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ORNL-4821

THIRD ANNUAL PROGRESS REPORT
ON STUDIES IN APPLIED SOLID MECHANICS
(NUCLEAR SAFETY)

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Printed in the United States of America. Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road, Springfield, Virginia 22151
Price: Printed Copy \$3.00; Microfiche \$0.95

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ORNL-4821
UC-80 - Reactor Technology

Contract No. W-7405-eng-26

Applied Mechanics Section

Reactor Division

THIRD ANNUAL PROGRESS REPORT ON STUDIES IN APPLIED SOLID MECHANICS
(NUCLEAR SAFETY)

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DECEMBER 1972

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UNION CARBIDE CORPORATION
for the
U.S. ATOMIC ENERGY COMMISSION

See

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ABSTRACT

Accomplishments during calendar year 1971 under ORNL nuclear-safety-related solid mechanics programs for developing structural design technology for piping components and pressure vessel nozzle attachments are summarized. Carefully coordinated experimental and theoretical studies are being conducted to develop improved design methods for assuring the structural integrity of nuclear reactor pressure vessels and piping systems. This technology, however, is not limited to nuclear systems, but may be applied to nonnuclear systems as well. These programs provide experimental data on structural behavior, from which experimentally proven methods of design analysis may be established. In this manner design rules and analytical procedures are developed for incorporation into current industry-developed codes and standards and USAEC Division of Reactor Development and Technology (DRDT) standards. The ORNL piping and nozzle programs are discussed in Parts I and II respectively.

Keywords: stress analysis, nuclear piping, pressure vessel nozzles, piping code, pressure vessel code, ASME III Boiler and Pressure Vessel Code, shell structures, limit loads, fatigue, piping design, stress indices.

INTRODUCTION AND SUMMARY

This third annual progress report describes work completed during calendar year 1971 under two ORNL programs on the development of structural design technology for nuclear power plant applications, one for the study of piping components and the other for studies of nozzles in pressure vessels. These programs are part of the general AEC-industry efforts to develop effective and economical means for assuring the continued adequate and safe design of nuclear plants and are thus an essential part of the growing technology on nuclear safety. Both programs are being conducted in close cooperation with industry through the Pressure

Vessel Research Committee (PVRC) of the Welding Research Council. For the piping program, the PVRC Subcommittee to Develop Stress Indices for Piping, Pumps, and Valves acts both as a consulting panel of experts and as an effective liaison between the research program and committees that write design rules for industrial codes and standards. For the nozzles program, these functions are carried out by the PVRC Subcommittee on Reinforced Openings and External Loadings.

The piping program is supported entirely by the USAEC Division of Reactor Development and Technology (DRDT) under the Nuclear Research and Development Program. Projects of the nozzle program, however, are sponsored by the PVRC as well as by the USAEC. These are also discussed since ORNL acts in a management capacity for the PVRC Subcommittee on Reinforced Openings and External Loadings. Periodic progress reports for both programs are included in the minutes of the PVRC meetings, which are issued three times a year, and in the bimonthly progress reports of the ORNL nuclear safety program.

The ORNL piping program, consisting of a number of interrelated studies on the structural behavior of piping system components, is discussed in Part I. With the work completed during 1971, the total program effort is about 50% complete. Much of the work was reported in the open literature, and many of the proposed recommendations for changes in the design rules of the codes and standards have been adopted.

Significant progress was made on the analytical stress analysis studies for straight pipe, in-line piping components, and attachments. These studies are now about 65% complete. Proposed design rules and stress indices developed for straight pipe with initial out of roundness were incorporated into Section III of the ASME Boiler and Pressure Vessel Code. An in-depth review of current code rules and stress indices for girth-welded joints and tapered transitions was completed, and a number of the recommendations were adopted by the code. Stress indices were developed for socket-welding and threaded fittings, a report on stress indices for concentric reducers was drafted, and work was initiated on developing stress indices for flanged joints.

Experimental and analytical studies originally outlined for thin-walled idealized tees and small branch connections are now about 50%

complete. Two of the four thin-walled tee models (cylinder-to-cylinder connections) have been tested and significant results reported. The remaining two models are being instrumented for testing during the coming year. Flexibility factors for small branch connections were developed, and a special code case is being prepared for consideration by the code committee. Our study showed that the present code flexibility formula is restrictive for certain branch connection configurations permitted by the code; hence the new formula will tend to ease these restrictions.

Good progress was made toward completing the analytical and experimental studies for elbows, curved pipe, and miter bends. An analytical study on stress indices and flexibility factors for elbows and curved pipe was completed which showed that the present code formulas are essentially correct if the discontinuity effects (end effects) of pipe or other components welded to the pipe can be ignored. Progress was made toward including these effects into the stress analysis methods. Experimental stress analysis tests were completed on one of four machined-elbow models to be tested under this program, and the second model was partially instrumented.

Planned studies for B16.9 standard tees are about 50% complete. These studies include elastic response tests and fatigue-to-failure tests of 17 tees, photoelastic stress analyses of four 1/2-scale models, thermal stress tests of two tees, and the development of a finite-element computer code for the elastic stress analysis of tees. Elastic response and fatigue tests have been completed on seven of the tees, and test model preparations are well under way for four more. The photoelastic model studies were completed and the final report was published. Test work was resumed on the thermal stress tests, and excellent progress was made on developing the finite-element computer code.

Two important limit-load studies were completed during 1970, the experimental limit-load studies for elbows subjected to bending loads and analytical elastic buckling studies for straight pipe restrained against thermal expansion. Publications describing both of these studies were prepared.

The nozzles program, discussed in Part II, consists of a series of closely coordinated theoretical and experimental investigations of nozzle-

to-vessel attachments. Included are studies of individual and closely spaced nozzle configurations that may be either radially or nonradially attached to pressure vessels or flat plates used to simulate a portion of a large-radius vessel. Considerable progress was made on these studies.

A satisfactory analysis of elastic behavior of nonradial nozzles attached to spherical shells has been developed, and theoretical results have compared favorably with data from recently completed experimental studies.

Two of the more extensive experimental investigations, a photoelastic study of single and closely spaced nozzles and a strain-gage study of closely spaced holes in steel plates, were completed. Good comparisons were obtained for theoretical and experimental results from the flat-plate study.

The analytical and experimental work has been extended to include clusters of nozzles attached to flat plates. This is a necessary second step toward the projected goal of developing an analysis for clusters of nozzles attached to pressure vessels.

Results of parameter studies of possible reinforcement configurations for nozzle/spherical shell junctions have been compared with code design rules, and modifications of these rules have been recommended. This work is part of a continuing program of code rule development which utilizes information developed under this and other relevant programs.

PVRC-sponsored upper- and lower-bound limit-load studies of nozzle/cylindrical shell attachments are continuing, and a significant portion of the experimental and theoretical studies has been completed.

PART I. ORNL PIPING PROGRAM – DESIGN CRITERIA FOR PIPING, PUMPS, AND VALVES

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PROGRAM DESCRIPTION

The ORNL piping program is supported by the U.S. Atomic Energy Commission under the Nuclear Safety Research and Development Program. The program, consisting of about 30 interrelated analytical and experimental studies on the structural behavior of piping system components, is organized into tasks and subtasks according to subject matter. These tasks and the present status of the work under each are given in the conclusions. The primary objective of these studies is to develop basic structural response information for use in writing and improving design and construction codes and standards for nuclear power piping systems. The studies currently being pursued provide information primarily for light-water commercial power reactor systems; however, because of the fundamental nature of the investigations, a substantial portion of the information is directly applicable to other critical piping systems as well.

Work under the program was initiated late in FY 1967 as an outgrowth of a research proposal¹ developed by an ad hoc committee of the Pressure Vessel Research Committee (PVRC) of the Welding Research Council in response to requests from several code-writing committees. Subsequently the AEC agreed to sponsor the piping components portion of the PVRC proposal as a cooperative effort with industry and charged ORNL with the

responsibility for planning, managing, and directing the AEC portion of the program. One of the specific directives given ORNL was to develop recommended requirements for specifications, codes, and standards needed to assure the continued safety of nuclear plant piping systems.

To facilitate management of the AEC-sponsored portion of the program, a comprehensive program plan² was developed which presently forms the basis for the work being carried out under the program. This plan includes all the work on piping components requested by the PVRC plus additional work requested by the AEC. During the past year several experimental stress analyses of thin-walled components (tees and elbows) were added to the plan to provide information of more direct use to liquid-metal- and gas-cooled reactor systems.

Close cooperation between the program and the code-writing bodies has been maintained continuously through the auspices of the PVRC Subcommittee to Develop Stress Indices for Piping, Pumps, and Valves. Stress indices and flexibility factors for code use were developed early in the program under a subcontract with Battelle Memorial Institute (BMI) and were based largely on information published in the open literature.³ These indices were incorporated into the ANSI* tentative standards^{4,5} for nuclear power piping issued in 1968 and 1969 respectively. The indices, with revisions based on information developed under this program, were also included in the first official code for nuclear power piping⁶ issued late in 1969. During 1970 and 1971 the ASME incorporated design rules for nuclear piping, pumps, and valves into the ASME Boiler and Pressure Vessel Code. The 1971 edition of ASME Section III,⁷ issued in July of that year, included the stress indices and flexibility factors from the B31.7 1969 code and more comprehensive rules for the evaluation of cyclic fatigue conditions for piping. At that time the ANSI B31.7 committee became officially inactive. As a result of work completed under the ORNL piping program, a number of changes in the design rules and stress index tables for piping components in ASME Section III are currently being processed for incorporation into the code.

*Formerly the United States of America Standards Institute (USASI) and the American Standards Association (ASA).

Summaries of work completed during CY 1971, including new work as well as the continuation of efforts reported^{8,9} for CY 1970, are presented here.

STRAIGHT PIPE, IN-LINE CONFIGURATIONS, AND ATTACHMENTS

This task consists of conducting a number of analytical parameter studies for components that can be classed under the general heading of straight pipe. Included are studies on the effects of lug attachments, out of roundness, and circumferential butt welds in straight pipe and the development of stress indices for 2-in. and smaller socket-welding and threaded fittings, reducers, transition joints, and flanged joints. According to the original program plan, these studies were to have been based on existing theoretical solutions and correlations of existing experimental data, although a certain minimal amount of analytical work and computer programming was expected. No experimental work for these components was planned; however, as pointed out last year, additional experimental data are needed in some cases in order to fully validate the analytical results.

Analytical studies and design code rules development for several of the components completed during the past year are summarized below.

Stresses in Out-of-Round Pipe

It was reported last year⁸ that analytical studies on the effects of out of roundness in straight pipe loaded with internal pressure were completed and published.¹⁰ The results of the study indicated that significant circumferential bending stresses are likely to occur in pipe purchased to the dimensional requirements of most of the current standards, including ASTM A530, which is most often used in the manufacture of straight pipe.¹¹ This has special significance for designers of nuclear power piping systems, who must satisfy the requirements of Section III of the ASME Boiler and Pressure Vessel Code.⁷ A strict interpretation of the

1971 rules, which in this case were adopted verbatim from the 1969 edition of USAS B31.7 (Ref. 6), would require that the cross sections of every piece of straight pipe, curved pipe, welding end elbow, or circumferential weld be dimensionally inspected to determine the degree of out of roundness. If the cross section is noncircular but approximately elliptical, the code gives a simple formula for calculating the stress index K_1 for use in the fatigue analysis. However, if the component has flat spots or peaks, the designer is required to "... make a suitable theoretical or experimental analysis ... to determine the stresses at such irregularities."

In most cases the above rules are not practicable because the design will have been completed before the pipe is ordered. Thus the designer must either make an arbitrary assumption about the shape of the cross section or write especially tight purchasing requirements. To correct this situation, proposed revisions, based on the results from the ORNL study on out-of-round pipe, were developed and adopted by the code committee. The following footnote to the stress index table* of Section III will appear in the Winter 1971 addenda¹² in place of the present footnote 1.

- "(1) a. The values of K_1 shown for these components are applicable for components with out of roundness not greater than $0.08t$, where out of roundness is defined as $D_{\max} - D_{\min}$, and

$D_{\max}$ = maximum outside diameter of cross section, in.,

$D_{\min}$ = minimum outside diameter of cross section, in.,

t = nominal wall thickness, in.

- b. If the cross section is out of round such that the cross section is approximately elliptical, an acceptable value of K_1 may be obtained by multiplying the tabulated value of K_1 by the factor F_{1a} :

$$F_{1a} = 1 + \frac{D_{\max} - D_{\min}}{t} \left(\frac{1.5}{1 + 0.455 (D_o/t)^3 (P/E)} \right),$$

*Table NB-3683.2-1, Stress Indices for Use with Equations 9, 10, and 11 of NB-3650. Footnote 1 as amended will apply to straight pipe, girth butt welds between straight pipe or between pipe and butt welding components, longitudinal butt welds in straight pipe, and curved pipe or butt-welding elbows.

where

D_o = nominal outside diameter, in.,

P = internal pressure, psi (use maximum value of pressure in the load cycle under consideration),

E = modulus of elasticity of component material at room temperature, psi.

Other symbols are defined in (a).

- c. If $D_{\max} - D_{\min}$ is not greater than $0.08D_o$, an acceptable value of K_1 may be obtained by multiplying the tabulated values of K_1 by the factor F_{1b} :

$$F_{1b} = 1 + \frac{MS}{PD_o} \frac{Y}{2t},$$

where

$M = 2$ for ferritic steels and nonferrous materials except nickel-chrome-iron alloys and nickel-iron-chrome alloys,

$M = 2.7$ for austenitic steels, nickel-chrome-iron alloys and nickel-iron-chrome alloys,

S_y = yield strength at design temperature, psi,

F = design pressure, psi,

D_o and t are defined in (a) and (b)."

Clause (a) in the new rules permits a small out of roundness to be disregarded. It will be useful for heavy-walled pipe and may cover a good part of class I piping for water-cooled reactors as well without special purchasing requirements, since large-size thick-walled pipe purchased to the ASTM A530 standard¹¹ tolerances will be exempted by this clause. Clause (b) is almost the same as in the previous rules, except that for out-of-round pipe which is approximately elliptical the peak stress index is to be taken as the product of the tabulated values of K_1 and the multiplication factor F_{1a} , whereas before F_{1a} replaced the tabulated values. The new rules also specify that room-temperature values for the elastic modulus E and the maximum value for the internal pressure P should be used in the stress calculations. Clause (c) gives an acceptable value for the stress index K_1 for any type of out of roundness of

rather large magnitudes, up to 8% of the outside diameter. This clause may be the most useful, since, if the component is specified to meet quite liberal diameter tolerances, no detailed inspection will be necessary. In most cases, it is believed that the large value of the K_1 stress index resulting from this clause will not present a design problem.

Girth-Welded Joints and Tapered-Wall Transitions

During the past year a rather extensive review and analysis of stress indices for girth-welded joints and tapered-wall transitions was completed.¹³ This study was conducted to summarize all the available analytical and experimental data for these components, to use this information to reassess the present code indices, and to develop recommendations for appropriate changes. Most of the basic information had been developed at the time the present code indices were formulated (1967), when it was felt that conservative and reasonably accurate indices could be developed with a minimum effort in order to meet the publication deadline for the B31.7 piping code.⁴ It was realized, however, that a more extensive study should be made using analytical models which could easily be developed from existing theory and experimental data that were available. On this basis the PVRC included tasks for the development of stress indices for girth-welded joints and tapered-wall transitions in its original program proposal.¹

The study by Rodabaugh and Dodge¹³ reviewed B, C, and K stress indices for use in the class I piping system design analysis procedures of NB-3652 and NB-3653 (ASME-III-1971, Ref. 7) for (1) girth-butt welds between segments of straight pipe or between straight pipe and butt-welding components, (2) girth fillet welds used with socket-welding fittings, slip-on flanges, or socket-welding flanges; and (3) tapered transition joints which meet the fabrication requirements of paragraph NB-3661.2 of the code. Table 1, abstracted from the code, gives the 1971 values of the stress indices. In addition to these 32 indices, the design procedures of NB-3653 require the use of an additional stress index, denoted as C'_3 , for the evaluation of membrane stresses due to thermal loadings. Values

Table 1. Stress indices for girth welds and tapered-wall transition joints
Abstracted from Table NB-3683.2-1, Ref. 7

Component	Internal pressure			Moment loading			Thermal loading	
	B ₁	C ₁	K ₁	B ₂	C ₂	K ₂	C ₃	K ₃
Girth butt weld between straight pipe or between pipe and butt-welding components ^a								
Flush	0.5	1.0	1.1 ^b	1.0	1.0	1.1	1.0	1.1
As welded	0.5	1.1	1.2 ^b	1.0	1.0	1.8	1.0	1.7
Girth fillet weld to socket-welding fitting, slip-on flanges, or socket-welding flanges	0.5	2.0	3.0	1.0	1.5	2.0	1.8	3.0
Tapered transition joints per NB-3661.2 and Fig. NB-4233-1 (the K ₂ indices are for joints where a girth-butt weld is made at the thin end of the taper)	0.5	1.4	1.5	1.0	1.2	1.8	1.0	1.5

^aFootnote 2 of Table NB-3683.2-1 defines the terms flush and as welded for purposes of this table and also gives rules for combining stress indices for intersecting longitudinal and girth welds. Wording in this footnote will be changed in the Winter Addendum (1971) of ASME Section III.

^bFootnote 1 of Table NB-3683.2-1 of ASME III refers to out of roundness requirements, which are discussed in ORNL-TM-3244 (Ref. 10).

for C'_3 were also developed for each of the components considered in this study.

The B stress indices, used with the appropriate equations of NB-3652, are intended to protect the piping component against gross plastic deformation under combined internal pressure and applied bending moment loads. The values of 0.5 for B_1 and 1.0 for B_2 in Table 1 for all the components imply that gross plastic deformation will not occur if the sum of the nominal axial membrane stress due to pressure and the nominal axial bending stress is equal to or less than the yield strength of the pipe. It is further implied that the weld itself will not adversely affect the plastic deformation characteristics of the structure. The first assumption is supported by analytical limit-load studies for straight pipe.¹⁴ The second implication seems reasonable if the weld material has a yield strength equal to or greater than the base material and if the welding procedures for class I piping are followed. It is recommended in our study¹³ that the present B_1 and B_2 stress index values remain unchanged. However, an experimental limit-load study will be conducted under a different task to verify these conclusions.

The C and K stress indices, used with the appropriate equations of NB-3653, are intended to protect piping components against fatigue failure. If the sum of the primary plus the secondary stress intensities* at every point on the structure is limited to twice the yield strength of the material, it is presumed that the cyclic behavior of the structure will become elastic within a few loading cycles. This phenomenon is called shakedown. For states of stress below the shakedown limit, the required fatigue evaluations can be carried out on an elastic basis. The C indices (primary plus secondary stress indices) are used to determine whether or not shakedown occurs.

In order to assess the adequacy of the existing C indices, a number of parameter studies based on elastic shell theory and suitable analytical models were conducted. This procedure essentially follows that used in the original development, except that more-realistic analytical models were

*For a more precise discussion of the stress categories and limits of the code, see Table NB-3222.1 of Ref. 7.

used to include the effects of code-allowable weld contours and allowable mismatch between components at the weld joint. In addition, a more extensive investigation was made on the influence of permissible variations in the dimensional parameters. Based on the results of more than 2000 individual elastic analyses, correlations were developed for calculating the maximum stresses. These correlations were then compared with the C indices shown in Table 1 and used as the basis for recommended changes as well as for developing appropriate values for the C'_3 stress indices. Recommendations concerning the C indices (and the K indices discussed below) for girth welds and tapered transitions are given in Table 2.

