual Strain Caused by Combined Cyclic Bending and Static Torsion", (translation), Ind. Lab., Vol 31, No. 3, pp 441-2, 1965.

16.66 Bree, J., "Ratchet and Fatigue Mechanisms in Sealed Fuel Pins for Nuclear Reactors", TRG Report 1214 (D), 1966.

16.67 Bree, J., "Ratchet and Enhanced Creep Strains in Sealed Fuel Pins for Nuclear Reactors", TRG-1311 (D), 1966.

16.68 Bree, J., "Elastic-Plastic Behaviour of Thin Tubes Subjected to Internal Pressure and Intermittent High Heat Fluxes With Application to Fast Nuclear Reactor Fuel Elements", Journal of Strain Analysis, Vol 2, pp 226-238, 1967.

16.69 Zamrik, S. Y. and Hu, L. W., "Radiation Effects on Creep Rupture and Fatigue Strength of Pure Aluminum", Experimental Mechanics, pp. 193-201, May 1967. (Fatigue test specimens showed progressive elongation and necking with high maximum tensions in test with mean tensile loadings.)

16.70 Ronay, M., "Second-Order Elongation of Metal Tubes in Cyclic Torsion", International Journal of Solids and Structures, Vol 4, No. 5, pp 509-516, May 1968.

16.71 Bree, J., "Incremental Growth Due to Creep and Plastic Yielding of Thin Tubes Subjected to Internal Pressure and Cyclic Thermal Stress", Journal of Strain Analysis, Vol 13, No. 2, pp 122-127, 1968.

16.72 Coffin, L. F., "A Study of the Effects of Cyclic Thermal Stress on a Ductile Metal", Trans. ASME, pp 931-950, August 1954.

16.73 Manson, S. S., "Thermal Stress and Low-Cycle Fatigue", McGraw-Hill Book Company, 1966.

16.74 Coffin, L. F., "Introduction to High-Temperature Low-Cycle Fatigue", Experimental Mechanics, pp 218-224, May 1968.

16.75 Benham, P. P., "High-Temperature Low-Cycle Fatigue: Survey of British Work", paper presented at S.E.S.A. Spring Meeting, Ottawa, Ontario, Canada, May 1967.

16.76 Stewart, W. C. and Schreitz, W. G., "Thermal Shock and Other Comparison Tests of Austenitic and Ferritic Steels for Main Steam Piping", Trans. ASME, Vol 72, pp 1051-1072, 1950.
Stewart, W. C. and Schreitz, W. G., "Thermal Shock and Other Comparison Tests of Austenitic and Ferritic Steels for Main Steam Piping - A Summary Report", Trans. ASME, Vol 75, pp 1051-1072, 1953.

16.77 Weisberg, H. and Soldan, H. M., "Cyclic Heating Test of Main Steam Piping Materials and Welds at the Sewaren Generating Station", Trans. ASME, Vol 76, pp 1085-1091, 1954.

16.78 Tidball, R. A. and Shrut, M. M., "Thermal-Shocking Austenitic Stainless Steels With Molten Metals", Trans. ASME, Vol 76, pp 639-643, 1954.

16.79 Gysel, W., Werner, A. and Gut, K., "Thermal Shock Behavior of Various Grades of Cast Steel", Proceedings of Joint International Conference on Creep, Institute of Mechanical Engineers, London, pp 3-33 to 3-41, 1963.

16.80 Coffin, L. F., "An Investigation of Thermal-Stress Fatigue as Related to High-Temperature Piping Flexibility", Trans. ASME, Vol 79, pp 1637-1651, 1957.

16.81 Carden, Vogel and Kyzer, "Low-Cycle Fatigue of Three Super Alloys Under Cyclic Extension and Cyclic Temperature Conditions", ASME Paper No. 67-MET-19.

16.82 Carden, A. E., and Sodergren, J. H., "The Failure of 304 Stainless Steel by Thermal Strain Cycling at Elevated Temperature", ASME Paper No. 61-WA-200.

16.83 Thielsch, H., "Defects and Failures in Pressure Vessels and Piping", Reinhold Publishing Company, 1965.

16.84 Markl, A.R.C., "Fatigue Tests of Piping Components", Trans. ASME, Vol 74, pp 287-303 (1952).

CHAPTER 17

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# 17. DYNAMIC EFFECTS

Dynamic effects on piping systems include such phenomena as water hammer, reaction forces (as developed at a safety or relief valve or by high mass-flow rates with directional change) and vibration of the piping system or components therein. Vibration in the system may be induced by such causes as fluid-flow oscillations or pressure pulses; by vibration of equipment to which the piping is attached or by vibration of foundations induced by earthquake or other sesimic vibrations.