The K or "peak" stress indices are used in performing the code-required fatigue analyses. They are intended to account for local stress concentrations which are not included in the normal elastic thin-shell theory. Since their primary purpose in the code analysis design procedures is to protect the component against fatigue failure, the values given in the codes should ultimately be based on actual fatigue data obtained from similar components. Therefore the existing fatigue test data were reviewed in some detail. For all but six of the K indices, the recommended values shown in Table 2 are essentially the same as the values shown in Table 1. Five of the six changes are for girth welds at tapered transitions and result mainly from the corresponding changes recommended for the C stress indices. The other change is for K_2 for an as-welded butt joint in thin-walled pipe, that is, with $t \leq 3/16$ in. This change results partly because of changes in the code welding procedures for thin-walled pipe and partly because of inadequate fatigue test data for thin-walled pipe. The report¹³ also recommends that all of the stress indices in Table NB-3683.2-1 of the code be restricted to components with diameter-to-thickness ratios less than 100. A number of the changes recommended in the study by Rodabaugh and Dodge¹³ have now been adopted by ASME-III (see the Winter 1971 Addenda), and the remaining are under consideration.

Socket-Welding and Threaded Fittings

The codes and standards permit the use of socket-welding pipe joints nominally 2 in. and smaller in class I piping systems (see Ref. 7, NB-

Table 2. Summary of recommended stress indices (for $D_o/t \leq 100$)^a

Component	Internal pressure			Moment loading			Thermal loading		
	B_1	C_1	K_1	B_2	C_2	K_2	C_3	C'_3	K_3
Girth butt weld between straight pipe or between pipe and butt-welding components ⁽²⁾ , ⁽¹²⁾									
Flush	0.5	1.0	1.1 ⁽¹⁾	1.0	1.0	1.1	1.0	0.5	1.1
As welded, $t > 3/16$ in. and $\delta/t \leq 0.1$	0.5	1.1	1.2 ⁽¹⁾	1.0	1.0	1.8	1.0	0.5	1.7
As welded, $t \leq 3/16$ in. or $\delta/t > 0.1$	0.5	1.1	1.2 ⁽¹⁾	1.0	1.4	2.5	1.0	0.5	1.7
Girth fillet weld to socket weld fittings, slip-on flanges, or socket-welding flanges	0.75	1.5	3.0	1.5	2.1	2.0	1.8	1.0	3.0
Tapered transition joints, per Subpar. 1-727.4.2(c) and Fig. 1-727.3.1 ⁽²⁾ , ⁽⁶⁾ , ⁽¹²⁾									
Flush	0.5	(6)	1.1	1.0	(6)	1.1	(6)	1.0	1.1
As welded	0.5	(6)	1.2	1.0	(6)	1.8	(6)	1.0	1.7

^aFootnotes (1) through (10) refer to existing footnotes in Table NB-3683.2-1 of ASME III, Ref. 7; footnotes (11) and (12) are recommended additions. Footnote (6) will be changed to read as follows:

"(6) Indices are applicable to tapered transition joints with a girth butt weld at the thin end of the transition.

$$C_1 = 1.3 + 0.003 (D_o/t) + 1.5 (\delta/t) \text{ but not greater than } 2.0$$

$$C_2 = 1.4 + 0.004 (D_o/t) + 3.0 (\delta/t) \text{ but not greater than } 2.1$$

$$C_3 = 1.2 + 0.008 (D_o/t)$$

For "flush" welds as defined in footnote (2), δ may be taken as zero."

Footnote (11) which refers to branch connections, curved pipe, butt welding tees, and butt welding reducers reads as follows:

"(11) The stress indices given predict stresses in the body of a fitting. It is not required to take the product of stress indices for two components (such as a tee and a reducer, or a tee and a girth butt weld) when welded together or joined by a piece of straight pipe, the length of which is less than one pipe diameter. For this specific case the stress index for the curved pipe or butt welding elbow must be multiplied by that for the girth butt weld."

Footnote (12) reads:

"(12) δ is defined as the maximum permissible mismatch as shown in Fig. NB-4233.1 (a value of δ less than $3/32$ in. may be used provided the smaller mismatch is specified for fabrication)."

3661.5), and they are often used in instrument lines, etc., in modern nuclear power plants. Threaded joints are not restricted as to size, but threaded joints in which the threads provide the only seal are not permitted. Inasmuch as there is no apparent advantage in using a threaded fitting with a seal weld as compared with a socket-welding fitting with a fillet weld, and the significant disadvantage of the required stress analysis, it appears likely that the use of threaded fittings in class I piping systems will be severely limited.

As discussed in the previous section, stress indices exist for the fillet weld between a fitting and the attached pipe. However, since no stress indices are given in the code for the fittings, a study was undertaken to develop stress indices for the most commonly used ANSI B16.11 standard¹⁵ socket-welding fittings. Some typical 1-in. socket-welding and threaded tees are shown in Fig. 1, where one of the tees has been sectioned

Fig. 1. Typical socket-welding and threaded piping tees for 1-in. nominal pipe.

to show the wall thickness and details of the socket into which the mating pipe is welded.

Stress indices for 3000-lb and 6000-lb rated fittings for tees, ⁴⁵ and 90° elbows, and couplings are discussed in a report¹⁶ that is currently being written. Twenty-seven new stress indices are being developed to cover analyses of the bodies of the fittings. These, along with the indices for the fillet welds, will provide the designer with means for satisfying the intent of the code in the design of 2-in. and smaller class I piping systems. A special code case is being prepared for consideration by the ASME code committee in order to expeditiously distribute this information for design use.

THIN-WALLED TEES AND BRANCH CONNECTIONS

This task consists of both experimental and analytical stress analysis studies of thin-walled idealized tees (cylinder-to-cylinder connections) and small branch connections fabricated according to the rules of Paragraph NB-3643, ASME Section III.⁷ Studies of these components were combined under a single task heading because of their similar geometry and because of the commonality of analytical methods used in determining the stresses. The studies of thin-walled tees as defined here refer to analytical and experimental stress analyses of a series of four very carefully constructed thin-walled models, whereas the small branch connection studies are limited primarily to analytical studies and the development of stress indices and flexibility factors for use in the codes and standards. Progress made during the past year is discussed under the two subtasks below.

Idealized Thin-Walled Tees (Cylinder-to-Cylinder Models)

Both experimental and analytical stress analyses are being conducted on the four carbon-steel thin-walled cylinder-to-cylinder models listed

in Table 3. The models are thin-shell structures consisting of two circular cylindrical shells intersecting at right angles. There are no transitions, reinforcements, or fillets at the junction, and the models are as geometrically perfect as could be obtained. Model 1 was made from two cylinders welded together, and models 2 and 3 were machined from solid forgings. Model 4 is obtained by boring out the nozzle of model 3. Each model in this series is being tested and analyzed for 13 different loading conditions, including internal pressure and each of three mutually perpendicular bending moments and each of three mutually perpendicular force loads applied to the branch end and to the run end of the model. In these tests the models are rigidly supported at one end of the run and are supported with a counterweight at the other end to balance the weight of the model and the pressurizing fluid.

During the past year the experimental and analytical studies for model 1, which were completed earlier,^{8,9} were reported in the open literature,¹⁷ and a more detailed technical account was prepared for publication as an ORNL report.¹⁸ We also completed the studies for model 3 and

Table 3. Thin-shell cylinder-to-cylinder tee models

Model No.	Major dimensions ^a (in.)				Dimensionless parameters			
	D_o	d_o	T	t	d_o/D_o	D_o/T	d_o/t	s/S^b
1	10.0	5.0	0.1	0.05	0.5	100	100.0	1.00
2	10.0	10.0	0.1	0.1	1.0	100	100.0	1.00
3	10.0	1.29	0.2	0.168	0.129	50	7.7	0.154
4 ^c	10.0	1.29	0.2	0.064	0.129	50	20.2	0.40

^a D_o is the outside diameter of the larger cylindrical shell, d_o is the outside diameter of the nozzle (the smaller cylinder), T is the wall thickness of the cylindrical shell, and t is the wall thickness of the nozzle.

^bThe dimensionless parameter s/S is the ratio of membrane stress in the nozzle to the membrane stress in the cylindrical shell due to internal pressure loading; in terms of other parameters, $s/S = d/D \times T/t$.

^cModel 4 to be made by boring out the nozzle of model 3.

reported the results in a paper¹⁹ presented at the First International Conference on Structural Mechanics in Reactor Technology, Berlin, 1971. Figure 2 shows model 3 in the loading frame with an in-plane bending moment being applied to the nozzle.

Model 3 was less extensively instrumented than model 1 (see Refs. 8 and 9) because of the small size of the nozzle and because of the experience gained in testing the first model. This model was instrumented with 165 three-gage strain rosettes, each having an individual gage length of 0.030 in. The strain-gage layout for model 3 is shown in Fig. 3. The maximum experimental stress values for both models and all 13 loading cases occurred at the junction between the nozzle and the cylinder. Comparisons between the normalized maximum principal stresses (absolute values) around

Fig. 2. ORNL thin-walled tee model 3 in the test frame with an in-plane bending moment being applied to the branch.

Fig. 3. Strain-gage layout for ORNL thin-walled tee model 3.

the junction of models 1 and 3 are shown in Fig. 4 for three of the loadings - internal pressure, an out-of-plane bending moment (M_{YN}), and an in-plane bending moment (M_{ZN}) applied to the nozzle. The values shown in the figure were normalized to the nominal membrane stress in the cylinder for the internal pressure loading ($PD/2T$). For the moment loadings, the nominal bending stress in the branch (M_c/I) was used as the normalizing factor. The solid and dashed lines show the experimental values obtained from the strain gages immediately adjacent to the nozzle. The "extrapolated" point values are estimates of the peak values at the junction obtained by extrapolating the data obtained from the gages positioned along

Fig. 4. Variation of maximum principal stress absolute value ratios around the nozzle-cylinder junctions of ORNL thin-walled tee models 1 and 3.

the respective ray line. For model 1 these peak values varied from 10.0 to 35.3, while for model 3 they varied from 2.6 to 5.1. Thus the stresses at the junction were considerably higher for the model with the larger-diameter branch connection (model 1, $d/D = 0.5$ vs model 3, $d/D = 0.129$).

During the past year we also completed the fabrication and instrumentation of model 4 and conducted 3 of the 13 basic loading tests on the model: internal pressure, an in-plane bending moment, and an out-of-plane bending moment applied to the nozzle. These tests were conducted with one end of the run fixed to the loading frame and the other end free and also with both ends of the run fixed. There was essentially no difference in the results from either set of restraint conditions.

Model 2 was partially instrumented during the past year, and we plan to complete the studies for both models during the coming year.

Small Branch Connections

In addition to the studies of intersecting cylindrical shells discussed above, analytical studies of branch connection models with diameter ratios d/D less than 0.5 are being conducted. These studies are to provide information needed to supplement the existing experimental data which were used as a basis for the stress indices and flexibility factors presently in the code.²⁰

During the past year a study was completed on flexibility factors for small branch connections, and the report has been partially drafted.²¹ The objective of the study was to determine the adequacy of the present code values on the basis of available experimental and analytical data. Experimental data from 22 models were compared with flexibility factors calculated using the present code rules as well as using Bijlaard's²² theoretical analysis (which is not strictly applicable) and with results from finite-element analyses of several unreinforced cylinder-to-cylinder models obtained using the JOINT computer code.¹⁷ The results of the study showed that Bijlaard's analysis consistently overpredicted the actual flexibility factors indicated by the available test data. However, even though Bijlaard's analysis is unconservative for flexibility, it does show

the proper trend as a function of the nondimensional variables, as indicated both by the test data and the finite-element analyses.

The present code formula is conservative, as shown by comparing the predicted flexibility factor with the test data, with Bijlaard's analysis, and with the finite-element analysis results. It is, however, apparently not uniformly conservative for certain of the reinforced nozzle configurations permitted by the code. To remedy this situation, a modified form of the code flexibility formula was proposed, and a special code case is being prepared for consideration by the ASME code committee. Additional analytical studies will be conducted for reinforced branch connections as soon as the three-dimensional finite-element computer code under development for the study of B16.9 tees is available.

STRUCTURAL ANALYSES OF ELBOWS

Piping components studied under this general heading are curved or bent pipe, elbows that meet the fabrication requirements of standards ASA B16.9 or ASA B16.28 (Refs. 23 and 24) for butt-welding fittings, and miter bends fabricated in accordance with the design rules of NB-3644 and NC-3644, ASME Section III.⁷ Three subtasks are identified in the ORNL piping program plan:² (1) analytical studies of elbows and curved pipe, including the evaluation of existing information and studies on the effects of end restraints (discontinuity effects); (2) experimental and analytical stress analysis studies of a series of machined-elbow models; and (3) theoretical and analytical stress analysis studies of miter bends. Work has been carried out for each of these except the miter bend studies, which were discontinued when funding levels were reduced. However, several papers²⁵⁻²⁸ on non-AEC-sponsored work have appeared in the literature which may be useful in developing stress indices and design rules for code use. We plan to do some work along these lines during the coming year.

Analytical Studies of Elbows

No distinction is normally made between elbows and curved pipe in the calculation of elastic stresses because of the assumed ideal shape of the

component. Even though the first rational stress analysis for elbows and curved pipe was published in 1911 by von Karman,²⁹ the structural behavior of elbows continues to be an important area for investigation. This is due to the complex nature of the theoretical formulations needed to describe the structural behavior of an elbow, to the refined experimental and analytical techniques needed to provide comprehensive data, and to the relative importance of elbows and curved pipe in the overall behavior of a piping system.

An analytical study was recently completed³⁰ in which a thorough review of the literature on the stress analysis and flexibility of elbows was made in order to select the most appropriate analytical method for the development of stress indices and flexibility factors for code use. Although more recent work has been published, the analytical method based on a minimum potential energy formulation by Rodabaugh and George,³¹ with certain modifications by Dodge and Moore,³⁰ was selected as the most appropriate. The resulting formulations for the longitudinal and circumferential stresses, as a function of angular position around the elbow, are expressed in terms of three independent nondimensional parameters: a bend parameter λ , a radius-ratio parameter γ , and an internal pressure parameter ψ . The stress variations along the length of the elbow are not considered.

A rather extensive study was conducted to determine the maximum stress intensity (twice the maximum shear stress) for an arbitrary combination of in-plane bending, out-of-plane bending, and torsional moments as a function of the three parameters λ , γ , and ψ . The numerical results were then used to develop a simple, conservative formula for calculating the primary plus secondary stress index C_2 for moment loadings on an elbow:

$$C_2 = \frac{2\lambda^{-2/3} (1 + 0.25\gamma^{-1})}{1 + \lambda^{-4/3} \exp(-\psi^{-1/4})} ; \quad \begin{cases} 0.05 \leq \lambda \leq 1.0 \\ 0 \leq \psi \leq 0.1 \end{cases} , \quad (1)$$

where the applicable range on λ and ψ is indicated to the right of the equation, and

$$\lambda = \frac{tR}{r^2 \sqrt{1 - \nu^2}} ; \quad \gamma = R/r ; \quad \psi = \frac{PR^2}{Ert} .$$

The symbol R is the bend radius of the elbow, r is the tube radius, t is the wall thickness, E is the modulus of elasticity, ν is Poisson's ratio, and P is the internal pressure. A similar formula was also developed for the flexibility factor, as follows:

$$k_p(\lambda, \psi) = \frac{1.66\lambda^{-1}}{1 + 1.75\lambda^{-4/3} \exp(-1.15\psi^{-1/4})} ; \begin{cases} 0.05 \leq \lambda \leq 1.0 \\ 0 \leq \psi \leq 0.1 \end{cases} \quad (2)$$

In this formulation the flexibility factor is a function of only λ and ψ and is the same for in-plane or out-of-plane bending. For torsion, the flexibility factor is $k_p = 1.0$.

For short-radius elbows ($\gamma = 2$) with zero internal pressure ($\psi = 0$), Eq. (1) gives a stress index C_2 which is about 12% higher than the present code formula.* For very long-radius bends and zero internal pressure ($\gamma \rightarrow \infty$, $\psi = 0$), Eq. (1) gives essentially the same value as the code formula. However, since the influence of internal pressure (which the code formula neglects) is to reduce the numerical values of both the stress indices and the flexibility factor and since Eqs. (1) and (2) do not include the discontinuity effects of end restraints (end effects), no changes are recommended in the code formulas at this time. One important point, however, which could be of interest to the liquid-metal-cooled reactor technology is that the applicable range of Eqs. (1) and (2) extends farther into the thin-shell region than the present code formulas cover.

Analytical studies on the effects of end restraints on the structural behavior of elbows have continued. Although definite conclusions cannot be made at this time, prospects for obtaining meaningful information within the coming year appear good. Three different analytical techniques are

*The 1971 ASME, Section III, code formula (Table NB-3683.2-1) for the bending moment stress index C_2 is

$$C_2 = \frac{1.95}{b_2^{2/3}} \geq 1.5 ,$$

where $b_2 = tR/r^2$ and t , R , and r are the wall thickness, the bend radius, and the mean pipe radius respectively.

currently being investigated: the finite-element method using a modified version of the JOINT computer program^{17,18} and two separate approaches to solving the thin-shell theory equations, one developed by Kalnins³² and the other³³ currently being developed in the Mathematics Division of ORNL. Each of these methods has been used during the past year with some success; however, sufficient accuracy has not been obtained to permit conducting parameter studies. When the methods are suitably developed, we plan to obtain information which will allow us to modify Eqs. (1) and (2) to include consideration of the arc length of the elbow, as well as the type of component which is welded to the elbow (pipe, flange, etc.), as additional parameters.

Machined-Elbow Model Studies

Experimental and analytical elastic-response studies are being conducted at ORNL on the four machined-elbow test models listed in Table 4. Each model consists of a 90° bend long-radius elbow (bend radius equal to three times the nominal pipe radius), with a nominal outside diameter of 10.75 in., that is welded to matching pipe stubs about two pipe diameters long on either end. Elastic-response tests for model ME-1 have been completed.

These models are being studied primarily to provide direct experimental information on the relative importance of geometric imperfections on the elastic stresses and the deflection characteristics of elbows. Model ME-1 is intended to provide: (1) base-line data for use as a comparator for the data from the other models, (2) direct experimental information on the influence of end effects, and (3) data for comparison with results from the analytical methods being developed. Models ME-2 and ME-3 are intended to give specific information on the effects of out of roundness and nonuniform wall thickness respectively. In addition to the elastic behavior studies, model ME-2 will be fatigue tested to failure under cyclic internal pressure to determine whether the small amount of ovaling which is characteristic of bent pipe will significantly influence the fatigue life of the component.

Table 4. Machined-elbow test models

Model designation	Identification	Model parameters
ME-1	Ideal torus	90° bend, long radius, 10.750-in. nominal OD, uniform wall 0.365 in. thick
ME-2 ^a	Ovalled torus	90° bend, long radius, 10.750-in. nominal OD, uniform wall 0.365 in. thick, flattened 5 to 8% of the diameter ^b
ME-3	Thinned elbow	90° bend, long radius, 10.750-in. nominal OD, bore eccentric 0.093 in., ^c minimum wall thickness 0.272 in., maximum wall thickness 0.458 in.
ME-4	Ovalled and thinned elbow	90° bend, long radius, 10.750-in. nominal OD, bore eccentric 0.093 in., ^c minimum wall thickness 0.272 in., maximum wall thickness 0.458 in., flattened 5 to 8% of the diameter ^b

^aIn addition to the elastic strain-gage tests, this model will be fatigue loaded to failure with a cyclic internal pressure loading.

^bAn attempt will be made to form an elliptical cross section with a major-to-minor diameter ratio between 1.05 and 1.08 and with the major axis lying in the plane of the bend.

^cThe eccentricity of the bore will be away from the center of the bend so that the minimum wall thickness is along the back of the elbow.

Model ME-4 is included primarily to test the assumption of linear superposition of stresses from different types of geometric deviations, such as ovality and nonuniform wall thickness. This assumption is inherent in simplified design procedures and underlies established practice for considering different loadings on the same structure. Geometric variations, on the other hand, change the shape of the structure and from purely theoretical considerations would be expected to interact in some nonlinear manner. It is postulated, however, that linear superposition may give reasonably accurate results if the geometric variations are small, as are those under consideration here.

The elastic response of each model is to be studied for internal pressure loading; for in-plane, out-of-plane, and torsional moments; and

for combinations of moment loadings and internal pressure. Approximately 80 three-gage strain rosettes will be mounted on each model, about 20 on the inside surface and the remainder on the outside surface on lines around the circumference and along the length of the model. Deflection gages will also be used to measure model deflections to obtain data for determining flexibility factors.

As mentioned above, elastic-response tests for model ME-1 were completed. This model was instrumented with 86 three-gage strain rosettes, 19 on the inside surface and 67 on the outside surface. Thirteen of the inside rosettes and nineteen of the outside rosettes were 0.062-in.-length gages placed on a circumferential line at the midplane of the elbow (station 7) as shown in Fig. 5. The remaining 54 rosettes were 0.125-in.-length gages, most of which were placed on the outside surface along circumferential lines at various positions along the model. Ten dial indicators were used to measure deflections and rotations during the test.

The tests were conducted with one end of the test assembly rigidly clamped to the loading frame as shown in Fig. 6, with bending loads transmitted to the model through a heavy flange mounted on the other end. The dial indicators and the instrument lines leading from the model to the data-acquisition system can be seen in Fig. 6. The model was tested with the 13 individual loading conditions listed in Table 5. These were internal pressure, in-plane bending, out-of-plane bending, and torsional moment

Table 5. Elastic-response test loadings for machined-elbow model ME-1

Loading	Maximum load	Constant internal pressure for combined loads (psi)		
		$1/3P_{\max}$	$2/3P_{\max}$	$P_{\max}$
Internal pressure	1,305 psi			
In-plane bending	94,080 in.-lb	435	870	1305
Out-of-plane bending	94,080 in.-lb	435	870	1305
Torsional moment	94,080 in.-lb	435	870	1305

Fig. 5. Strain-gage layout for machined-elbow model ME-1.

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Fig. 6. Machined-elbow model ME-1 in the test frame with an out-of-plane bending moment being applied.

applied independently; and combinations of internal pressure and applied moments as indicated. The combined loadings were conducted by first pressurizing the model to the indicated pressure and then adding the moment load in four equal steps to the indicated maximum value. A small additional bending moment was then applied, and the moment loading was removed in the same steps which were used during the loading sequence. For each of the combined loading tests, a complete set of strain data was obtained at each of the ten loading steps.

Typical stress plots from the in-plane bending test case are shown in Fig. 7, where the circumferential and longitudinal stress intensification factors are given as a function of the angular position around the center section of the elbow (station 7, Fig. 5). The stress intensification factor as defined here is the ratio of the measured stress to the

Fig. 7. Experimental and analytical stress intensification factors for machined-elbow model ME-1 loaded with an in-plane bending moment.

maximum bending stress in straight pipe of the same diameter and wall thickness, that is, $\sigma_{nom} = M/Z$, where Z is the section modulus of the pipe. Also shown for comparison are the analytical results obtained using the analysis of Dodge and Moore³⁰ discussed above. For this case the maximum experimental stresses appear to be about 15% less than the maximum analytical values. This is considered good, especially since the analysis is slightly conservative and since we are, as yet, unable to include end effects in the analysis. Unfortunately, several of the inside surface gages, where the maximum stresses occurred, failed during the test and could not be replaced. A different type of gage installation will be used for the inside gages on the remaining models to minimize this risk.

STUDIES OF ASA STANDARD B16.9 TEES

ASA B16.9 tees are commercially available butt-welding fittings fabricated in accordance with the ASA B16.9 or MSS-SP-48 standards.^{23, 34} The branch pipe to run pipe diameter ratio is normally greater than 1/2 but may be as small as 1/3. These fittings are characterized by a smooth transition region between the branch and run and are normally forged as a single unit. Since the wall thickness is large in comparison with the diameter, thin-shell theory is not expected to apply.

At the beginning of this program, essentially no information was available in the literature which could be applied directly in the development of stress indices and flexibility factors for piping tees. Consequently, an extensive program of both analytical and experimental investigations was undertaken. The investigations outlined under this task include various experimental analyses on a series of 17 tees ranging in nominal size from 6 to 24 in. (see Table 6). These tees represent the "standard" product of three manufacturers and include both full outlet and reducing tees; nominal wall thickness ratings of sched 10, 40, 80, and 160; and two materials, A106 grade B carbon steel and 304L stainless steel. The analytical studies include the development of an elastic finite-element computer code with three-dimensional elements specifically designed for the analysis of tees and verified against the experimental

Table 6. Experimental and analytical studies for ABA B16.9 tees

Tee No.	Manufacturer	Nominal size (in.)	Nominal wall thickness (sched)	Material ^a	Type of strain gage and fatigue test ^b	Type of photoelastic test ^c	Thermal stress test ^d	Detailed finite-element analysis ^e
T-1	I	6 x 6 x 6	40	SS	1B-M			
T-2	I	6 x 6 x 6	160	C	1B-P			
T-3	I	6 x 6 x 6	40	SS	1B-M			
T-4 ^f	I	12 x 12 x 12	80	C	1A-M			
T-5	II	12 x 12 x 12	80	C	1B-P			
T-6 ^f	III	12 x 12 x 12	80	C	1A-M	8B-M	8A-T	4A-F
T-7 ^f	II	12 x 12 x 12	160	SS	1B-M	8B-P		4A-F
T-8 ^f	II	12 x 12 x 6	40	SS	1A-M	8B-M	8A-T	4A-F
T-9	II	12 x 12 x 6	160	SS	1B-P	8B-P		
T-10 ^g	III	24 x 24 x 24	40	C	1B-M			
T-11 ^g	III	24 x 24 x 24	160	C	1B-P			
T-12 ^g	III	24 x 24 x 10	40	C	1B-M			
T-13 ^g	III	24 x 24 x 10	160	C	1B-P			
T-14	III	12 x 12 x 6	40	SS	1B-U			
T-15 ^f	I	12 x 12 x 6	40	SS	1A-M			
T-16 ^g	III	24 x 24 x 24	10	SS	1A-M			
T-17 ^g	III	24 x 24 x 10	10	SS	1A-M			

^aSS = type 304L stainless steel; C = A106 grade B carbon steel.

^bTest 1A is a detailed elastic response test using approximately 225 three-gage strain rosettes; test 1B will use fewer gages to measure the maximum stress intensity; the loading for the fatigue test is designated as M for moment loading, P for internal pressure, and U as yet undecided.

^cPhotoelastic analysis with either internal pressure (P) or out-of-plane bending moment loading on the branch (M).

^dSteady-state temperature distribution thermal stress tests.

^eFinite-element computer code development and comparison of analytical results with experimental data.

^fTested at Southwest Research Institute (complete).

^gTo be tested at Combustion Engineering at Chattanooga.

data from three of the tees (T-6, T-7, and T-8; see last column and footnote e in Table 6). The computer code will have thermal stress capabilities and will be used to analyze the thermal stress tests (column 8, Table 6) as well as to conduct parameter studies on all the tees.

Planned experimental work includes elastic response tests for internal pressure and applied bending moment and thrust loads on the branch and run, low-cycle fatigue-to-failure tests, thermal stress tests, and photoelastic model studies. The test assemblies for all the experimental work were to be fabricated by welding pipe extensions of the same nominal size and material to all three outlets of each tee according to the welding requirements of USAS B31.7 - 1969 or the ASME code which is applicable at the time the test assembly is fabricated. The indicated photoelastic studies were to be made using scaled models of the respective tee assemblies, including the pipe extensions and weld design.

During the past year, portions of the B16.9 tee studies were conducted at ORNL and, under subcontract, at the University of California, Berkeley, Southwest Research Institute, Combustion Engineering, Inc., Chattanooga, and Westinghouse Research Laboratories. These studies are described below under the appropriate subtask, along with the current status and a summary of important results. The structural behavior studies of ASA B16.9 tees were started in 1969 and should be completed in 1975. In terms of man-effort and cost these studies are essentially 50% completed.

Elastic-Response and Fatigue Studies

Both elastic-response and low-cycle fatigue-to-failure tests are being conducted on each of the tees listed in Table 6. Since each individual test is relatively expensive, a great deal of care is taken to assure that construction of the test assemblies and all phases of the test work are carried out to high professional standards. Detailed job specifications were written for each tee covering the requirements for test assembly fabrication, instrumentation, execution of the elastic-response and fatigue tests, and for data acquisition and reduction. Since adequate loading facilities and data-acquisition systems did not exist at ORNL, all

the experimental studies on tees were done under subcontract. As shown in Table 6, five 12-in. tees were tested at Southwest Research Institute (SwRI) and two 24-in. tees, T-10 and T-11, were tested at Combustion Engineering, Inc. (C.E.) in Chattanooga, Tennessee. The other four 24-in. tees will also be tested at Combustion Engineering during the remainder of this fiscal year and next (FY 1972-73). The two 24-in. sched-10 tees T-16 and T-17 were added to the test program since the last annual report in order to provide experimental information of more direct use to liquid-metal-cooled nuclear reactor systems. Orders for both tees have been placed with the manufacturer.

The elastic-response tests were initially divided into groups 1A and 1B, as indicated in Table 6, to reduce the overall cost of the test program. It was planned to use fewer strain gages for the 1B tests in the belief that significant savings could be realized from the reduced labor costs for instrumentation and data acquisition and reduction. Several factors, however, have altered this plan. All the tees which have been tested were fully instrumented with approximately 225 three-gage strain rosettes, and we plan to fully instrument the remaining tees. Roughly 70% of the cost of testing one of these tees is for fabricating the test assembly, with the remaining 30% being divided about evenly between instrumenting the model, loading the model and collecting the strain-gage data, and conducting the low-cycle fatigue test. Thus any reduction in the number of strain gages placed on the model, or the addition of a small number of gages, has a relatively small influence on the total cost of a test. Furthermore, since efficient computer methods have been developed for reducing and analyzing the data, the number of gages placed on the test model does not have a significant influence on the cost of these operations within the limits considered here.

We reported last year on the testing phase of the experimental stress analysis for the five 12-in. tees tested at Southwest Research Institute, and the 24-in. tee, T-10, tested at C.E. Since that time, more complete descriptions of the tests have been written,³⁵⁻⁴¹ and tests were completed at C.E. for the 24-in. sched-160 tee T-11. During the past year the elastic-response data obtained from those tests have been extensively studied, and an interpretive report on the results is being written. The

data were first examined for accuracy and consistency using a numerical evaluation scheme⁴² developed from the hypothesis that the strain-gage data should be proportional to the load in a linear-elastic test. The ORNL computer program LINDA identifies individual gage readings which lie outside a preselected tolerance band, determines a best-fit load-strain relationship, and calculates stresses. Those gage readings identified by the computer program as being suspect were then examined by the analyst and either discarded or adjusted. Calculated stresses were tabulated, plotted relative to the gage position, and used to construct contour maps. This information provided a means for conveniently examining the data in some detail, and we were thus able to identify any additional unacceptable data from the gages even though they had passed the earlier diagnostics.

The locations and magnitudes of the maximum stresses for each of the 13 loadings have now been identified with a high degree of certainty for all seven of the tees that have been tested. In every case the stress distributions were smooth, that is, without sharp peaks, and the maximum values occurred either at or very near one of the gage positions. Table 7 gives the magnitudes of the maximum normalized stress intensities for the internal pressure load and for the applied bending moment loads on each of the tees tested. The transverse force loadings gave results which were essentially the same as the corresponding moment loadings; for example, results from the transverse shear force applied to the branch were equivalent to the out-of-plane bending moment results. The table also indicates the location of the maximum stress intensity. For internal pressure the maximum stress was always on the inside surface near the longitudinal plane of the tee, and for moment loadings on the branch the maximum value was always on the outside surface.

Figure 8 shows the locations of the maximum stress intensities for tee T-8. For this tee all but three of the loadings - internal pressure, torsion, and in-plane moment on the run - gave maximum values on the outside surface. The maximum stress intensity for the pressure loading was on the inside surface of the crotch about 11° on either side of the longitudinal plane (0°) of the tee because of greater reinforcement on the

Table 7. Maximum normalized^a stress intensities for B16.9 tees^b

Tee	Internal pressure loading	Branch moment loadings			Run moment loadings		
		In plane	Out of plane	Torsional	In plane	Out of plane	Torsional
T-4	4.42*	2.21	2.73	2.74	2.01*	1.23*	2.32
T-6	3.31*	2.27	2.71	2.66	2.25*	1.22*	2.24*
T-7	4.43*	2.19	2.00	1.73	1.43	1.45	1.74
T-8	2.70*	1.21	2.24	1.17	2.00*	1.37	1.82*
T-10	2.87*	3.80	3.66	3.60	3.10*	1.60	3.57*
T-10	3.65*	1.61	2.31	1.11	1.90*	1.37	1.98*
T-11	3.69*	2.13	2.25	2.12	1.81*	1.19	1.78*

^aFor pressure loading, the maximum stress intensities are normalized with regard to the run pipe by the factor $PD_o/2T$, where D_o is the nominal outside diameter and T is the nominal wall thickness. For moment loadings on the branch and run, the normalizing factors are M/Z_b and M/Z_r , respectively, where Z_b and Z_r are the nominal section modulus of the branch pipe and the run pipe.

^bThe maximum stress intensities were located on the inside surface for all the numbers in the tables that are marked with an asterisk; otherwise the maximum value was on the outside surface of the tee.

Fig. 8. Maximum stress intensity locations for internal pressure and bending moment loads on B16.9 tee T-8.

inside at the 0° plane. As the figure indicates, all but two of the loadings gave maximum stress intensities at the transition (crotch) region between the branch and run. Torsion on the branch and out-of-plane bending moment on the run gave maximum values at locations near the respective pipe-to-tee welds, labeled as positions 2 and 5 in Fig. 8.

The low-cycle fatigue tests for all five of the 12-in. tees tested at SwRI and for the 24-in. tee T-10 tested at C.E. were conducted with a constant internal pressure equal to the code-allowable design pressure of the pipe (NB-3641.1, Ref. 6) and either an in-plane or an out-of-plane alternating moment on the branch, depending on which loading gave the highest stress intensity factor in the elastic-response tests. These tests were deflection controlled rather than load controlled, with the deflection of a reference point on the branch pipe limited to $\pm \delta$ throughout the test, where δ was the deflection caused by a preselected load

during the first quarter cycle. The magnitude of the initial load was established by linear extrapolation of the elastic results to the load that would give an equivalent elastic stress-intensity amplitude of $S_a = \pm 83,500$ psi at the position of maximum stress intensity on the tee. The value of S_a was based on an expected life of 7000 cycles from Markel's relationship:⁴³

$$S_a = 490,000 N^{-0.2},$$

where N is the number of cycles to fail. The maximum deflection $\pm \delta$ was then held constant throughout the test and used to control the load cycling equipment. The maximum load was adjusted during each cycle as necessary. In all six of the alternating moment fatigue tests, the change in the maximum load between the first cycle and the last cycle was small. Failure was defined as a through-the-wall crack which caused leakage of the pressurizing fluid, and in every case this crack was located near the strain gage which gave the maximum stress intensity during the elastic-response tests.

Tee T-11 was fatigue tested with a cyclic internal pressure with lower and upper limits of 100 and 7000 psi respectively; we plan to fatigue test tee T-13 in the same manner. For various reasons, including the length of time required for a test and the performance requirements of the test equipment, we were interested in a low number of cycles to failure. The maximum pressure for T-11, however, was limited to less than would be required to yield the pipe in order to preclude the possibility of plastic ratchetting. An internal pressure of 7000 psi gives a maximum stress in the pipe that is about 93% of the code minimum yield strength (35,000 psi) for the A106 grade B pipe used in the test. However, since tensile tests conducted on material from the pipe after the test was completed gave yield strength values between 40,000 and 45,000 psi, the maximum stress in the pipe during the test was actually less than 90% of yield. Failure, as defined by a leaking crack in the tee, occurred after 2875 cycles.

The results from all the fatigue tests are summarized in Table 8. For the fatigue bending tests of the 12-in. tees the actual number of

Table 8. Low-cycle fatigue failure results for
12- and 24-in. ASA B16.9 tees

Tee ^a	Loading condition ^b	Internal pressure	Cycles to failure ^c
T-4	Out-of-plane bending - branch	1925 psi constant	2,062
T-6	Out-of-plane bending - branch	1925 psi constant	1,309
T-7	In-plane bending - branch	3240 psi constant	11,475
T-8	Out-of-plane bending - branch	950 psi constant	8,970
T-15	Out-of-plane bending - branch	950 psi constant	10,200
T-10	In-plane bending - branch	1025 psi constant	18,532
T-11	Cyclic internal pressure	100 to 7000 psi	2,875

^aThe first five tees are 12 in. and the other two are 24 in. nominal diameter. See Table 6 for further identification.

^bThe maximum equivalent elastic stress for the bending moment loadings on the branch was $\pm 83,500$ psi for all five tees.

^cThe target failure for the bending tests was 7000 cycles; the average number of cycles for the 12-in. tees was 6803.

cycles to failure ranged from 1309 for T-6 to 11,475 for T-7, with an average value of 6803 cycles. The average value is quite close to the expected value of 7000 cycles even though the fatigue life of T-6 was only about 20% of that expected from Markel's correlations. On the other hand, the fatigue life of the single 24-in. tee (T-10) tested under bending load (18,532 cycles) was over 2 1/2 times that expected from Markel's correlation.

The fatigue life of all seven tees was greater than the code-allowable design life as calculated by the design procedures specified in paragraph NB-3653.6 of ASME-III,⁷ even though the cyclic pressure test conditions for T-11 were considerably more severe than would have been allowed under the complete set of code design rules. Table 9 compares each tee as to the fatigue life N_t determined in the test; the code-allowable design life N_c calculated by NB-3653.6, including the present code values for the stress indices; and an expected design life N_e calculated by the procedures of NB-3653.6 but using the normalized stress intensities of

Table 9. B16.9 tee fatigue test results and calculated allowable design cycles

Tee	N_t^a	N_c^b	N_t/N_c	N_e^c	N_t/N_e
T-4	2,062	38	54	40	51
T-6	1,309	38	34	40	32
T-7	11,475	110	104	70	163
T-8	8,970	16	560	70	128
T-15	10,200	18	566	70	145
T-10	18,532	15	1235	40	463
T-11	2,875	550	5	155 ^d	18

^a N_t is the number of cycles at which failure occurred in the test.

^b N_c is the code-allowable design life based on paragraph NB-3653.6 and the stress indices of Table NB-3683.2-1 of ASME-III-1971 (Ref. 7).

^c N_e is the expected design life based on the procedures of NB-3653.6 and the maximum elastic stress amplitude imposed during the tests, that is, using the normalized stress intensities of Table 7 instead of the code values for the stress intensities (K_2 was taken as 1.0).

^dThis value of N_e for the cyclic pressure test of T-11 was calculated using stress index values of $C_1K_1 = 3.69$ from Table 7 and $K_1 = 1.2$ from the photoelastic tests of T-7 and T-9. If K_1 had been taken as 1.0, then N_e would have been 80; if $K_1 = 1.5$, then $N_e = 350$.