Table 17.1 is taken from Par. 1-701.5 of USAS B31.7[17.1] and will be used as an outline for discussion in the following.

## 17.1 Impact

Impact or shock loading is a somewhat loosely defined aspect of vibration wherein the excitation is non-periodic; e.g., in the form of a pulse or step input. In piping systems, perhaps the most common impact loading is caused by "water hammer". One aspect of water hammer concerns the relatively sudden stoppage of the flow in a long pipeline. A discussion of water hammer in pipelines is given by King[17.2]. It might be noted that water hammer arises not only due to rapid closing of a valve in a piping system but also to such operations as:

a) Delayed closing of a check valve.

b) Shutting off a pump motor.

c) Slug flow of liquid in a nominally vapor flow line.

In general, design allowances and operating procedures can take care of water hammer due to valve closing or pump shut down. The check valve

TABLE 17.1 DYNAMIC EFFECTS INCLUDED IN USAS B31.7

1-701.5 Dynamic Effects

1-701.5.1 Impact

Impact forces caused by either external or internal conditions shall be considered in the piping design.

1-701.5.3 Earthquake

The effects of earthquake shall be considered in the design of piping, piping supports, and restraints. The loadings, movements (anchor movements), and number of cycles to be used in the analysis shall be part of the design specification. The stresses resulting from these earthquake effects must be included with weight, pressure, or other applied loads when making the analysis required in Part 2 of this chapter, or in Appendix F.

1-701.5.4 Vibration

Piping shall be arranged and supported so that vibration will be minimized (see Paragraph 1-721.2.5). The designer shall be responsible by design and by observation under startup or initial operating conditions to assure that vibration of piping systems is within acceptable levels.

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1-721.2.5 Sway Braces

Sway braces or vibration dampeners may be used to limit the effects of vibration on piping systems.

problem involves selection and maintenance so that the valves close before significant reverse flow occurs. The slugging problem is common in steam piping systems during start-up and requires adequate line drainage and warm-up rate commensurate with drainage provisions. However, a water hammer possibility remains in steam lines where upset-conditions may lead to water carry-over into the steam line.

Shock loadings are significant for piping on combatant naval vessels. Considerable work has been done in this area, part of which is covered in the Shock and Vibration Bulletins[17.3].

## 17.2 Earthquake (Seismic)

The terms seismic and earthquake are used almost interchangeably in reference to dynamic effects on piping systems. However, there is usually an implication that earthquake is a "natural" earth vibration whereas seismic can include earth vibrations due to other causes such as that caused by blasting operations, vibration of heavy machinery, etc.

A general discussion of earthquake loadings and structural design procedures for such loadings is given by Housner[17.4]. A more extensive discussion, with particular reference to nuclear reactors and some reference to piping systems, is given by the AEC Document, "Nuclear Reactors and Earthquakes"[17.5]. Both of these references give extensive bibliographies on the subject.

With regard to non-nuclear piping, the American Standard Code for Pressure Piping, Sections 1, 3, and 4[17.6] all include earthquake as a loading to be considered. The pertinent paragraphs from these three codes are quoted in the following:

Section 1: Power Piping

"101.5.3 Earthquake

The effect of earthquakes, where applicable, shall be considered in the design of piping, piping supports, and restraints, using data for the site as a guide in assessing the forces involved...."

Section 3: Petroleum Refinery Piping

"301.5.3 Earthquake

Piping systems located in regions where earthquakes are a factor, shall be designed for a horizontal force in conformity with good engineering practice using governmental data as a guide in determining the earthquake force...."

Section 4: Liquid Petroleum Transportation Piping

"401.5.3 Earthquake

Consideration in the design shall be given to piping systems located in regions where earthquakes are known to occur."