Table 7 instead of the stress indices of the code.* The ratios of N_t/N_c and N_t/N_e can be taken as safety factors for the two different methods of calculating the design life.

If one arbitrarily assumes that a safety factor of 20 on the allowable design cycles is adequate to assure that the component would have performed satisfactorily in service, then all the tees except T-11 met

*For the moment loading tests, this is equivalent to setting S_n and S_p from the code equations (NB-3653) equal to the equivalent elastic stress range ($2 \times 83,500$ psi) of the fatigue tests.

this criterion. The relatively low value of $N_t/N_c = 5$ as well as the photoelastic test results for pressure loading discussed in the next section strongly suggest that changes are needed in the code values for the stress indices C_1 and K_1 for B16.9 tees. More appropriate values are reflected in the calculated value of N_e for T-11 given in Table 9. Since we have at present only one fatigue data point, firm recommendations cannot be made; however, it seems clear that the code value of $C_1 = 1.5$ is too low and that $K_1 = 4.0$ is too high.

A more complete discussion on comparisons between experimental fatigue test data and code rules for design evaluation is given in the study by Rodabaugh and Moore,⁴⁴ which includes all the available fatigue test data for piping components. At the time this study⁴⁴ was completed, we had only completed five of the seven tee tests reported here; test results from T-6 and T-11 were not included. However, the conclusions for piping components in general and for forged tees (ASA B16.9) in particular remain valid; that is, the present code method is conservative even though changes in some of the stress indices may be required. Furthermore, any changes in the code stress index values for fatigue evaluation should be based largely on experimental data of the type being developed in the ORNL piping program.

Photoelastic Model Studies

Photoelastic model studies of the four 12-in. tees listed in Table 6 (T-6 through T-9) were completed during the past year under subcontract at Westinghouse Research Laboratories, and the final report⁴⁵ was published. One-half-scale epoxy models of the 12-in. prototype tees were machined from blocks of epoxy on a contour milling machine with very carefully constructed half-scale wooden patterns used as templates. The models were loaded with either internal pressure or out-of-plane bending moment, as indicated in Table 6, and were analyzed by the frozen-stress technique. Figure 9 shows the scale model of tee T-7 as it appeared before the internal pressure test, and Fig. 10 shows the scale model of T-8 mounted in the loading frame with an out-of-plane bending load being applied on the branch. All four models were cured under load at 166°C in

Fig. 9. One-half-scale photoelastic model of B16.9 tee model T-7 ready for testing under internal pressure loading.

order to freeze the stress patterns into the epoxy before the models were sliced and analyzed in a polariscope.

The principal objective of this investigation was to determine and analyze the stress distributions through the wall thickness at positions of maximum surface stresses. This information is needed to provide guidance for separating the maximum stresses into the three categories (primary, secondary, and peak) used by the nuclear vessel and piping codes to fix allowable stresses. Surface stresses were first determined from slices cut from one quadrant of each of the models at angular locations 22.5° apart around the branch opening. Angular positions $\varnothing$ around the branch were measured from the longitudinal plane in the clockwise direction. For the bending test models (T-6 and T-8), the $\varnothing = 0^\circ$ position was toward the free end of the run and $\varnothing = 180^\circ$ was toward the fixed end.

Fig. 10. One-half-scale photoelastic model of B16.9 tee model T-8 in the loading frame with an out-of-plane moment load on the branch pipe extension.

For both models the 180 to 270° quadrants, corresponding to the tension side of applied bending moment, were selected for detailed analysis. For the pressurized models (T-7 and T-9), the $\theta = 180^\circ$ positions on the models correspond with the fixed end for the prototype tests which were conducted at SwRI. For these models the 0 to 90° quadrants were selected for detailed analysis. Radial slices were also removed from each of the four models at $\theta = 0, 90, 180$, and 270° around the branch. Slices taken from the photoelastic models provide data corresponding to data from lines of strain gages on the prototype tees tested at SwRI as shown in Fig. 8 for T-8.

By studying the surface stress distributions, we chose three locations per model for stress measurements along lines through the wall thickness that are normal to the midsurface. Normal stresses and shear stresses referenced to the plane of the slices as well as the three principal stresses were determined. At each section through the wall thickness, the average membrane stress, the secondary bending stress, and the peak stress components were also determined. The secondary bending stress component was defined as the maximum difference between the average stress and a hypothetical equivalent linear distribution having the same first moment about the average as the measured distribution. The peak stress component was defined as the maximum difference at either the outside or the inside surface between the measured distribution and the equivalent linear distribution.

For the internal pressure tests on T-7 and T-9, the normalized stress intensity curve, $S_{\max} = (\sigma_1 - \sigma_3)/p$, was used to obtain the average, secondary, and peak stress components, where σ_1 and σ_3 are the maximum and minimum principal stresses, respectively, and p is the internal pressure. Models of T-6 and T-8 were each subjected to an out-of-plane bending moment applied to the branch. For these tests the directed stress components either normal or tangential to the plane of the slice were used, depending on which curve had the greatest magnitude. The bending test results were normalized to the maximum bending stress in the branch pipe,

$$(\sigma_{\text{nom}})_b = \frac{32M d_o}{\pi(d_o^4 - d_i^4)},$$

where M is the applied bending moment, d_o is the outside diameter of the branch pipe, and d_i is the inside diameter.

Figure 11 is a plot of the normalized principal stresses through the crotch region of T-7 in the longitudinal plane of the tee ($\theta = 180^\circ$). For this case the maximum principal stress σ_1 was in the circumferential direction, and the minimum principal stress σ_3 was in the radial direction throughout the wall thickness. Figure 12 shows the variation of the normalized stress intensity at the same point along with the average, linear bending, and peak components.

Figure 13 shows the variations of the normalized stress components σ_y , σ_z , τ_{yz} , and τ_{zx} through-the-wall thickness (directed relative to the plane of the slice) for the out-of-plane bending test of T-6. For this tee the maximum surface stress intensity, $S_{\max} = 3.03\sigma_{\text{nom}}$, was compressive and was located in the 225° section at the outside surface. Because of the complicated manner in which the stress intensity varies through the

Fig. 11. Normalized principal stress distributions through the wall thickness in the crotch region of the longitudinal plane ($\theta = 180^\circ$) for the photoelastic model of tee T-7 due to internal pressure loading.

Fig. 12. Normalized stress intensity distribution through the wall thickness and corresponding stress category values for the photoelastic model of tee T-7 due to internal pressure loading.

wall thickness for the bending tests, the average, secondary, and peak stress components were determined using the directed stress curve with the largest absolute stress value, which in this case was a circumferential stress σ_z . The component values for T-6 are shown in Fig. 14, which also shows that the stress distribution was essentially linear through the thickness.

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Fig. 13. Normalized maximum stress ratios as a function of distance through the wall thickness for the photoelastic model of tee T-6 due to an out-of-plane bending moment on the branch.

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Fig. 14. Normalized maximum circumferential stress distribution through the wall and corresponding stress category values for the photo-elastic model of tee T-6 due to an out-of-plane bending moment on the branch.

Table 10 gives the maximum normalized surface stress intensities found anywhere on the models (without regard to location), and the average, bending, and peak stress components as determined from the through-the-wall thickness measurements at the point of maximum surface stress. The internal pressure tests on the models of T-7 and T-9 and the out-of-plane bending test on T-6 each gave maximum surface stresses on the inside surface. These were located in the crotch region in the longitudinal plane of the tee for T-7 and T-9 and about 45° from the fixed end ($\beta = 225^\circ$) for T-6. In the out-of-plane bending test for T-8, the maximum surface stress was on the outside surface about 11° from the transverse plane of the tee toward the fixed end ($\beta = 259^\circ$). The stress distributions through the wall for both bending tests were very nearly linear at all locations where measurements were made, whereas the corresponding distributions for the internal pressure tests were quite nonlinear. This is reflected by the relative difference in normalized peak stress components given in Table 10.

Table 10. Maximum normalized^a surface stress intensities and stress-category components from photoelastic tests of ASA B16.9 tees

Tee	Loading	Maximum surface stress intensity	Average component	Bending component	Peak component
T-6	Out-of-plane bending	3.03 ^b	0.43	2.23	0.08
T-7	Internal pressure	15.34	6.10	5.40	3.84
T-8	Out-of-plane bending	1.82	0.44	1.26	0.11
T-9	Internal pressure	14.9	7.28	5.23	2.39

^aSee text for explanation of the normalizing procedures used in obtaining these values.

^bThis particular value is slightly higher than the sum of the three components given for this tee (2.74) since the position of the through-the-wall-thickness measurements was slightly displaced from the maximum surface stress location.

Experimental Thermal Stress Analysis

In addition to the experimental studies for examining mechanical load responses, stresses due to thermal loadings are also being investigated. Two of the tees, T-6 and T-3, made from carbon steel and stainless steel respectively, were selected for thermal stress tests at Oak Ridge (Table 6). These tees were obtained from the same manufacturer and were made by the same process as those which were strain gaged and fatigue tested at SwRI. The physical dimensions were almost identical for both sets of tees. The objective of these tests is to provide experimental data on thermal stresses and strains that can be compared with finite-element elastic analyses and used as a basis for evaluating the thermal stress capabilities of the computer code discussed in the next section. We are also interested in whether or not significant thermal stresses can be induced in the tees by relative temperature differences which might be encountered in water-cooled nuclear power plant service. Thermal stresses are to be induced in the tees by developing temperature gradients between one of the outlets and the other two with heating and cooling coils located on pipe extensions welded to the tees. Maximum temperature differences of up to 400°F are planned. Temperatures and strains will be measured and used to compute the thermal stresses.

Test assemblies were fabricated for both tees by welding short pipe extensions of the same material and nominal size, about 2 ft long, to each of the outlets. The carbon-steel tee, T-6, was instrumented on both the inside and the outside surface with 0.125-in. gage-length high-temperature (600°F maximum) 45° three-gage strain rosettes. Thermocouples were also attached at the same locations and at additional positions on the model to measure temperature distributions. Heating coils were wrapped around the pipe extensions, and the model was insulated to minimize temperature gradients through the wall thickness. Figure 15 shows the insulated model resting in the support frame before the instrument lines were connected to the data-acquisition system. The entire test assembly was enclosed in a steel box which was filled with argon gas to prevent corrosion of the strain gages, and the first series of tests was started.

Fig. 15. Thermal stress test assembly for B16.9 tee TT-6 after installation of external heaters and insulation.

The purpose of the first series of tests was to obtain temperature calibration curves for the strain gages while the tee was heated uniformly (no temperature gradients) in steps of 50°F from room temperature to the maximum test temperature of about 400 to 450°F. About halfway through the heating cycle, data obtained from several of the strain gages indicated that some type of failure had occurred in the data-acquisition circuits. The situation rapidly deteriorated until only about one-half of the gages were operational during cooldown. Investigations showed that most if not all of the strain-gage whisker wires, which connect the active portion of the gage to the instrument lead wires, had become embrittled and many had broken. The exact cause of the embrittlement has not been determined; however, since these wires are an integral part of the strain-gage rosettes, it is clear that the tee must be reinstrumented before the tests can be resumed. We are presently developing new procedures for the strain-gage instrumentation and plan to complete the tests during FY 1973.

Finite-Element Elastic Analyses

Concurrently with the experimental studies of tees discussed in the preceding sections, finite-element computer code development is under way at the University of California, Berkeley. The objective of this work is, in broad terms, to develop suitable analytical techniques to be used for conducting parameter studies over a wide range of sizes and wall thicknesses. The results obtained will then be used to provide stress indices which may be included in codes and standards and used in the design of piping systems. For this purpose the method of finite-element numerical analysis was selected as being the most promising, because of the relatively complex external shape and wall thickness variations of ASA B16.9 tees.

Even though the finite-element method is being used extensively in structural design analysis, application of the method is not a routine matter, and a number of theoretical and practical questions required resolution before a completely satisfactory computer program could be written

for the analysis of ASA B16.9 tees. The scope of the University of California work includes the following:

1. selection of an optimum finite-element description for the analysis of ASA B16.9 tees,
2. determination of the best procedure for subdividing the structure into finite elements,
3. definition of the geometry of the structure in terms of a small number of parameters and developing of procedures for generating the finite-element input data automatically,
4. detailed stress analysis of three tees (T-6, T-7, and T-8) for the same 13 loading conditions used in testing the tees at SwRI (pressure and three mutually perpendicular force and moment loads applied through the branch and run pipes with one end of the run pipe rigidly fixed),
5. presentation of the computed stresses in a convenient form for interpretation,
6. development of thermal stress analysis capability for temperature gradients which may vary both through the wall thickness and over the surface,
7. development of an efficient production-type computer program suitable for conducting parameter studies at reasonable cost.

In addition to these items we have also asked the University of California to develop a suitable means for calculating both steady-state and transient temperature solutions which can be used as input for the thermal stress analysis of tees. Our present plan is to write a separate computer program which will calculate temperatures at discrete node points and time intervals (for the transient case), print out the results for examination by the analyst, and save the temperatures on some suitable storage device (tapes, disk, etc.) for later use in the stress analysis program. The finite-element mesh for the thermal analysis program will be generated automatically by the same input routines used in the stress analysis program, but it may contain additional node points in order to obtain a more refined temperature distribution. The stress analysis program will then select only the needed temperature data from the storage device as input. Since many of the subroutines already developed for the stress analysis

program can be used in the thermal analysis program, we expect to have this capability developed by July 1972.

In regard to item 1, several possible finite-element descriptions exist, including the flat-plate elements used for the analysis of the thin-shell idealized tees discussed earlier.¹⁷⁻¹⁹ From the time the project was initiated, however, it was expected that three-dimensional elements would be needed for the analysis of the relatively thick B16.9 tees in order to obtain reasonably good agreement with the experimental results. Both 8- and 20-node isoparametric hexahedral elements were under consideration, and at this time last year we had selected a modified version of the Zienkiewicz and Irons 8-node brick element,⁴⁶ denoted as ZIB8R5, which has five additional degrees of internal freedom. All the various elements which were considered and the background work leading to this choice were reported⁴⁷ at the ASME Pressure Vessels and Piping Conference held in San Francisco in May. Later work, however, showed that the ZIB8R5 element was too stiff in bending. This led to selection of the ZIB8R9 element with nine additional degrees of internal freedom, as reported⁴⁸ at the First International Conference on Structural Mechanics in Reactor Technology held in Berlin, Germany, in September. This element has been incorporated into the B16.9 tee stress analysis program, and several studies have been conducted. Figures 16 and 17 show comparisons between the finite-element results and experimental results for tee T-8 (12 x 12 x 6 in., sched 40) loaded with internal pressure and an in-plane bending moment applied to the branch pipe respectively.

Some of the more difficult aspects of the B16.9 tee stress analysis computer program work were the development of a suitable analytical description of the shape of B16.9 tees, suitable procedures for dividing the structure into suitable finite elements, and the automatic mesh generation subroutines needed to conduct parameter studies effectively. The manufacturing standards, such as ASA B16.9 and MSS-SP-48 (Refs. 23, 34), offer very little dimensional information which can be used to describe the geometry. Plastic model replicas⁴⁹ of the tees tested at SwRI proved invaluable for this purpose.

Detailed dimensional measurements of these models as well as dimensional information from tees which have been tested were used to help

FINITE ELEMENT RESULTS
 —●— MANUALLY DEVELOPED MESH
 - - -○- - AUTOMATICALLY DEVELOPED MESH
 EXPERIMENTAL RESULTS
 ▲ STRAIN-GAGE DATA

Fig. 16. Maximum principal stresses from finite-element calculations compared with the experimental elastic response test results from B16.9 tee T-8 loaded with internal pressure.

Fig. 17. Maximum principal stresses from finite-element calculations compared with the experimental elastic response test results from B16.9 tee T-8 with an out-of-plane bending moment on the branch.

develop a suitable means for analytically describing both full outlet and reducing outlet ASA B16.9 tees of different sizes and nominal wall thicknesses. For automatic mesh generation the assumption was made that B16.9 tees are made up of three shallow cones joined by a circular fillet transition piece whose radius varies linearly around the branch cone outlet. Both the cone angle and the radius of the transition are input parameters. Other parameters define the wall thickness and the amount of taper at the pipe-to-tee welds. The computer will also simulate tees fabricated by machining operations such as boring the inner surfaces of the branch and run outlets. Figure 18 shows the computer-developed finite-element mesh

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Fig. 18. Computer-generated finite-element mesh layout for the stress analysis of B16.9 tee T-7.

for the outer surface of tee T-7, a 12 x 12 x 12 in., sched-160 tee. Note especially the significant amount of taper at the welds. Since the wall thickness of this particular tee was considerably greater than that of the matching sched-160 pipe, a significant amount of machining was performed by the manufacturer both at the weld preps and in the core in order to meet the dimensional requirements of the B16.9 standard.

The computer program performs symmetric and antisymmetric analyses using a 1/4 section of the model such as shown in Fig. 18. These separate analyses are then combined by linear superposition as required for treating the various loading conditions specified by the user. Thus, in addition to the 13 basic loading conditions, it is also possible to obtain results for any arbitrary combination of these loads. The complete computer program has now been assembled, including all the service function subroutines and the automatic input preparation and stress output packages. The graphic output packages will, however, be developed at ORNL after we receive the program from the subcontractor since the graphics software is computer system dependent. Final check-out of the computer code is under way, and we expect to receive it about July 1972.

LIMIT LOADS

This task (6) of the piping program contains a number of studies whose main purpose is to determine a lower bound for the maximum load which a given piping component can support. This information is needed to help establish allowable limits for the primary category stresses as defined by the design codes and standards. Both experimental and analytical limit-load studies are presently planned under this task. The experimental studies include determining limit loads for elbows, tees, and other components such as straight pipe, welds, and reducers, although detailed plans have not been made for the latter. The analytical studies include an elastic buckling study for straight pipe, the development of classical limit-load analyses, and elastic-plastic studies of piping components under various loading conditions. Two studies completed during the past year are discussed below.

Limit-Load Studies for ASA Standard Elbows

Last year we reported⁸ the results of experimental studies conducted to determine the plastic collapse or limit load for a series of 14 commercial 6-in. (nominal diameter) carbon-steel elbows and one 6-in. stainless steel elbow. These studies were conducted primarily to test the validity of the design values for the primary stress index B_2 (Table NB-3683.2-1, footnote 8, Ref. 7) for curved pipe or butt-welding elbows that meet the requirements of standards ASA B16.9, ASA B16.28, or MSS-SP-48 (Refs. 23, 24, 34). The test assembly for each elbow was made by welding pipe extensions of the same material and the same nominal wall thickness to the ends of the elbows. One extension was rigidly mounted on a pedestal attached to a loading frame, a normal force loading was applied near the end of the free extension (between four and five pipe diameters long), and load-deflection response curves were measured with the aid of dial indicators mounted on the free extension pipe 7 1/2 and 15 in. from the plane of the pipe-to-elbow weld. The results were interpreted in terms of classical limit analysis concepts and were used to determine the so-called collapse loads.

Significant variables considered in the study were wall thickness, bend radius, direction of applied loading, influence of internal pressure, and relative behavior of carbon-steel (ASTM A106, grade B) and stainless steel (ASTM A312, type 304L) elbows. Six of the carbon-steel elbows were sched-40 long-radius elbows (6.625 in. OD, 0.280 in. wall thickness, and 9 in. bend radius); three were sched-80 long-radius elbows (0.432 in. wall thickness); and five were sched-40 short-radius elbows (6 in. bend radius). The stainless steel elbow was sched 40 long radius. All of the elbows were 90° bends.

The results of the study, presented in a paper⁵⁰ last spring, are as follows.

1. Classical limit-load concepts can be used to describe the post-yield behavior of piping components such as elbows.
2. For an elbow of given wall thickness, radius, and material under external load alone, the collapse moment is smaller for an in-plane bending moment, which tends to close the elbow, than for either an in-plane

moment, which tends to open the elbow, or for an out-of-plane moment loading.

3. The addition of internal pressure in combination with the moment loading generally increases the collapse moment, although departure from linear load-deflection response (yielding) occurs earlier than for an elbow loaded only with a bending moment.

4. The ratio of the maximum stress at the limit load to the yield stress of the material increases with an increase in wall thickness.

5. The collapse load for the stainless steel elbow, tested with an in-plane bending moment that tended to close the elbow, was considerably less than that for the corresponding carbon-steel elbow tested in the same manner. In this case the collapse load of the stainless steel elbow was 56% of the collapse load for the carbon-steel elbow.

In order to confirm the latter conclusion, a second stainless steel elbow was tested. This model (PE-16) was made from the same lot of material as the first elbow (PE-15) and was tested in the same manner, that is, with an in-plane bending moment that tended to close the elbow. It was dimensionally inspected more thoroughly than PE-15 prior to testing, however, and instrumented more extensively during the test. Before testing, the ovality (out of roundness) of PE-16 was 1% at the ends and 2% at the midplane; after testing, the ovality was 5 1/2 and 11% at the ends and midplane respectively.

The test assembly for PE-16 is shown mounted in the loading frame ready for testing in Fig. 19. Ten three-gage strain rosettes, four dial indicator gages, and one load cell were used to monitor the strains, deflections, and the applied load respectively. Eight of the strain rosettes were located at the midplane of the elbow in the region of expected maximum stress, and one was mounted on either end at the pipe-to-tee welds. As shown in the figure, two of the dial indicators were mounted near the weldment joining the elbow and pipe extension to measure the vertical and horizontal displacements of the weld. The other two dial indicators were mounted at 7 1/2 and 15 in. from the weld to measure vertical displacements of the pipe extension. We were thus able to use the angular rotation of the elbow in addition to the pipe displacements in determining the collapse load as was done for the previous test of PE-15.

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Fig. 19. Test assembly for the limit-load test of elbow model PE-16 mounted in the loading frame.

The entire outside surface of PE-16 was coated with a brittle lacquer which had a tensile strain sensitivity of 850 μ in. During the test, cracks first appeared in the coating in the midplane (45°) of the elbow at the sides where three of the strain rosettes had been located (see Fig. 19), thus confirming the adequacy of the strain-gage data for determining the collapse load. For a more detailed discussion on the methods of using strain-gage and displacement data for determining the collapse load, see Ref. 50.

This second test gave essentially the same collapse load as the first; that is, it confirmed the earlier indication that the type of material is an important variable in governing the plastic collapse of structures and should be considered in design rules. In terms of the B_2 stress index, the study indicated that for all the 6-in. elbows tested the present code values are conservative and may be more conservative than necessary for the carbon-steel elbows. However, even though the present code formula for B_2 is adequate for the stainless steel elbow, additional information is needed before changes in the code design rules can be recommended. Both experimental and analytical studies are planned for the coming year to provide this information.

Elastic Buckling of Straight Pipe

Elastic buckling studies of straight pipe heated uniformly and restrained from axial expansion were completed, and a report⁵¹ has been prepared for publication. This study was undertaken to develop convenient and accurate methods for calculating the critical temperature rise T_c at which elastic buckling of a long segment of straight pipe would occur as well as methods for calculating the maximum stresses at temperatures below and above T_c . Under certain conditions the critical temperature for Euler-type column buckling will occur even though the nominal stress values in the pipe are well below code-allowable design values. If the temperature is increased further the maximum stresses will increase rapidly because of bending stresses induced in the postbuckled condition.

In the report, a number of practicable design considerations are also discussed including the effects of internal pressure, hanger spacing, and initial geometric imperfections. The results of this work are expected to be useful in assessing accident situations in which straight lines of pipe are heated above the normal design conditions.

CONCLUSIONS

With the projects completed during the past year, work under the ORNL piping program is essentially 50% complete. Table 11 shows the current status (February 1, 1972) of each of the studies originally outlined in the program plan. The literature survey⁵² under Task 1 was completed and reported last year, and the two remaining studies are about 75% complete. We plan to publish these reports during the coming year.

A significant amount of work was completed on the Task 2 studies for straight pipe, in-line configurations, and attachments. Much of this work either has been adopted or is being considered for adoption by the ASME code committees. Notable are the design rules for calculating stresses in straight pipe with initial out of roundness, the extensive changes proposed for design code stress indices for girth-welded joints and tapered transitions, and the proposed stress indices for socket-welding and threaded fittings. Final reports for these studies, as well as for the concentric reducers study⁵³ discussed last year, are being prepared and will be published during the next few months. Work on flanged joints was initiated during the past year and is scheduled for completion during FY 1973.

Experimental and analytical studies outlined for thin-walled idealized tees and small branch connections under Task 3 are about 50% completed. Two of the four thin-walled tee models have been tested and analyzed, and the other two models are being instrumented for testing during the coming year. Significant results from the two completed model studies were reported in the open literature,^{17,19} and a more detailed account has been prepared for publication as an ORNL report.¹⁸

Table 11. ORNL piping program tasks, major subtasks, and status^a

Task and subtask ^b	Percentage completion				
	0	25	50	75	100
Task 1. INFORMATION SURVEYS					
1.1 Literature Survey					
1.2 Dimensional Study of Piping Components					
1.3 Survey of Class I Piping System Components					
Task 2. STRAIGHT PIPE, IN-LINE CONFIGURATIONS, AND ATTACHMENTS					
2.1 Straight Pipe and Auxiliary Attachments					
2.2 Socket-Welding and Threaded Fittings					
2.3 Concentric and Eccentric Reducers					
2.4 Transition Joints					
2.5 Flanged Joints					
Task 3. THIN-WALLED TEES AND BRANCH CONNECTIONS					
3.1 Idealized Thin-Walled Tees ^c					
3.2 Small-Branch Connections					
Task 4. ELBOWS, CURVED PIPE, AND MITER BENDS					
4.1 Analytical Studies of Elbows and Curved Pipe					
4.2 Machined-Elbow Model Studies					
4.3 Elastic Stress Analysis of Miter Bends					

Discontinued

Table 11 (continued)

Task and subtask ^b	Percentage completion				
	0	25	50	75	100
Task 5. ASA STANDARD B16.9 TEES					
5.1 Elastic Response and Fatigue Studies					
5.2 Photoelastic Model Studies					
5.3 Experimental Thermal Stress Analysis					
5.4 Finite-Element Computer Code Development					
5.5 Finite-Element Stress Analyses	Not started				
5.6 Transient Temperatures and Fluid Flow in Mixing Tees	Not started				
Task 6. LIMIT LOADS					
6.1 Experimental Limit Loads for ASA Standard Elbows					
6.2 Experimental Limit Loads for ASA Standard Tees					
6.3 Analytical Limit Load Development					
6.4 Limit Loads for Other Piping Components	Not started				
6.5 Elastic Buckling of Straight Pipe					

^aFebruary 1, 1972.

^bFor additional detail, see Program Plan for the ORNL Program - Design Criteria for Piping, Pumps, and Valves.

^cJointly sponsored by the ORNL Piping Program and other ORNL programs; fully supported by other ORNL programs after FY 1971.

Flexibility factors for small branch connections ($d/D \leq 1/3$) were developed for use in codes and standards, and a special code case is being prepared for consideration by the ASME code committee. These factors were based on a review of published experimental data, the well-known analyses of Bijlaard, and results from analytical studies of unreinforced cylinder-to-cylinder models conducted under this program. Our study showed that the present code formula is restrictive for certain nozzle configurations permitted by the code; hence the new formula will tend to ease these restrictions. We plan to complete the remaining studies outlined under Task 3 - reinforced branch connections - following development of the three-dimensional finite-element computer code which is being written for the analysis of B16.9 tees.

Task 4 studies on elbows, curved pipe, and miter bends are about 40% complete. An analytical study³⁰ on the development of stress indices and flexibility factors for elbows based on information published in the literature was completed and prepared for publication. The results of this study showed that the present code formulas are essentially correct if the discontinuity effects (end effects) of pipe or other components welded to the elbow can be ignored. Methods for including these effects in the analysis are currently being developed.

The experimental studies on four machined-elbow models also being conducted under Task 4 are approximately 25% complete. Tests were completed on model ME-1, and ME-3 which will be tested early in 1973 was partially instrumented. Comparisons between the experimental results and the analysis discussed above indicate that the maximum stresses were about 15% less than predicted. This is considered good, especially since the analysis is slightly conservative even though we are as yet unable to include end effects in the analysis. We plan to test the remaining three models during the coming year and to compare the experimental results with appropriate analyses.

In terms of man effort and cost, the B16.9 tee studies, which are grouped together under Task 5, are about 50% complete. As shown in Table 11, the experimental elastic-response and low-cycle fatigue tests are 50% complete. The photoelastic model studies conducted under subcontract at Westinghouse Research Laboratories were completed, and the final report⁴⁵

was published. The experimental thermal stress analysis tests are perhaps 25% complete in spite of a major problem with the high-temperature strain gages, and the finite-element computer code development work being done under subcontract at the University of California, Berkeley, is nearing completion.

We have completed our examinations of the elastic-response strain-gage data from 7 of the 17 B16.9 tees being tested in this series (Task 5.1) and have identified the maximum normalized stress intensities (experimental stress indices) for each of the 13 basic loadings. Processing the large amount of experimental data proved to be a major undertaking, thus prompting the development of logical diagnostic procedures⁴² which could be implemented on the computer. Elastic-response tests for four more 24-in. B16.9 tees are scheduled for completion by the end of FY 1973.

Low-cycle fatigue tests have also been completed under Task 5.1 for seven of the B16.9 tees, six of which were tested with an alternating bending moment on the branch pipe and one with a cyclic internal pressure. Comparisons⁴⁴ between the fatigue test data from these tees and the fatigue analysis procedures of the ASME Boiler and Pressure Vessel Code, Section III-1971 (Ref. 7), showed that in every case the present code procedures are conservative, even though some of the code stress index values may not be conservative for certain loading conditions (e.g., internal pressure). Fatigue data obtained from the literature for other piping components as well as for tees indicate that the basic code procedures for fatigue evaluation are conservative. The data also suggest that any changes in the code stress index values for fatigue evaluation should be based primarily on fatigue data obtained from actual piping components tested to failure.

Data obtained from the four photoelastic model studies⁴⁵ of B16.9 tees (Task 5.2) completed during the past year at Westinghouse Research Laboratories showed that: (1) the stress distributions through the wall thickness of heavy walled tees (sched 160) loaded with internal pressure are quite nonlinear and (2) the through-the-wall-thickness stress distributions for thinner-walled tees (sched 40) loaded with a bending moment on the branch are very nearly linear. Through-the-wall thickness stress distributions were determined at three positions on each of the four

models tested, and primary, secondary, and peak stress components as defined by the codes were identified for each position. As expected, the peak stress component was very small for the bending tests and rather significant for the pressure tests. For the pressure loading tests, however, the peak stress component was not as large as indicated by the present code stress index values.

Finite-element computer code development is progressing smoothly under subcontract at the University of California, and we expect to receive the completed program by July 1972. Much of the effort during the past year has been devoted to developing efficient means for automatically generating the large amount of input required by the code. This has now been accomplished, and the entire program has been assembled and successfully operated for several test cases. We have also asked the University of California to develop a thermal analysis program which will calculate temperature data for semiautomatic input to the stress analysis program. We expect to receive this program as well by July 1972.

Two important studies were completed under Task 6 on limit-load studies: experimental limit-load studies for elbows⁵⁰ and analytical elastic buckling studies for straight pipe which is restrained against thermal expansion.⁵¹ An open-literature publication was prepared for the elbow studies, and an ORNL report is ready for publication for the elastic buckling studies.

Very good overall progress was made under this program during the past year, and a significant amount of work was completed. Moreover, the results obtained thus far are rapidly being adopted for use in the national codes and standards for the design of nuclear reactor piping systems.

PART II. ORNL NOZZLES ANALYSIS PROGRAM — EXPERIMENTAL AND ANALYTICAL INVESTIGATIONS OF NOZZLES

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PROGRAM DESCRIPTION

Analytical and experimental investigations of the stress distributions near the intersection of nozzles with pressure vessels are being conducted to develop satisfactory design procedures for nuclear reactor pressure vessels. This program is concerned with both the development of methods of analyses and the generation of highly reliable experimental data to be used to verify these analyses. Geometries being investigated include single nozzles attached radially and nonradially to cylindrical and spherical shells and clusters of nozzles or holes in flat plates, cylinders, and spheres.

The loading conditions under study include axial thrust, bending moments, and torsion, all applied to the nozzle, and internal pressure. A series of parameters have been selected to represent the various possible nozzle and vessel geometries. Included are the nozzle diameter-to-thickness ratio d/t , the shell diameter-to-thickness ratio D/t , the nozzle-to-shell membrane hoop stress ratio s/S , the length of extension of a nozzle inside the vessel, and the angle of inclination of the nozzle to the vessel. The nozzle cluster studies also consider the number, spacing, and geometric pattern of nozzles.

The work is divided, for administrative purposes, into the AEC(ORNL) program concerned primarily with multiple-nozzle connections and a joint program between the AEC and industry which is concerned primarily with single nozzle connections. The joint program is carried out under the auspices of the Pressure Vessel Research Committee (PVRC) of the Welding Research Council, and the AEC-financed portion is coordinated by ORNL through the PVRC Subcommittee on Reinforced Openings and External Loadings. Much of the research is being done under subcontract to either ORNL or to the PVRC, depending on the source of funds. Table 12 lists projects currently under way and those recently completed under both programs.

In the coordination of these programs, ORNL has been delegated the tasks of (1) review and evaluation of the research, (2) direction of and participation in the AEC-sponsored work, (3) generation of analytical parameter studies in support of correlation and evaluation studies undertaken at Battelle Memorial Institute, (4) preparation of reports and recommendations to the Subcommittee on non-AEC-sponsored projects, and (5) solicitation of recommendations from relevant PVRC subcommittees on AEC-sponsored projects.

As part of this task, ORNL staff members attend the scheduled PVRC meetings of the Design Division and the Subcommittee on Reinforced Openings and External Loadings where recent progress is discussed. Project status was also reported at the ORNL-sponsored information meeting - Studies in Applied Solid Mechanics - held in Oak Ridge April 26-28, 1971. A description of the programs was given in a project report⁵⁴ (being revised) and in a more recent addendum.⁵⁵

The progress achieved during CY 1971 is discussed here. The work of the two programs is described, with the discussion in each section being organized according to study subject. For a complete listing of the work done by each subcontractor, the reader is referred to Table 12.

JOINT AEC(ORNL)-PVRC COOPERATIVE PROGRAM

The projects being carried out under this part of the overall program as well as the organizations involved are listed in Table 12. The

Table 12. Current experimental and analytical investigations of nozzle attachments

Program sponsor	Funding agency ^a	Subcontractor	Program
Joint AEC-FVRC	AEC	University of Tennessee I	Experimental stress analysis of single radial and nonradial nozzles attached to spherical shells subjected to internal pressure, thrust, shear, or moment loadings
	AEC	Auburn University II	Theoretical solution for single nonradial nozzles attached to a spherical shell at angles of obliquity less than 20° for internal pressure loading
	AEC	Oak Ridge National Laboratory	Expansion of present theoretical solution of single nonradial nozzle attached to a spherical shell to include angles of obliquity greater than 20° and external loads
	AEC	Westinghouse Research Laboratories	Photoelastic studies of radial and nonradial nozzles attached to spherical and cylindrical shells and closely spaced pairs of reinforced nozzles attached to cylindrical shells under internal pressure loading
	AEC	Oak Ridge National Laboratory	Experimental stress analysis of single nozzles attached to cylindrical shells under 13 loadings and development of analytical solutions for internal pressure and moment loadings
	FVRC, NRCC	University of Sherbrooke	Analytical methods development for calculating lower bounds to limit loads for cylindrical shells with radially and nonradially attached nozzles where the composite structure is subjected to internal pressure or external loadings on the nozzle or combinations of internal pressure and external loads; and use of resulting analyses in parameter studies
	NRCC		Analytical and experimental studies of stress-strain behavior and plastic redistribution of stresses around oblique holes in flat plates subjected to uniaxial tension, and experimental study of the effect of corner cut on elastic stress concentration for oblique holes in flat plates
	FVRC, NRCC	University of Waterloo	Analytical methods development for upper bounds to limit pressures, and moments applied either separately or in combination with internal pressure, to radial or nonradial nozzles attached to cylindrical shells and experimental studies to provide data for analysis verification
AEC(ORNL)	AEC	University of Tennessee II	Experimental stress analysis of flat plates with clusters of holes
	AEC	Auburn University I	Experimental stress analyses of spherical shells with clusters of holes and nozzles
			Experimental stress analyses of flat plates with clusters of nozzles
	AEC	Battelle Memorial Institute II	Theoretical analysis development for flat plates with clusters of holes
			Theoretical analysis development for flat plates with attached clusters of nozzles
Joint AEC-FVRC	AEC	Oak Ridge National Laboratory	Finite-element studies of reinforcement configurations for spherical shells and shell attachment regions
			Analyses of models using recently developed computer programs and comparisons between analytical and experimental results
			Parametric studies for program guidance and code use
	AEC	Battelle Memorial Institute I	Engineering evaluation and correlation of results for nozzle-to-shell attachments and preparation of material based on these results for use in support of ASME Code Committee work

^aNRCC is National Research Council of Canada.

Current experimental and analytical investigations of nozzle attachments

Subcontractor	Program	Status
University of Tennessee I	Experimental stress analysis of single radial and nonradial nozzles attached to spherical shells subjected to internal pressure, thrust, shear, or moment loadings	In progress
University II	Theoretical solution for single nonradial nozzles attached to a spherical shell at angles of obliquity less than 20° for internal pressure loading	Completed
National Laboratory	Expansion of present theoretical solution of single non-radial nozzle attached to a spherical shell to include angles of obliquity greater than 20° and external loads	In progress
Case Research Laboratories	Photoelastic studies of radial and nonradial nozzles attached to spherical and cylindrical shells and closely spaced pairs of reinforced nozzles attached to cylindrical shells under internal pressure loading	Completed
National Laboratory	Experimental stress analysis of single nozzles attached to cylindrical shells under 13 loadings and development of analytical solutions for internal pressure and moment loadings	In progress
University of Sherbrooke	Analytical methods development for calculating lower bounds to limit loads for cylindrical shells with radially and nonradially attached nozzles where the composite structure is subjected to internal pressure or external loadings on the nozzle or combinations of internal pressure and external loads; and use of resulting analyses in parameter studies	In progress
	Analytical and experimental studies of stress-strain behavior and plastic redistribution of stresses around oblique holes in flat plates subjected to uniaxial tension, and experimental study of the effect of corner cut on elastic stress concentration for oblique holes in flat plates	Completed
University of Waterloo	Analytical methods development for upper bounds to limit pressures, and moments applied either separately or in combination with internal pressure, to radial or nonradial nozzles attached to cylindrical shells and experimental studies to provide data for analysis verification	In progress
University of Tennessee II	Experimental stress analysis of flat plates with clusters of holes	Completed
University I	Experimental stress analyses of spherical shells with clusters of holes and nozzles	In progress
	Experimental stress analyses of flat plates with clusters of nozzles	In progress
Memorial Institute II	Theoretical analysis development for flat plates with clusters of holes	Completed
	Theoretical analysis development for flat plates with attached clusters of nozzles	In progress
National Laboratory	Finite-element studies of reinforcement configurations for spherical shells and shell attachment regions	In progress
	Analyses of models using recently developed computer programs and comparisons between analytical and experimental results	In progress
	Parametric studies for program guidance and code use	In progress
Memorial Institute I	Engineering evaluation and correlation of results for nozzle-to-shell attachments and preparation of material based on these results for use in support of ASME Code Committee work	In progress

Canada.