In-so-far as the writer is aware, earthquake loads are usually not included in the design of commercial piping systems. Where such loads are included, they are usually considered as a static horizontal force as implied by Par. 301.5.3 of USAS B31.3. This horizontal force is often specified as being in the range of to 0.1 to 0.2 g; i.e., 0.1 to 0.2 of the weight load applied in a horizontal direction. With this input, the calculation of earthquake load effects becomes relatively routine if one has available a piping flexibility computer program which includes distributed loads*. Most such programs include weight loads; by simply

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* Such analysis considers the piping system as an assemblage of beams, accordingly any shell effects would not be included. The one exception is that curved pipe in such computer programs usually includes a flexibility and stress intensification factor based on shell effects.

interchanging the horizontal and vertical axes, one can obtain the horizontal load effects on the piping system. This horizontal force presumably is to be considered as existing in all horizontal directions. Hence, for multi-plane systems it may be necessary to make several runs to obtain "worst cases" at various piping sections. A vertical g-load might also be specified with no great complication in the analysis. It might be remarked that the design philosophy of an equivalent g-load is analogous to certain building codes with respect to earthquake loading. See, for example, Reference [17.7].

It is generally recognized that the equivalent static force method discussed above may not be conservative, even as applied to determination of maximum stresses. If the earthquake loading spectra includes a frequency close to the natural frequency of some part of the piping systems, resonance can occur. Large stresses might then develop, depending upon the time duration of the earthquake and damping in the piping system.

It is pertinent at this point to discuss the requirements of B31.7[17.1] with respect to earthquake loading. It may be noted from Table 17.1 that B31.7 requires that "the loadings, movements (anchor movements), and number of cycles to be used in the analysis shall be part of the design specification." B31.7 thus divorces itself from the complex problem of determining actual dynamic characteristics of the piping system with earthquake loading. By implication, at least, B31.7 requires that cycles due to earthquake loading be included in the fatigue analysis. B31.7, Table F-104, places earthquake loadings into two categories.

1) Inertia earthquake effects - placed in primary bending category.
2) Anchor point motions - placed in the "expansion" category.

This separation is not clearly apparent in Par. 1-705.1, but perhaps is implied by the "single amplitude" of the definitions under equation (9) of B31.7 and "double amplitude" under equation (10). The philosophy behind this separation is that the inertia loads are not "self limiting". Hence, the stresses imposed thereby should be limited to the equivalent of a limit or collapse load. The anchor displacements are, like other displacements in the expansion category, self-limiting insofar as collapse is concerned. As a result, the stresses imposed thereby can be permitted to be higher, and the 3 $S_m$ (or $2S_y$) limit is used.

The type of dynamic analysis which may be necessary for piping systems is described by the following quote from Reference (17.8). This is specifically directed towards nuclear reactor vessels but might be considered for nuclear power piping.

> "Where earthquake loadings are specified in the Design Specifications, the determination of the seismic-induced stresses shall be based upon the application of acceptable methods of dynamic analysis for the calculation of the structural response of the vessel to earthquake motions. The analysis shall take into account the response spectra of the ground motions, the degree of structural damping, and the amplification of ground motions as dictated by specific site conditions.
>
> "In determining the maximum stresses, the effect of vertical components of seismic motion shall be combined directly and linearily with the effect of horizontal components of earth quake motion, and both vertical and horizontal components shall be combined directly and linearly with other loadings specified.....

"The cyclic loading associated with design seismic-induced vibrations shall be included in the fatigue analysis.

"Consideration shall be given to out of phase displacements of the vessel supports, or components of vessels (e.g., control rod assemblies on reactor vessels, connected piping, etc.) resulting from differences in seismic-induced motions of vessels, components, and appurtenances connected thereto, and to the possibility of tilting or rotation of structural foundations upon which the reactor vessel rests.

"Explanation - A principal safety requirement for a nuclear power plant is the assurance of the capability for a safe and secure shutdown of the facility in the event of an earthquake occurring at the plant site. Such a capability must be provided for by designing nuclear power plant components (i.e., vessels) to resist the design basis earthquake without impairment of their structural integrity.

"Because of the uncertainties associated with the effects of earthquake loadings on nuclear power plant components, it is imperative that safe shutdown be reliably achieved in order to render the plant secure for the protection of public health and safety. This shutdown capability is also essential to reverify the functional operability of the protective systems and engineered safeguards for the reactor coolant system prior to resumption of plant operation."

It is perhaps obvious that the analysis suggested in the above quotation, as applied to piping systems, constitutes an involved and lengthy task. It might be remarked that the response spectra of the ground motion furnishes input data for the pipe supporting structures. These may be pressure vessel nozzles, pumps, turbines or other equipment or the pipe may be supported from building framework or from frames specially constructed to support the piping. Accordingly, the piping system analysis must include or begin with a response analysis of these supporting structures.