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analytical studies are of two general types: those for analytical methods development and verification and those for support of structural design code development. The results of the latter are used to verify acceptability of present rules, to provide bases for drafting new rules, and to provide data correlations for direct use in design. During the past year, several analytical studies were carried out jointly at ORNL and at Battelle Memorial Institute in support of code development work.

In addition to the analytical studies, experimental investigations to examine elastic responses of nozzles attached to cylindrical and spherical vessels were continued. Some of the planned studies were completed, and significant progress was made on others. Although this portion of the total program is generally concerned with single nozzle attachments, limited studies on closely spaced, fully reinforced nozzles attached to cylindrical shells were defined, and the associated experiments were completed.

The work associated with developing suitable analytical methods for nozzles nonradially attached to spherical shells and for single-nozzle to cylindrical-shell attachments continued. In the latter case, a two-pronged attack is being maintained by continuing work on closed-form-type solutions to the governing differential equations and by continuing the development of finite-element analysis capabilities. The steel model tests and finite-element analysis work being carried out at ORNL on nozzle-to-cylinder attachments were described in Part I and will not be discussed further here.

The topics discussed thus far have been related to elastic-response-type studies. Plastic collapse, or limit-load, studies are also included under the joint program to provide information for use in establishing design rules to limit total deformation. Presently, methods for calculating upper- and lower-bound limits to collapse pressures and externally applied loads for single nozzles attached to cylindrical shells are being developed. Limit analysis methods for application to nozzles attached to spherical shells were developed earlier under this program. As can be seen from Table 12, the limit-load studies on cylindrical shells are conducted by treating the analytical work associated with upper and lower bounds as separate projects. In the case of upper bounds to limit loads,

both tees and laterals are under consideration and may be subjected separately to internal pressure and to external loads applied to the nozzles as well as to combinations of internal pressure and external load. Methods for calculating lower bounds are being developed for tees only. In addition to the analytical studies, plastic collapse tests are being conducted to provide experimental evidence required to assess the calculational methods.

A complementary study to the nonradial nozzle elastic-response investigations and the limit-load studies on laterals is a study of the stress distributions in flat plates with single hole perforations. The holes are skewed, that is, the direction of the center line of the hole in each case is not normal to the plane of the plate, and various angles of skewness are investigated. The plates are loaded by in-plane edge forces. This type specimen and loading provides important information concerning the behavior of the acute corners. In addition, insight regarding the validity of limit-load concepts as applied to laterals can be obtained. Of particular importance is the behavior at the acute inside corner in the crotch region as the load approaches that corresponding to plastic collapse.

The projects under the joint program are discussed in more detail below, and the progress during the past year is described.

Analytical Studies in Support of Structural Codes and Standards Development

During the past year three sets of parameter studies on single reinforced nozzles attached to spherical shells were pursued jointly by Battelle Memorial Institute (BMI-I) and ORNL. The reinforcement configurations considered were triangular pads, fillets, and combinations of inside and outside reinforcing. The scope of each study area and the results are described below.

Effects of fillet radii for triangular pad reinforcement

A report recently published by Battelle Memorial Institute⁵⁶ contains the results of an analytical study of the effect of fillet radii on maximum stresses for triangular pad reinforcement for nozzle/spherical shell attachments. The radii of the fillets at the nozzle and at the shell ends of the pad can have pronounced effects upon the magnitudes of the stresses due to internal pressure. Thus the magnitudes of these effects are examined in this report, which also contains rules for calculating fillet radii required for meeting a selected stress criterion and gives reinforcement zone limits. The general criterion used and recommended for determining fillet radii requirements was based on the adoption of a maximum allowable stress intensity that is dependent upon dimensions of the vessel.

A parameter study was conducted using 33 models with triangular pad reinforcement to determine the acceptability of proposed design rules for minimum fillet radii, denoted by r_3 and r_4 in Fig. 20. The models are listed in Table 13, where the symbols have the usual meanings (see Fig. 20), and the lower-case letters apply to the nozzle and the upper-case letters to the vessel. The symbol A denotes the cross-sectional area of the reinforcement. The limiting value (maximum) of the stress intensity used as the acceptance criterion is given by

$$\bar{\sigma}_{\max} = [2.4 - d/D]S ,$$

where $S = PD/2T$ and P is the internal pressure. This limit was taken as satisfactory because of agreement with results for fillet radius reinforcing. Kalnins' axisymmetric shell computer program⁵⁷ was used to analyze each of the models listed in Table 13. The analytical results are plotted in Figs. 21 and 22, which also show the limiting stress intensity curve.

A compilation of available experimental and analytical data is included in the report to substantiate the use of the simplified maximum stress intensity equation as a general upper bound for various nozzle-to-shell reinforcement configurations with values of D/T up to 250 and d/D up to 0.5. This bound can be applied so long as the prescribed minimum

Fig. 20. Typical model employed in parameter study of triangular pad reinforcement.

ORNL-DWG 72-1231

Fig. 21. Maximum stress intensities, nozzles in spheres with triangular pad reinforcing, $\theta = 30^\circ$.

ORNL-DWG 72-1232

Fig. 22. Maximum stress intensities, nozzles in spheres with triangular pad reinforcing, $\theta = 60^\circ$.

Table 13. Summary of models included in study of nozzles in spherical shells with triangular pad reinforcing

D/T	d/D	2A/dT	r_3/t		r_4/T		Model No.	
			$\theta = 30^\circ$	$\theta = 60^\circ$	$\theta = 30^\circ$	$\theta = 60^\circ$	$\theta = 30^\circ$	$\theta = 60^\circ$
10	0.08	0.819 ^a	1.29	1.82	0.333	0.167	30-1	60-1
	0.16	0.987	1.29		0.333		30-2	60 ^b
	0.32	0.947	1.29		0.605		30-3	60 ^b
	0.50	0.866					30 ^b	60 ^b
20	0.04	0.305 ^a	1.83	2.58	0.333	0.167	30-5	60-5
	0.08	0.997	1.83	2.58	0.333	0.167	30-6	60-6
	0.16	0.987	1.83		0.427		30-7	60 ^b
	0.32	0.947	1.83		0.855		30-8	60 ^b
	0.50	0.866					30 ^b	60 ^b
40	0.04	0.819 ^a	2.58	3.65	0.333	0.167	30-10	60-10
	0.08	0.997	2.58	3.65	0.333	0.167	30-11	60-11
	0.16	0.987	2.58	3.65	0.605	0.262	30-12	60-12
	0.32	0.947	2.58		1.208		30-13	60 ^b
	0.50	0.866	2.58		1.890		30-14	60 ^b
80	0.02	0.305 ^a	3.65	5.16	0.333	0.167	30-15	60-15
	0.04	0.999	3.65	5.16	0.333	0.167	30-16	60-16
	0.08	0.997	3.65	5.16	0.427	0.186	30-17	60-17
	0.16	0.987	3.65	5.16	0.855	0.372	30-18	60-18
	0.32	0.947	3.65		1.710		30-19	60 ^b
	0.50	0.866	3.65		2.670		30-20	60 ^b
160	0.02	0.819 ^a	5.16	7.31	0.333	0.167	30-21	60-21
	0.04	0.999	5.16	7.31	0.333	0.167	30-22	60-22
	0.08	0.997	5.16	7.31	0.605	0.263	30-23	60-23
	0.16	0.987	5.16	7.31	1.208	0.525	30-24	60-24
	0.32	0.946	5.16		2.420		30-25	60 ^b
	0.50	0.866	5.16		3.780		30-26	60 ^b
250	0.01	0.143 ^a	6.46	9.15	0.333	0.167	30-27	60-27
	0.02	1.000	6.46	9.15	0.333	0.167	30-28	60-28
	0.04	0.999	6.46	9.15	0.378	0.167	30-29	60-29
	0.08	0.997	6.46	9.15	0.756	0.328	30-30	60-30
	0.16	0.987	6.46	9.15	1.512	0.658	30-31	60-31
	0.32	0.947	6.46		3.024		30-32	60 ^b
	0.50	0.866	6.46		4.730		30-33	60 ^b

^a(d/D) $\sqrt{D/T}$ is between 0.1414 and 0.2828.

^bModel is geometrically impossible.

fillet radii given in the report⁵⁸ are used and the reinforcement areas conform to the rules proposed in Ref. 58.

Since a shell-type analysis was used for this study, some of the results for the relatively thick-walled shells may be only approximations; therefore, three of the models were also analyzed using Wilson's finite-element computer program.⁵⁹ Comparisons of the results for these analyses together with additional available information indicate that the

shell-type analysis appears to be valid for D/T values of 20 or higher. In any case, for D/T values of 10 or less its use is questionable.

Effects of fillet reinforcement under combined internal pressure and bending moment loadings

A second BMI report⁶⁰ summarizes work undertaken to develop a satisfactory method for assessing the capability of nozzles in pressure vessels, having external fillet reinforcing at the junction, to withstand simultaneous loadings of internal pressure and bending moments without exceeding code limits. The report includes results for bending moment loading of 42 reinforced nozzle/spherical shell attachments. Analyses for internal pressure loading of these models were reported previously.⁶¹

A typical model, shown in Fig. 23, consists of three basic parts: (1) a cylindrical shell with inside diameter d and wall thickness t ; (2) a toroidal shell with centers at P_1 and P_2 (Fig. 23) with one surface considered for analytical purposes to be at a distance of r_o from P_2 (the wall thickness of the torus ranged from t to t_m to T as shown); and (3) a spherical shell with inside diameter D and wall thickness T .

Parts 1 and 3 were assumed to be sufficiently long that the edge conditions would not influence the analysis. The outer surfaces of all three parts were used as reference surfaces in the analyses.

The validity of the analysis method employed in this study was examined earlier⁵⁵ through comparisons between analytical predictions and results for five photoelastic test models having fillet radius reinforcing. The results indicated satisfactory agreement, with the calculated maximum stresses being within 10% of the measured maximum stresses.

A portion of the parameter study results is given in Table 14. The normalized stresses are listed for both internal pressure and bending moment loadings (represented by P and M , respectively, in the load column). In the table, r_o is the outer radius of the nozzle ($d/2 + t$) and A_1 is the cross-sectional area of reinforcing provided by the fillet; the remaining symbols are defined on the model in Fig. 23.

Considering current water-cooled reactors, for which the diameter-to-thickness ratios of the spherical heads are usually less than 30, it

Fig. 23. Typical model employed in parameter study of radial nozzle reinforcement.

Table 1b. Stresses due to pressure or constant loading of nozzles in spherical vessels with fillet radius reinforcing, $t/t = D/2T$

Model No.	D/T	d/D	r_0/r	$r_0/\sqrt{at}$	$2A_1/dT$	Load	Normalized stress ^a			
							At A	At B	At C	At D
20	10	0.01	0.25	5.62	0.246	P	2.00	1.25	1.07 _{m1}	0.55
						M	0.00	0.00	0.07	1.11(1)
21		0.02	0.25	2.81	0.119	P	1.95	1.50	1.15 _{m1}	0.33
						M	-0.01	0.00	0.07	1.20(1)
22		0.04	0.25	1.40	0.0545	P	1.85	1.83	1.51 _{m1}	0.39
						M	-0.01	0.01	0.09	1.37(1)
8		0.08	1.72	4.82	0.888	P	1.54	1.34	1.09 _{m0}	0.58
						M	-0.10	0.05	0.06	1.30(1)
23		0.16	3.22	4.50	0.987	P	1.42	1.44	1.09 _{m1}	0.55
						M	0.23	0.14	0.09	1.31(1)
24		0.32	6.82	4.78	0.947	P	1.27	1.41	1.13 _{m1}	0.48
						M	-0.21	0.21	0.09	1.33(1)
25		0.50	15.9	7.13	0.866	P	1.21	1.35	1.15 _{m1}	0.54
						M	-0.22	0.19	0.05	1.27(1)
1	20	0.01	0.25	3.96	0.125	P	1.97	1.41	0.97 _{m1}	0.38
						M	0.00	-0.01	0.12	1.23(1)
2		0.02	0.25	1.98	0.0616	P	1.91	1.73	1.26 _{m1}	0.27
						M	-0.02	0.00	0.11	1.27(1)
3		0.04	1.11	4.40	0.515	P	1.70	1.26	0.98 _{m0}	0.54
						M	-0.06	0.03	0.06	1.21(1)
4		0.08	2.47	4.90	0.997	P	1.54	1.30	1.02 _{m0}	0.57
						M	-0.12	0.08	0.05	1.20(1)
5		0.16	4.16	4.12	0.987	P	1.36	1.38	1.00 _{m0}	0.54
						M	-0.20	0.19	0.11	1.23(1)
6		0.32	8.35	4.14	0.947	P	1.20	1.39	1.09 _{m1}	0.48
						M	-0.19	0.24	0.16	1.26(1)
7		0.50	16.9	5.35	0.866	P	1.17	1.34	1.06 _{m1}	0.49
						M	-0.13	0.18	0.13	1.25(1)
14	640	0.01	3.72	10.4	1.00	P	1.86	1.17	0.97 _{m1}	0.97
						M	-0.05	0.04	1.40	1.40(2)
16		0.02	5.67	7.34	1.00	P	1.64	1.28	1.06 _{m1}	1.06
						M	-0.11	0.14	1.33	1.33(2)
17		0.04	8.30	5.82	0.999	P	1.54	1.46	1.32 _{m1}	-0.27
						M	-0.09	0.41	1.14	1.27(1)
13		0.08	12.35	4.32	0.997	P	1.45	1.62	1.48 _{m1}	1.63
						M	0.13	0.90	0.90	1.66(3)
18		0.16	19.4	3.39	0.987	P	1.37	1.75	1.46 _{m1}	1.79
						M	0.36	1.50	0.77	2.76(3)
19		0.32	34.2	2.99	0.947	P	1.30	1.72	1.15 _{m1}	1.73
						M	0.62	1.83	0.74	3.46(3)
15		0.50	56.4	3.15	0.866	P	1.24	1.57	0.85 _{m1}	1.49
						M	0.22	1.70	0.66	3.30(3)
8	10	0.08	1.72	4.82	0.888	P	1.54	1.34	1.09 _{m0}	0.58
						M	-0.10	0.05	0.06	1.30(1)
4	20	0.08	2.47	4.90	0.997	P	1.54	1.30	1.02 _{m0}	0.57
						M	-0.06	0.03	0.06	1.20(1)
9	40	0.08	3.36	4.70	0.997	P	1.50	1.31	0.93 _{m0}	0.45
						M	-0.13	0.12	0.06	1.18(2)
10	80	0.08	4.60	4.55	0.997	P	1.49	1.35	0.81 _{m0}	0.47
						M	-0.14	0.21	0.08	1.24(2)
11	160	0.08	6.36	4.45	0.997	P	1.46	1.41	0.65 _{m0}	0.59
						M	-0.11	0.36	0.16	1.27(2)
12	320	0.08	8.83	4.37	0.997	P	1.44	1.51	0.92 _{m1}	0.77
						M	-0.03	0.59	1.14	1.27(2)
13	640	0.08	12.35	4.31	0.997	P	1.45	1.62	1.48 _{m1}	1.63
						M	0.13	0.90	0.90	1.66(3)

^a Normalized stress = σ/S , where for internal pressure, $S = P/2T$, and for bending moments, $S = M/\pi r^2 t$; see Fig. 23 for locations A, B, and C. Location D is where the largest stress due to constant loading occurs and is identified for each model as D1(1), D2(2), or D3(3). Stresses at A and B are normal to the section. At location C: n_0 = normal, outside surface; n_1 = normal, inside surface; t_1 = tangential, inside surface.

^b Tangential stress is slightly higher at these locations.

can be concluded from this study that for nozzles with fillet radius reinforcing the following conditions exist:

1. Stresses at the inside corner, location A where the maximum stress under pressure loading occurs, will not be increased significantly by an added moment loading producing a nominal stress in the nozzle ($M/\pi r^2 t$) equal to the internal pressure nominal stress ($PD/4T$).
2. For a nozzle-to-sphere diameter ratio greater than 0.05, the highest stress due to combined pressure and moment loading when $M/\pi r^2 t = PD/4T$ will be a tangential stress at location D1.

Parameter study of inside vs outside reinforcing

A limited parameter study was conducted by using the finite-element method of analysis to investigate the effectiveness of various amounts of internal and external compact reinforcement for spherical-vessel radial-nozzle configurations subjected to internal pressure loadings. The results of these studies were reported in the last annual report⁶² and in an ASME paper.⁶³ Because of the findings from the results obtained and the significance of this type of investigation, additional studies were conducted during the past year.

Although the finite-element method of analysis has been validated and successfully used for this particular type of investigation, a faster, less-expensive method of analysis was sought for carrying out the extended studies. The computer code STATIC,⁵⁷ which is based on a numerical shell analysis scheme, was selected for this purpose, and models from the finite-element studies were used as standards in evaluating the code for the purpose intended. One of the models chosen for examination had been analyzed experimentally (model WN-10B in Ref. 63) as well as by the finite-element method (this model was later designated as model S-24E). The second, model S-72D, had been subjected to finite-element analysis only.

Model S-24E is a relatively thick shell with diameter-to-thickness ratios of $D_1/T = 24$ and $d_1/t = 11.4$ for the shell and nozzle, respectively, and a nozzle-to-shell stress ratio of $s/S = 0.95$. This model has 65% outside fillet reinforcement, based on Section III of the ASME Boiler and Pressure Vessel Code cross-sectional area replacement rule. Model S-72D

is a relatively thin-shell model with $D_i/T = 71.9$, $d_i/t = 35.7$, and $s/S = 1.0$. This model has 100% outside fillet reinforcement, based on the ASME code area replacement rule.

Comparisons of results are shown in Fig. 24 for model S-24E and in Fig. 25 for model S-72D. Although the comparisons for the relatively thick-walled model S-24E were not particularly good, this model represents the upper limit of thicknesses to be considered in the extended parameter study.

Five additional models were analyzed at BMI using the STATIC computer code for comparison with the ORNL finite-element results.⁶³ The resulting

Fig. 24. Comparisons of theoretical and experimental stresses for model S-24E.

Fig. 25. Comparisons of finite-element and static analytical results for model S-72D.

Fig. 25. Comparisons of finite-element and static analytical results for model S-72D.

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comparisons are listed in Table 15 in the form of maximum values of tangential stress, meridional stress, and stress intensity for each configuration. The reinforcement values listed in the table were calculated using the Alternate Rules for Nozzle Design of Section VIII, Division 2 (Section AD-560) which were developed under this program. With only one exception, the finite-element and shell analyses give the maximum principal stresses at essentially the same location. The one exception is model S-24D, for which the finite-element analysis predicts the maximum to be a tangential stress at the inside corner of the nozzle-shell junction, while the shell analysis gave the maximum as a tangential stress on the outside surface of the fillet near its intersection with the sphere. A complete description of the models and maximum stress locations is given in Ref. 63.

These results indicate that the shell program can be used to provide adequate guidance for code rule development, with substantial savings of time and money over those associated with the use of the finite-element method. The theoretical results obtained for the 28 models thus far analyzed using the shell program are given in Table 16. The locations of high stress as well as the symbols employed in the table are shown in Fig. 26. The reinforcement cross-sectional areas (A_i = area inside; A_o = area outside) are listed as fractions of the shell cross section removed (dT) in forming the opening.

The relationship of the results of the current parameter studies to the ASME Section III Code⁶⁵ requirements is shown graphically in Fig. 27 for $D/T = 24$ and $d/D = 0.10$ and in Fig. 28 for $D/T = 72$ and $d/D = 0.10$. These two plots show that by satisfying the code area replacement requirements for the reinforcement of nozzles, the code maximum stress intensity limits can be exceeded when a large portion of the reinforcement is located on the inside of the vessel. Also shown in the two figures are the locations of the controlling stresses for the various fractions of outside reinforcement.

As soon as the parameter studies are completed, proposed rule changes for ASME Section III will be formulated. The current series of studies will then be expanded to include bending moment loadings applied to the nozzles.

Table 15. Comparison of maximum calculated stresses for reinforced nozzle/spherical shell attachments

Model No.	Outside fillet radius	Shell D_1/T	Effective reinforcement ^a (% area replacement)		Maximum normalized stress ^b					
					Tangential		Meridional		Stress intensity ^c	
			Inside	Outside	F.E.	Shell	F.E.	Shell	F.E.	Shell
S-24D	3.31T	24	0.0	44.0	1.73	1.52	1.25	1.39	1.87	1.63
S-24F	0.33T	24	53.0	0.0	2.63	3.26	2.55	3.23	2.91	3.77
S-24G	1.42T	24	53.0	9.2	1.64	2.35	2.04	1.59	2.04	2.39
S-72D	8.62T	72	0.0	85.0	1.40	1.50	1.44	1.43	1.41	1.50
S-72F	0.40T	72	45.0	0.0	4.12	4.69	4.71	5.49	4.71	6.06
S-72F ^d	0.40T	72	45.0	0.0		5.56		5.83		5.83
S-72G	2.40T	72	45.0	9.6	2.33	3.09	1.83	2.33	2.50	3.42

^aArea within limits prescribed by NB-3339 Alternate Rules for Nozzle Design, ASME Boiler and Pressure Vessel Code, Section III (Winter 1971 Addenda).

^bStresses were normalized by dividing by shell membrane stress $S = PD_1/4T$, where P = internal pressure, D_1 = shell inside diameter, and T = shell thickness, F.E. = finite-element analysis (Ref. 64), and Shell = Kalnins' shell analysis (Ref. 57).

^c $\bar{\sigma}$ = largest absolute difference of principal stresses, i.e., largest of $|\sigma_1 - \sigma_2|$, $|\sigma_2 - \sigma_3|$, or $|\sigma_3 - \sigma_1|$.

^dAnalyzed for Poisson's ratio, $\nu = 0.3$.

Table 16. Summary of calculated stresses in nozzles in spheres
with various proportions of inside and outside reinforcing

Model No.	Dimensions		Reinforcement					Normalized stress at locations					
	D/T	d/D	A_i/dT	A_o/dT	r_o/T	A		B		C or D		$\bar{\sigma}$	
						σ_n	σ_t	σ_n	σ_t	σ_n	σ_t		
S-24/10H1	24	0.10	1.00	0.03	0.50	2.83	1.99	2.02	-0.65	1.00 C	1.51	1.18	
HD			1.00	0.06	0.695	2.54	1.57	1.96	-0.51	1.01 C	1.47	1.21	
H.8			0.80	0.20	1.286	1.83	0.94	1.57	-0.26	1.04 C	1.25	1.41	
H.6			0.60	0.40	1.870	1.40	0.75	1.31	-0.22	1.07 C	1.01	1.47	
H.4			0.40	0.60	2.340	1.08	0.56	1.07	0.53	1.13 C	0.72	1.54	
H.2			0.20	0.80	2.750	1.10	0.59	1.08	0.53	1.32 D	1.22	1.54	
B			0.00	1.00	3.130	1.11	0.62	1.09	0.52	1.34 D	1.25	1.51	
S-72/10H1	72	0.10	1.00	0.01	0.5	3.94	3.56	2.12	-2.52	1.01 C	1.85	0.75	
HD			1.00	0.06	1.20	3.07	2.08	2.11	-1.20	1.02 C	1.75	0.94	
H.8			0.80	0.20	2.16	1.95	0.88	1.61	-0.35	1.04 C	1.45	1.29	
H.6			0.60	0.40	3.10	1.04	0.41			1.08 C	1.13	1.42	
H.4			0.40	0.60	3.85	1.04	0.55	1.03	0.50	1.12 C	0.77	1.46	
H.2			0.20	0.80	4.48	1.06	0.58	1.04	0.49	1.36 D	1.31	1.62	
B			0.00	1.00	5.07	1.08	0.63	1.04	0.49	1.42 D	1.39	1.48	
S-24/50H1	24	0.50	1.00	0.001	0.50	2.39	2.53	1.18	-1.41	1.06 C	2.51	0.49	
HD			1.00	0.06	3.47	1.99	1.48	1.40	-0.48	1.12 C	2.13	0.80	
H.8			0.80	0.20	6.41	1.61	0.94	1.28	-0.10	1.12 C	1.83	0.85	
H.6			0.60	0.40	10.02	1.25	0.40	1.32	0.63	1.13 C	1.31	0.97	
H.4			0.40	0.60	13.30	1.10	0.52	1.15	0.49	1.14 C	0.93	1.05	
H.2			0.20	0.80	16.43	1.08	0.55	1.08	0.54	1.28 D	1.16	1.07	
B			0.00	1.00	19.40	1.09	0.56	1.08	0.53	1.35 D	1.22	1.09	
S-72/50H1	72	0.50	1.00	<0.001	0.5	3.12	3.85	1.10	-2.78	1.09 C	3.09	0.35	
HD			1.00	0.08	6.00	2.31	1.84	1.50	-0.88	1.13 C	2.42	0.66	
H.8			0.80	0.20	9.60	1.80	1.13	1.37	-0.29	1.11 C	2.03	0.75	
H.6			0.60	0.40	14.40	1.33	0.61	1.22	0.31	1.09 D	1.34	0.90	
H.4			0.40	0.60	18.45	1.09	0.54	1.08	0.59	1.15 D	1.08	0.99	
H.2			0.20	0.80	22.10	1.04	0.54	1.03	0.51	1.33 D	1.29	1.03	
B			0.00	1.00	25.50	1.05	0.55	1.03	0.51	1.42 D	1.33	1.01	

^a A_i = inside reinforcing area; A_o = outside reinforcing area; σ_n = stress normal to plane;
 σ_t = stress index, and $\bar{\sigma}$ = stress intensity.

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Fig. 26. Stress locations in nozzle-to-spherical shell model reinforcement parameter study.

Fig. 27. Influence of fraction of nozzle reinforcement located inside the vessel on maximum stresses for models with $D/T = 24$ and $d/D = 0.10$ under internal pressure loading.

Fig. 28. Influence of fraction of nozzle reinforcement located inside the vessel on maximum stresses for models with $D/T = 72$ and $d/D = 0.10$ under internal pressure loading.

Single-Nozzle to Spherical-Shell Attachments

Experimental stress analyses of strain-gaged steel models are being continued with emphasis on nonradial nozzle configurations. Also being continued is the development of a closed-form solution type shell analysis for single nonradial nozzles. The achievements in both areas are detailed below.

Experimental stress analysis of single nozzles in spherical shells - University of Tennessee (I)

An extensive experimental investigation of single radial and nonradial nozzles attached to hemispherical shells is under way at the University of Tennessee. The data obtained from these tests will form the basis for evaluating candidate theoretical analyses.

The models are steel hemispheres having a nominal mean radius of 15.25 in. and an inside diameter/thickness ratio of 80.35. The hemispheres were spun to approximate dimensions and then machined to the final dimensions. The nozzles were machined from round steel stock and welded to the hemisphere with a full penetration weld, which was subsequently ground and hand filed to essentially a zero-radius fillet. Included in the study are the various radial and nonradial nozzles listed in Table 17. The lengths of the nozzle both inside and outside the hemisphere were selected to assure that any discontinuity stresses at the junction were eliminated at the extreme ends of the nozzle. The models are instrumented both inside and outside with foil resistance-type strain-gage rosettes and tested under internal pressure loading and nozzle loadings consisting of axial thrust, bending moments, torsional moments, and shear forces.

The strain-gage data for each loading are processed by computer⁶⁶ to produce tables of normalized stresses for the various rosette positions and plots of normalized stress vs model profile. The normalizing factor employed differed according to the type of loading.

The status of each model is indicated in Table 17 by appropriate footnotes. The model designations not footnoted indicate configurations for which testing has been completed.

A report containing the plots of stress vs model profile for the various loadings applied to the 22 1/2" nonradial nozzle TS 802-221 has been published.⁶⁷ The results in tabular form were published previously.⁶⁸⁻⁷⁰ The test model consists of a 2 5/8-in.-OD nozzle attached to the hemisphere as shown in Fig. 29.

The strain-gage rosettes were positioned both inside and outside the model along the lines shown in Fig. 30. The resulting strains were measured for the following loadings:

1. internal pressure applied to the hemisphere and nozzle assembly,
2. an axial load applied colinearly with the nozzle,
3. a pure bending moment or axial couple applied to the nozzle at angles of 0, 15, 30, 45, 90, 135, and 180°, measured as indicated in Fig. 30,

4. a transverse or shear load applied to the nozzle at angles of 0, 45, 90, 135, and 180°,
5. a pure torque applied in a plane normal to the axis of the nozzle.

Typical results are presented as plots of normalized principal stresses in Figs. 31 to 34 for loads of internal pressure and axial force.

Table 17. University of Tennessee single-nozzle configurations

Model designation	Dimensions ^a (in.)				
	L_o	L_i	d_o	d_i	t
7.875-in.-OD radial nozzle					
TS 807-001	14.0	10.0	7.875	7.125	0.375
TS 807-002	14.0	10.0	7.875	7.500	0.187
TS 807-003 ^b	14.0	0.0	7.875	7.500	0.187
7.875-in.-OD 45° nozzle					
TS 807-451 ^b	14.0	0.0	7.875	7.125	0.375
2.625-in.-OD radial nozzle					
TS 802-001	8.5	4.0	2.625	2.125	0.25
TS 802-002	8.5	4.0	2.625	2.375	0.125
TS 802-003	8.5	0.75	2.625	2.375	0.125
TS 802-004	8.5	0.375	2.625	2.375	0.125
TS 802-005	8.5	0.0	2.625	2.5	0.062
2.625-in.-OD 22 1/2° nozzle					
TS 802-221	8.5	4.0	2.625	2.125	0.25
TS 802-222 ^c	8.5	4.0	2.625	2.500	0.0625
TS 802-223 ^b	8.5	0.0	2.625	2.500	0.0625
2.625-in.-OD 45° nozzle					
TS 802-451 ^b	8.5	4.0	2.625	2.125	0.25
TS 802-452 ^b	8.5	4.0	2.625	2.500	0.0625
TS 802-453 ^b	8.5	0.0	2.625	2.500	0.0625

^a L_o = length of nozzle extension outside vessel; L_i = length of nozzle extension inside vessel; d_o = outside diameter of nozzle; d_i = inside diameter of nozzle; t = wall thickness of nozzle.

^bConfiguration to be tested.

^cConfiguration being tested.

NOTE: ALL PLANES IN THE HEMISPHERE PRODUCE GREAT CIRCLES ON THE DOME.

2.875 OD
(120 THREADS/INCH)

NOTE:
DIMENSIONS IN INCHES

0.250 WALL
2.625 $\pm$ 0.003
2.125 $\pm$ 0.003

1.031 DIAM
(36 HOLES)

0.390
0.375 WALL

0.75

2.833

34.0

40.0

45°

22.5°

45°

A diagram of a goniometer. A vertical rod is shown with a horizontal dashed line labeled 'A' at both ends. The rod is marked with angles: 45° PLANE, 90° PLANE, 135° PLANE, and 180° PLANE. A semi-circular arc is attached to the rod, with labels for 0° PLANE, 45° PLANE, 90° PLANE, 135° PLANE, and 180° PLANE. The rod is positioned at the 0° PLANE mark on the arc.

Fig. 30. Strain-gage reference-plane locations for the dome and nozzle.

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Fig. 31. Normalized principal stress vs profile for 0° plane and internal pressure load.

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Fig. 32. Normalized principal stress vs profile for 0° plane and axial force on nozzle.

ORNL-DWG 71-8684

Fig. 33. Normalized principal stress vs profile for 0° plane and couple on nozzle in a plane normal to the nozzle axis.

ORNL-DWG 71-8685

Fig. 34. Normalized principal stress vs profile for 0° plane and couple on the nozzle in the 0° axial plane.

on the nozzle, a couple on the nozzle in a plane normal to the axis of the nozzle, and a couple on the nozzle in the zero-degree axial plane. All plots are for strain gages in the zero-degree plane.

After the testing of model TS 802-221 was completed, the thickness of the nozzle was reduced from 0.25 to 0.0625 in., thereby forming model TS 802-222. The strain gages removed or damaged during the boring of the nozzle were replaced and the prescribed loadings repeated for the new configuration. The last of the planned loadings consisting of bending moments applied to the nozzle are nearing completion. When the data have been subjected to preliminary examination, the model will be modified by removing the nozzle internal protrusion to form model TS 802-223, the final configuration for the 2 5/8-in.-OD, 22 1/2° nozzle.

Model TS 807-003, the last of the present radial nozzle configurations, was produced by removing the 10.0-in. internal protrusion from model TS 807-002. Since testing of this model has been assigned a lower priority than the nonradial nozzle models, replacement of damaged strain gages is being done as fill-in work.

A paper⁷¹ describing the University of Tennessee single-nozzle testing program was presented at the ASME-sponsored First National Congress on Pressure Vessels and Piping. The paper includes a summary of the results obtained thus far and representative computer-generated stress vs profile plots of the 2.625-in.-OD nozzle configurations.

Theoretical analysis of stresses in a single nonradial nozzle in a spherical shell - Auburn University and ORNL

Methods of analysis and associated computer programs were developed for both nonprotruding and protruding nonradial nozzles attached to spherical shells^{72, 73} by Auburn University (AU-II), and a paper⁷⁴ describing results of the theoretical developments was presented at the ASME-sponsored First National Congress on Pressure Vessels and Piping. The paper contains a description of the analysis for internal pressure loading and demonstration numerical analyses for four models having a variety of dimensional parameters and angles of inclination of 10, 20, and 22 1/2°.

The method of analysis developed at Auburn for nonradial nozzles attached to spherical shells⁷³ was subsequently modified and expanded at

ORNL. One of the most significant changes is associated with treatment of the boundary conditions. The original and the modified methods employ a least-squares, point-matching technique for satisfying boundary conditions at the nozzle-shell intersection. As originally written, the associated computer program treated a series of points spaced at 12° intervals along the intersection curve, thereby producing 192 equations and 126 unknowns. In the modified computer program the number of points along the intersection has been increased by using an angular spacing of 2° , which resulted in a total of 1092 equations with 126 unknowns. This modification has resulted in a significant improvement in the accuracy of matching boundary conditions between the nozzle and sphere.

The improved computer program, **NONRAD2**, was used to analyze the University of Tennessee 22 $1/2^\circ$ nonradial nozzle model TS 802-221 shown in Fig. 29. Analytical results were obtained for an internal pressure loading, as well as axial thrust and bending moments applied to the nozzle. Comparisons between analytical and experimental results for the internal pressure case are shown in Figs. 35 and 36 for the zero and 180° planes respectively (oriented as shown in Fig. 30). Each figure consists of a series of plots of normalized stresses for the nozzle inner surface, nozzle outer surface, internal protrusion inner surface, internal protrusion outer surface, sphere inner surface, and sphere outer surface. The stresses were normalized to the nominal membrane stress ($PD/4T$), where P is the internal pressure, D is the nominal diameter of the sphere, and T is the thickness of the sphere. The points in the figures represent experimental results, and the lines represent analytical results. The theoretical solution appears, on the basis of these comparisons, to give a satisfactory solution for the internal pressure loading of nonradial nozzles having oblique angles of 22 $1/2^\circ$ or less.

Photoelastic Studies of Closely Spaced Reinforced Nozzles Attached to Cylindrical Shells

Westinghouse Research Laboratories

Photoelastic studies were conducted to determine the stress distributions associated with two closely spaced, fully reinforced nozzles

Fig. 35. Comparisons of analytical and experimental results for the 0° plane of 22 1/2" nonradial nozzle 802-221 (2.625 in. OD, 2.125 in. ID, and 4.00 in. internal protrusion).

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Fig. 35. Comparisons of analytical and experimental results for 0° plane of 22 1/2° nonradial nozzle 802-221 (2.625 in. OD, 2.125 in. ID, and 4.00 in. internal protrusion).

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Fig. 36. Comparisons of analytical and experimental results for the 180° plane of 22 1/2° nonradial nozzle 802-221 (2.625 in. OD, 2.125 in. ID, and 4.00 in. internal protrusion).

36. Comparisons of analytical and experimental results for plane of $22\frac{1}{2}^\circ$ nonradial nozzle 802-221 (2.625 in. OD, 2.125 in. ID, and 4.00 in. internal protrusion).

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radially attached along the generators of cylindrical vessels. A two-fold purpose was to be served: basic information regarding stress distributions would be obtained and the data would be used in assessing code rules for closely spaced nozzles. To further the first objective, the vessels tested also contained single nozzles with the same design and reinforcement as each of the closely spaced nozzles. Two photoelastic models were used in the investigation; vessel 1 was relatively thick walled, with $D_1/T = 12.25$, and vessel 2 was thin walled, with $D_1/T = 101.7$.

In addition to the nozzles described, the models also contained other nozzles. Each had a single nozzle of the same dimensions as the closely spaced nozzles but which was reinforced internal to the vessel. The nozzles and openings in these vessels are described in Ref. 62.

The results from the photoelastic investigations of the closely spaced nozzles in the two vessels have been published.⁷⁵ This report also contains results for the companion single nozzles both with reinforcement identical to that for each of the two closely spaced nozzles in a given vessel and with reinforcement inside the vessel. Cross-sectional views of vessels 1 and 2 are shown in Figs. 37 and 38 respectively. The two closely spaced reinforced nozzles are designated WC-12DD and WC-10DD, respectively, and the companion single nozzles WC-12D and WC-10D. The nozzles with internal protrusions and reinforcing are designated WC-12E and WC-10E. Also shown in Fig. 38 is a 60° hillside reinforced nozzle, WC-100C1, which was attached to vessel 2. All components were machined from solid castings of anhydride-cured epoxy resin. The nozzles were machined separately and cemented into holes bored into the vessels.

The principal stresses for each nozzle were determined by the photoelastic method of freezing and slicing.⁷⁶ The stresses along and normal to the plane, σ_t and σ_n respectively, were analyzed. Comparisons of stress patterns for the longitudinal sections (0 and 180°) of single nozzle WC-12D with those for the closely spaced nozzle WC-12DD (A) are shown in Fig. 39. The stress patterns for the transverse sections of the same two nozzles are shown in Fig. 40. Taking into consideration experimental error, the shapes of the stress distributions for the single and closely spaced nozzles do not differ significantly.