References (17.4) and (17.5) discuss, in some detail, the general problem of designing structures to resist earthquake loadings. These methods appear to be an extension of methods used to design buildings and similar large structures for earthquake loadings. Such methods do not consider fatigue as a failure mode. Accordingly, there is no guidance therein as to the number of cycles to be used in design. In addition to the severity of the "design earthquake," this would be a function of both duration and frequency of occurence of earthquakes.

A simplified analysis of piping systems for earthquake loading would be useful; at least in the preliminary design stage for selecting restraint locations. At present (August, 1969), the USAS B31.7 Committee, Subgroup on Design, is attempting to establish such a simplified earthquake analysis. The general concepts considered so far involve spacing of the piping supports so that the first mode natural frequency is either well above or well below the dominant frequency of the supporting structure at the restraint point.

One kind of approach, involving spacing so that the piping frequency is higher than the dominant forcing frequency, might consist of the following:

1) It is assumed that the design specification will give:

   a) The highest frequency of the design earthquake spectra, e.g., 25 cps

   b) The design equivalent static g-loading (horizontal and vertical plane); e.g., 0.15 g

   c) The design duration of earthquakes during the design lifetime; e.g., 3 earthquakes at 2 minutes each = 6 minutes.

2) The designer then spaces the piping system supports so that the first-mode natural frequency is not less than $\sqrt{2}$ times the highest frequency of the earthquake spectra. This will assure that dynamic amplification is negligible.*

3) Maximum stresses would be calculated from the specified equivalent g-load. These would be checked against permissible values in accordance with equation (9) of USAS B31.7.

4) The maximum stresses obtained from the specified equivalent g-load would also be included in the check of secondary stresses, equation (10) of USAS B31.7 and fatigue evaluation, equation (11) of USAS B31.7. In the later case, the

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* The relative-amplitude magnification factor for a single-degree-of-freedom oscillator is given by(17.13):

$$\frac{Z}{Y} = \frac{(\omega/\omega_n)^2}{\{[1 - (\omega/\omega_n)^2]^2 + [2\zeta\ \omega/\omega_n]^2\}^{1/2}}$$

For $\omega/\omega_n = 1/\sqrt{2}$, $Z/Y \leqq 1.0$.

number of cycles would be taken as the highest frequency times the design duration; e.g., if highest frequency = 25 cps, design duration = 6 minutes,

$$\text{cycles} = 25 \times 60 \times 6 = 9000 \quad .$$

There are a number of assumptions involved in the above procedure which impose significant limitations on its application. These are discussed in the following:

a) In using the earthquake spectra, it is assumed that the dominant frequency of the supporting structure (building, pressure vessel, pump, etc.) will not be higher than that of ground motion. A perhaps better alternative would require direct information on the motion of the restraint points; e.g., as its dominant frequency and acceleration or, better yet, in the form of a response spectrum.

b) In step (2), the designer must (in order for this to be a simplified analysis) model an actual three-dimensional piping system (with perhaps some concentrated loads such as valves and curved piping bends or elbows) into an equivalent straight pipe span between restraint points. The model must not be stiffer than the actual piping, otherwise the estimated frequency will be higher than the actual piping with a resulting unconservatism in the method. However, in designing on the "stiff side", the problem of amplification of higher order harmonies of the piping system does not arise because these will have higher frequencies.

c) An alternate approach would be to design the piping so that its frequency is well below the lowest significant forcing frequency at the restraint points. In this case the model should not be more flexible than the actual piping which is a little easier modeling task than designing on the stiff side. However, in this case the higher order harmonies could become significant.

d) Table 17.2 gives some indication of the kind of support spans required for frequency control as compared to those typically used for weight stresses or drainage control. This table is based on an arbitrary assumption that a dominant forcing frequency of 25 cps exists at both ends of the span. For the "stiff" design, the pipe-span-frequency is to be $\sqrt{2}$ x 25 cps, and for the "flexible" design the pipe-span-frequency* is to be 0.5 x 25 cps. Table 17.2 shows directly the span lengths required for a span modeled as having simply supported ends. This is basically conservative for the "stiff" design. An increase in span length could be justified only if the actual restraints can be shown to be more rigid than a simple support. For the "flexible" design, a conservative approach would involve a fixed-fixed ends assumption, for which the tabulated lengths would be multiplied by 1.5. Table 17.2 indicates, at least for an assumed dominant forcing frequency of 25 cps, that the restraint spacings required for frequency control are not impractical since they are in the same "ball park" as spacings used for weight/drainage control.

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* This gives a relative-amplitude magnification factor of about 4/3 for a lightly damped, single-degree-of-freedom oscillator.

TABLE 17.2. FREQUENCY AND LENGTH RELATIONSHIPS FOR PIPE SPANS

| Pipe | | L, Ft. (1) | $f_n$ (4) | | $L_s$ $f_n$ = 35.35 cps (5) | $L_f$ $f_n$ = 12.5 cps (5) |
|---|---|---|---|---|---|---|
| Size | Sch. | | Empty (2) | Full (3) | | |
| 1 | 80 | 7 | 12.8 | 12.3 | 4.1 | 6.96 |
| | 160 | 7 | 12.9 | 12.7 | 4.2 | 7.04 |
| 2 | 80 | 10 | 13.4 | 12.4 | 5.9 | 9.96 |
| | 160 | 10 | 13.5 | 12.9 | 6.0 | 10.17 |
| 4 | 80 | 14 | 14.0 | 12.6 | 8.4 | 14.0 |
| | 160 | 14 | 14.2 | 13.3 | 8.6 | 14.4 |
| 8 | 80 | 19 | 15.9 | 13.6 | 11.8 | 19.8 |
| | 160 | 19 | 15.8 | 14.5 | 12.2 | 20.5 |
| 12 | 80 | 23 | 16.6 | 13.9 | 14.4 | 24.2 |
| | 160 | 23 | 16.3 | 14.9 | 14.9 | 25.1 |
| 16 | 80 | 27 | 15.2 | 12.7 | 16.2 | 27.2 |
| | 160 | 27 | 14.9 | 13.6 | 16.7 | 28.1 |
| 24 | 80 | 32 | 16.6 | 13.6 | 19.9 | 33.4 |
| | 160 | 32 | 16.2 | 14.6 | 20.6 | 34.6 |

(1) L is support spacing taken from the Piping Handbook(17.2), p 5-4. This value of L is based on 1500 psi stress or 1/10" deflection, water-filled pipe.

(2) Empty includes weight of pipe plus weight of insulation. Insulation assumed to weigh 16 lb/cu-ft., 2" thick for 1" and 2"; 2.5" thick for 4", 8", and 12" and 3" thick for 16" and 24" pipe.

(3) Full includes weight of pipe, insulation and water.

(4) $f_n$ = first mode frequency in cycles per second for span with supported ends. For fixed-supported ends, multiply $f_n$ by 1.56; for fixed-fixed ends, multiply $f_n$ by 2.27.

(5) $L_f$ = support spacing (in feet) to obtain a frequency of $\sqrt{2}$ x 25 cps. $L_f$ = support spacing (in feet) to obtain a frequency of 0.5 x 25 cps. $L_f$ and $L_s$ are calculated for the pipe full of water. Values shown are for supported ends. For fixed-supported ends, multiply $L_n$ by 1.25; for fixed-fixed ends, multiply $L_n$ by 1.51.

It is perhaps apparent from the preceding that the development of a simplified analysis that is conservative yet not overly conservative is itself not a simple task. Additional development work is needed and may eventually lead to an acceptable and useful simplified analysis.

## 17.3 Vibration

Vibration, in a broad sense, includes the aspects discussed in Pars. 17.1 and 17.2. In this paragraph, a few brief comments will be given on vibration in piping due to (1) external excitation and (2) fluid flow pulsation. An excellent reference on the problem is contained in Chapter 9 of the M. W. Kellogg book(17.9) on Design of Piping Systems.

From a structural aspect, the piping designer is concerned with vibration as it may cause fatigue failure. In addition, vibration may lead to excessive wear in valves (particularly check valves) and other equipment.

In general, it is quite difficult to design a piping system so as to eliminate vibration problems. This difficulty arises, in part, because in the design stage the excitation sources are not completely known. As a result, vibration problems in piping systems are quite often first observed in operation. They are then assessed as to potential damage and, if deemed necessary, they are "field-fixed"; usually by additions or changes in supports or restraints. The B31.7 Code recognizes this practical aspect in that (see Table 17.1) it states: "The designer shall be responsible by design and observation under start-up or initial operating conditions to assure that vibration of piping systems is within acceptable levels."

## 17.31 External Excitation

External excitation of piping systems normally arises from the vibration of attached equipment such as pumps or compressors. These excitation frequencies are usually above the first-mode beam frequencies of typical pipe spans. However, they may induce higher-modes of beam bending or may induce some of the shell-bending frequencies. In outdoor piping, wind-flow may cause vibration or wind-flutter.

## 17.32 Fluid Flow Pulsation

From a structural design aspect, fluid flow or pressure pulsations is a potential problem both in that as a cyclic pressure it produces cyclic stresses, and in that the cyclic pressures may excite mechanical vibration of the piping system. Additional problems arise due to wear on valves, loss of efficiency in line flow and in gas compressor performance and difficulties with flow measurement.

The problem of pressure pulsation at natural gas pipeline compressor stations has received considerable attention over the past few years. It has been found economical to install relatively complex pulsation dampeners in the form of acoustic bottles, baffles, and choke tubes. Two recent papers on the subject are by Scheel[17.10] and Nimitz[17.11].

Finally, it should be noted that under certain conditions a "steady-state" flow in straight pipe can induce vibrations in the piping. This aspect is discussed by Stein[17.12], who includes 14 references on the subject as well as additional development of the theory. In general, the flow-velocities/span lengths involved are not encountered in normal piping systems except for upset flow conditions or possibly at relief or safety valve discharge conditions.

## 17. REFERENCES


17.1 USAS B31.7, American Standard Code for Pressure Piping, Section 7, Nuclear Power Piping, dated February, 1968, Issued for Trial Use and Comment, Published by the American Society of Mechanical Engineers, 354 E. 47th Street, New York, N. Y. 10017.

17.2 Piping Handbook, 5th Edition, McGraw-Hill (1967), Chapter 3 by R. C. King and Chapter 5, prepared by Committee of the Manufacturers Standardization Society of the Valves and Fittings Industry.

17.3 Stress and Vibration Bulletins of the Shock and Vibration Information Center, Naval Research Laboratory, Washington, D. C., Specifically on Piping:

Neubert, V. H., "Dynamic Shock Analysis of Structural Components and Piping Networks, Bulletin No. 27, Part 1, p 92.

Neubert, V. H., "Vibration Interaction of Foundation Equipment and Piping", Bulletin No. 29, Part IV, p 307.

Blackstock, W. J., and Loria, J. C., "Shock Effects on the Propellant Loading System of a Missile Complex", Bulletin No. 32, Part III, p 115.

Lipner, N., and Fay, F. B., "Acoustic Waves Generated by the Motion of Piping Containing a Fluid", Bulletin No. 35, Part III, p 161.

17.4 Housner, G. W., "Vibration of Structures Induced by Seismic Waves", Chapter 50 of Shock and Vibration Handbook, McGraw-Hill, 1961.

17.5 Nuclear Reactors and Earthquakes, Prepared by Lockheed Aircraft Corporation and Holmes & Narver, Inc. for the USAEC, Division of Reactor Development, August, 1963, TID-7024.

17.6 American Standard Code for Pressure Piping, Section 1 - Power Piping, Section 3 - Petroleum Refinery Piping, Section 4 - Liquid Petroleum Transportation Piping Systems. Published by the American Society of Mechanical Engineers, 345 E. 47th St., New York, N. Y. 10017.

17.7 The Uniform Building Code, 1958, Pacific Coast Building Officials Conference, 610 South Broadway, Los Angeles, California.

17.8 Tentative Supplementary Regulatory Criteria for ASME Code-Constructed Nuclear Pressure Vessels, August 23, 1967, U.S. Atomic Energy Commission, Washington, D. C.

## 17. REFERENCES (contd)


17.9 Design of Piping Systems, prepared by Staff of M. W. Kellogg Company., Second Edition, 1956, John Wiley and Sons.

17.10 Scheel, L. F., "Compressor Pulse Dampers - When Are They Necessary", Hydrocarbon Processing, Vol. 46, No. 2, pp 149-154 (1967).

17.11 Nimitz, W., "Pulsation and Vibration Causes and Effects", Pipe Line Industry, August, 1968, and September, 1968.

17.12 Stein, R. A., "Vibration of Tubes Containing Flowing Fluid", Ph.D. Thesis, Ohio State University, September, 1967; also available as a report from Battelle Memorial Institute, Columbus, Ohio.

17.13 Handbook of Engineering Mechanics, Chapter 56 on Systems of One Degree of Freedom, by W. T. Thomson, McGraw-Hill Book Co., 1962.