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Fig. 38. Shape and nominal dimensions of cylindrical vessel 2 containing two closely spaced reinforced openings WC-100DD (A) and (B), single opening WC-100D, inward protruding nozzle WC-100E, and the 60" hillside nozzle WC-100C1. (Westinghouse Electric Corporation drawing)

WC-100D AND WC-100DD, LONGITUDINAL (0° AND 180°) SECTIONS

PRINCIPAL STRESS RATIO σ_t/S

- WC-100D, 0° AND 180° SECTIONS
 - - - WC-100DD, (A) AND (B), 0° SECTIONS
 - - - WC-100DD, 180° SECTIONS

Fig. 41. Comparison of principal stress ratio σ_t/S on the inner and outer surfaces of the axial section of symmetry (0 and 180° sections) for the two closely spaced reinforced openings WC-100DD (A) and (B) and the identical single opening WC-100D in a thin-walled cylindrical vessel subjected to internal pressure. (Westinghouse Electric Corporation drawing)

patterns for the transverse sections (90 and 270°) of nozzles WC-100D and WC-100DD (A) are shown in Fig. 43.

The maximum measured circumferential principal stresses σ_n developed at the inner corners of the nozzles are listed in Table 18 for the 0 and 180° sections. Also shown in the table are ratios of peak circumferential

WC-1000 AND WC-100DD: LONGITUDINAL (0° AND 180°) SECTIONS

Fig. 42. Comparison of principal stress ratio σ_p/S on the inner and outer surfaces of the axial section of symmetry (0 and 180° sections) for the two closely spaced reinforced openings WC-100DD (A) and (B) and the identical single opening WC-100D in a thin-walled cylindrical vessel subjected to internal pressure. (Westinghouse Electric Corporation drawing)

WC-1000 AND WC-10000 (A) AND (B) TRANSVERSE (90° AND 270°) SECTIONS

- PRINCIPAL STRESS RATIO σ_t/S , WC-1000, 90° AND 270° SECTIONS
- PRINCIPAL STRESS RATIO σ_t/S , WC-10000 (A) AND (B), 90° AND 270° SECTIONS
- PRINCIPAL STRESS RATIO σ_r/S , WC-1000, 90° AND 270° SECTIONS
- PRINCIPAL STRESS RATIO σ_r/S , WC-10000 (A) AND (B), 90° AND 270° SECTIONS

Fig. 43. Comparison of the principal stress ratios σ_t/S and σ_r/S on the inner and outer surfaces of the transverse sections (90° and 270°) for the two closely spaced reinforced openings WC-100DD (A) and (B) and the identical single opening WC-100D in a thin-walled cylindrical vessel subjected to internal pressure. (Westinghouse Electric Corporation drawing)

stresses for closely spaced nozzles to those for single nozzles. The four ratios in Table 18 are plotted in Fig. 44 and show excellent agreement with the results of Ling⁷⁷ for closely spaced holes in an infinite plate subjected to biaxial stress. The stresses given in Table 18 for closely spaced nozzles were higher than those obtained for the companion single nozzles tested.

Table 18. Ratio of peak circumferential stresses obtained for closely spaced nozzles to those of the single nozzle

Model	Maximum σ_n		Ratio of $(\sigma_n)_{DD}$ to $(\sigma_n)_D$		L/d_i (see Fig. 44)
	0° Section	180° Section	0° Section	180° Section	
WC-12D	2.81S	2.81S			∞
WC-12DD	2.87S	2.96S	1.02	1.05	1.34
WC-100D	2.27S	2.27S			∞
WC-100DD	2.63S	3.00S	1.16	1.32	0.42

Fig. 44. Comparison of stresses at points A and B for two closely spaced reinforced openings in a pressurized cylinder with the stresses at two holes in an infinite plate. (Westinghouse Electric Corporation drawing)

Determination of Plastic Collapse
Loads for Tees and Laterals

(Sponsored by PVRC)

The progress during the past year on developing methods for calculating upper and lower bounds to limit loads for nozzle-to-cylinder attachments is discussed here. The results of the experimental collapse load studies and the results of studies on flat plates with skewed holes are briefly summarized.

Upper-bound limit analysis studies for
tees and laterals - University of Waterloo

Computerized analysis methods are being developed to calculate upper-bound limit loads for tee and lateral connections. The loadings considered are external force and moment loadings applied to the branch or nozzle and internal pressure. Computational methods have been developed for models having branch/run diameter ratios (d/D) ranging from zero to 1.0, diameter/thickness ratios (D/T) greater than 20, elastic branch/run hoop stress ratios (s/S) from 0.25 to 3.0, and branch/run angles θ between 30 and 90° (for tees, $\theta = 90^\circ$). The analyses can be used for configurations with fillets at the junctions between the nozzles and the shells.

Two separate formulations and methods for determining upper-bound limit pressures are used to cover the full d/D range. This is necessary because, for laterals having d/D ratios approaching one, the plastic region has a complex geometry and high strain concentrations develop at the inside acute corner of the junction. The analyses are discussed in a two-part document; Part I⁷⁸ applies to d/D ratios varying from 0 to 0.7 and Part II⁷⁹ to values of d/D from 0.7 to 1.0. In each case, the upper bounds are extremized with respect to the velocity parameters and the extension of the plastic region. Because of the complex energy expressions, a computer is used for numerical integration of dissipation energies and work rates using Simpson's rule. A nonlinear programming technique in the form of a computer program called CLIMBS is used to extremize the upper bounds.

In addition, analysis methods and associated computer programs are being developed to calculate upper bounds to in-plane and out-of-plane limit couples for d/D ratios up to 0.9. Two programs are under development, both of which will have the added capability of calculating upper bounds for each respective couple loading in combination with internal pressure for d/D values up to 0.7.

A series of experimental studies were conducted in conjunction with the theoretical work. The dimensions of the models and preliminary test results are listed in Table 19. Included are thirteen tees (designated T), ten 45° laterals (Y), and four straight pipe sections (F). Represented in the experimental study on tees and laterals are d/D ratios of 0.5 to 1.0, D/T ratios of 14.6 to 34.5, and branch/pipe hoop stress ratios (s/S) of 0.55 to 1.0. Models have been subjected to loadings of internal pressure P_i and both in-plane and out-of-plane couples applied to the branch (designated as C_i and C_o respectively). Three of the tees have been tested under combined loadings of internal pressure and either an in-plane or out-of-plane couple. The yield strengths listed in Table 19 are the uniaxial test results both for unannealed specimens cut from the undeformed ends of the models, denoted as T_E , and for annealed uniaxial specimens machined from the plastically deformed zones (T_I).

The limit loads were determined for each model by graphically determining the intersection of the initial and final tangents to the load vs deflection experimental curves. Typical curves resulting from the limit testing are shown in Fig. 45 for models T_5 , T_{12} , and Y_3 . All three were subjected to in-plane couples applied to the branch; T_{12} was also pressurized to 500 psi. It should be noted that the d/D ratio of model T_{12} is not the same as for models T_5 and Y_3 ; consequently, the differences in loading curves for the two tees are not due solely to the 500 psi internal pressure applied to model T_{12} .

Lower-bound limit analysis studies for nozzle-to-cylindrical shell attachments - University of Sherbrooke

The equations for calculating lower bounds to limit pressures of cylindrical nozzles attached to cylindrical shells (tees) were described in a paper⁸⁰ presented at the First National Congress on Pressure Vessels

Table 19. University of Waterloo plastic limit test results for cylinder-to-cylinder connections

P = internal pressure, C_1 = in-plane couple on branch, C_0 = out-of-plane couple on branch

R_A = external fillet radius on crotch for angles $\geq 90^\circ$, R_0 = external fillet radius on crotch for angles $\leq 90^\circ$

T_2 = unannealed tensile specimen from undisturbed ends of model; T_1 = annealed tensile specimen from junction region

Model No.	D (in.)	d/D	D/T	a/c	R _A (in.)	R _O (in.)	Loading	Limit		Yield strength (ksi)	
								(ksi)	(in.-kips)	T ₂	T ₁
Machined 45° wyes											
Y ₁	3.45	1.00	34.5	1.00	1.00	0.25	P	0.82		31.5	35.4
Y ₂	3.48	1.00	26.8	1.00	1.00	0.25	P	1.10		29.2	
Y ₃	3.49	0.75	25.0	1.00	1.00	0.25	C ₁		22.50	35.4	
Y ₄	3.47	0.63	29.0	0.94	1.00	0.25	P	1.15		32.5	31.4
Y ₁₀	3.48	1.00	26.8	1.00	1.00	0.25	P	1.05		30.3	
Welded 45° wyes											
Y ₅	3.20	1.00	14.6	1.00			P	3.40			35.6
Y ₆	4.30	1.00	17.0	1.00			P	2.10			36.9
Y ₇	4.30	0.77	17.2	0.80			P	2.60			33.9
Y ₈	4.30	1.00	17.7	1.00			P	1.90			37.7
Y ₉	3.50	0.63	22.0	1.00	1.00	0.25	C ₀		9.00	33.2	
Machined tees											
T ₁	3.45	1.00	32.8	1.00	0.50	0.50	P	1.45		33.6	35.3
T ₂	3.45	0.63	34.5	0.90	0.50	0.50	P	1.40		30.6	33.0
T ₃	3.64	0.83	26.0	1.00	0.50	0.50	P	1.88		42.0	38.5
T ₄	3.49	0.50	25.0	1.00	0.25	0.25	C ₀		5.00	30.8	
T ₅	3.49	0.75	25.0	1.00	0.50	0.50	C ₁		18.50	31.8	
T ₆	3.49	1.00	25.0	1.00	0.25	0.25	C ₀		24.00	31.4	
T ₇	3.49	1.00	25.0	1.00	0.50	0.50	P	1.70		36.5	
T ₈	3.49	0.50	25.0	1.00	0.25	0.25	C ₁		6.50	31.6	31.0
T ₉	3.49	1.00	25.0	1.00	0.50	0.50	C ₁		35.00 ^a	31.0	
T ₁₀	3.45	0.70	34.5	0.55	0.38	0.38	C ₀		5.40	29.5	
T ₁₁	3.49	0.75	25.0	1.00	0.50	0.50	P + C ₁	To be tested			
T ₁₂	3.48	1.00	26.8	1.00	0.50	0.50	P + C ₁	0.50	22.00	25.2	
T ₁₃	3.45	1.00	34.5	0.55	0.50	0.50	P + C ₀	1.00	6.50	30.1	
Straight pipes											
P ₁	3.15		21.0				C		37.0	29.2	
P ₂	3.10		31.0				P	2.10		28.2	
P ₃	3.60		41.0				P + C	0.75	27.0	28.2	
P ₄	3.60		51.0				C		24.0	27.7	

^aThe limit couple for this model is approximated since the test was stopped prematurely.

Fig. 45. In-plane limit couple tests of models T_3 , Y_3 , and T_{12} . Model T_{12} was tested with a 500-psi constant internal pressure.

and Piping. The equations were obtained by solving a series of mathematical programming subproblems. A solution is obtained by maximizing the pressure loading subject to equality and inequality constraints for all values of the nozzle angular coordinate θ over the range $0 \leq \theta \leq \pi/2$. Comparisons of the resulting lower-bound predictions with test results indicated that the solution is quite conservative.

To reduce the conservatism of the original lower-bound analysis, an improved solution⁸¹ was recently developed which satisfies the stress

continuity requirement at the nozzle-shell junction independent of equality constraints. The stress fields are functions of arbitrary parameters chosen together with spatial coordinates to satisfy equilibrium, symmetry, and boundary conditions as well as stress continuity requirements at a finite series of discrete points. These parameters are restricted to prevent violation of the specified yield condition when the pressure loading is maximized. Simplifying assumptions were not required at the junction of the nozzle and vessel since generalized geometric relationships are used. The problem was formulated using nonlinear programming, and numerical solutions are obtained using the Sequential Unconstrained Minimization Technique (SUMT).⁸²

Initially, the yield constraints were enforced at 80 points and the yield conditions were verified at 100 intermediary points.⁸¹ Figure 46

Fig. 46. Force and moment distributions in the longitudinal plane along the nozzle ($X = 0$, $Y = r$).

shows the distribution of normalized forces and moments along the nozzle in the longitudinal plane (plane of the two axes) resulting from an applied normalized pressure of 0.65. The normalized forces per unit length, n_θ , n_z , and q_z , and the pressure p were obtained by dividing each force per unit length by the maximum membrane plastic strength of the vessel cross section; the normalized moments m_θ and m_z were obtained by dividing each moment per unit length by the maximum lower-bound plastic bending moment capacity of the vessel cross section. The maximum value in each case shown in Fig. 46 occurred at the nozzle-vessel junction, and no abrupt stress variations were revealed. In the figure, l_2 shows the location of the free end of the nozzle.

Parameter studies have been conducted using inequality constraints at 81 points. Both the nozzle and vessel are assumed to be of sufficient length so that the study could be limited to three geometric variables - the vessel diameter/thickness ratio D/T , the nozzle/vessel diameter ratio d/D , and the nozzle/vessel nominal hoop stress ratio s/S . With respect to materials properties, a single nozzle/vessel yield stress ratio of $\sigma_y = 1$ was used.

Results of the study are presented in Figs. 47 and 48, where the ordinate of both plots is $\bar{p}$, the normalized lower bound to the limit pressure. The normalizing factor is obtained by dividing the pressure by the maximum membrane plastic strength of the vessel cross section, $\sigma_y R/T$, where σ_y is the yield strength of the material, R is the radius, and T is the thickness. Figure 47 is a series of curves for various values of the nozzle membrane/shell membrane stress ratio s/S for a constant D/T of 30; Fig. 48 is a series of curves for selected values of D/T when $s/S = 1$. The limit pressure is a minimum in both figures for $0.5 < d/D < 0.6$. In Fig. 47, substantial increases in the limit pressure occur for corresponding small increases in nozzle thickness which are indicated by decreases in the value of s/S . This provides an indication of possible benefit of nozzle reinforcement in a nozzle-to-vessel attachment.

Comparison of calculated limit pressures and experimentally determined plastic collapse pressures

As the various limit analysis results become available, comparisons are made with the experimental results. One set of comparisons is shown

Fig. 47. Variation of the limit pressure with nozzle-vessel diameter ratios for different nozzle-to-vessel hoop stress ratios.

Fig. 48. Variation of the limit pressure with nozzle-vessel diameter ratios for different vessel diameter-to-thickness ratios.

ii. Fig. 49 for both upper- and lower-bound limit pressure predictions together with experimental results for models T_1 , T_2 , T_3 , and T_7 that were tested at the University of Waterloo. The calculated curves are for $D/T = 30$ and $s/S = 1.0$.

The membrane-stress ratios (s/S) for all but model T_2 were the same as used in the calculations (see Table 19). However, the D/T ratios ranged from 25 to 34.5, but despite these differences, the agreement with results for T_1 and T_3 is considered good. The value for T_7 appears to be low, and further investigation is needed. Similarly, the value for T_2 appears to be high, but in this case the smaller s/S ratio of 0.9 is influential. It should be emphasized that these comparisons are preliminary and, as indicated previously, the geometry of some of the experimental and theoretical models disagree significantly. More specific evaluations will be made when additional experimental data become available.

Studies of skewed holes in flat plates - University of Sherbrooke

Studies of elastic stresses near skewed holes in flat plates are described in Welding Research Council Bulletin 153, which includes both theoretical analyses and experimental results.⁸³ Subsequent to these

Fig. 49. Theoretical upper and lower bounds to limit pressures, $s/S = 1$ and $D/T = 30$, including data points for University of Waterloo experimental models. (See Table 19 for model s/S and D/T values.)

studies, the relationship between sharpness of acute corners of skewed holes in flat plates and stresses and strains has been investigated experimentally at the University of Sherbrooke. These follow-on investigations were carried out under the partial sponsorship of the National Research Council of Canada and le Ministère de l'Éducation du Gouvernement du Québec and are reported here because of the relevance of the results to the understanding of nonradial nozzle-to-shell attachment behaviors.

The additional work includes a study of elastic strain (or stress) concentrations⁸⁴ and a study of the elastic-plastic strain distributions⁸⁵ for steel plates with skewed holes having various angles of inclination. Foil-type electrical-resistance strain gages were used to measure strains both around and through the depth of the hole when the specimen was loaded uniaxially in tension. The center lines of the circular holes were in the plane normal to the loading direction. The studies encompass holes having hole diameter/plate thickness ratios (d/t) from 1.53 to 2.62 and angles of inclination of 0, 30, 45, and 60°.

In the elastic study, 14 plate specimens were used to determine the magnitude of stress concentrations along the face, or surfaces, of the holes and the effect of reducing edge sharpness of skewed holes. The edge sharpness was reduced either by rounding the acute corner to give a circular profile of radius r or by making a cylindrical (vertical) cut normal to surface of the plate at the acute corner.

The results show that the maximum strain is dependent upon both the hole diameter/plate thickness ratio and the angle of inclination of the hole. A decrease in the sharpness of the acute corner along the edge of skewed holes will result in a reduction of the maximum circumferential strain for edge corner radius/plate thickness ratios (r/t) less than 0.2; however, a flat corner made by a vertical cut is more effective than a rounded edge for ratios greater than 0.2. In general, removing the acute corner causes the location of the maximum strain concentration to shift from the surface of the plate toward the interior.

A series of six plate specimens were tested in the study of elastic-plastic behavior. For a plate having a right circular cylindrical hole stressed elastically, the strain concentration at the midsurface is slightly greater than at either the inside or outside surface of the plate. As the

material is stressed to produce elastic-plastic response, the strain concentration at the midsurface increases appreciably compared with those at the plate surfaces. This behavior was not detected by surface strain measurements nor was it predicted using a two-dimensional analysis.

For plates with oblique holes, the maximum strain first occurs near the acute angle side of the hole. As the material becomes plastically strained, the variation of strain concentration with position through the plate thickness is not smooth as in the case of a straight circular hole. The strain distributions appear to be at least fourth-order functions of the distance along the oblique cut through the plate. This type of strain distribution is attributable to the degree of multiaxiality of the state of strain. Plates with skewed holes having an angle of inclination of 60° appeared to reach a state of plastic instability at a relatively low nominal strain level. It was evident that in each case the plastic zone remained highly localized in the acute-angle portions of the plate, where stress relief starts at low levels of nominal stress. The high levels of strain concentrations in very localized regions as observed in these tests have important effects on the determination of fracture and fatigue behavior of pressure vessels having skewed openings.

AEC(ORNL) PROGRAM - MULTIPLE-NOZZLE STUDIES

Under this portion of the overall program, the emphasis is on multiple-nozzle studies (see Table 12). Included are tests on flat plates with clusters of holes and with clusters of attached nozzles. Tied with these tests are analytical methods development phases in which the developments are carried out stepwise, progressing from methods for perforated flat plates to methods applicable to flat plates with attached nozzles. Experimental studies on spherical shells with clusters of holes and with clusters of nozzles are also included. It is planned that analytical methods appropriate to these configurations be developed in a manner similar to that being pursued for the flat plates. In the case of the experimental models under this portion of the program, particular efforts were made to impose elements of geometrical similarity throughout.

Experimental Stress Analysis of Holes in Flat
Plates - University of Tennessee (II)

In the stress analysis of relatively small nozzles attached to large-radius cylindrical or spherical vessels, vessel curvature has little influence on stresses resulting from internal pressure as well as nozzle loadings due to bending moments, torsional moments, and axial forces. Consequently, flat plates with attached nozzles can be used to study the behavior of large-radius vessels. As a first step in this work, a series of perforated flat plates have been subjected to experimental stress analysis at the University of Tennessee. The results of these tests are used in the development and evaluation of methods of analysis.

The models consist of four $36 \times 36 \times 0.375$ in. plates having clusters of 1, 2, 3, and 5 holes, respectively, as shown in Fig. 50. The plates are subjected to uniaxial and biaxial loadings using 32 hydraulic rams positioned 8 to a side along the edges. Initially, proof tests were

Fig. 50. Hole-cluster pattern and details for flat-plate models.

conducted using strain-gaged unpierced plates to evaluate the capability of the loading system to develop the specified stress conditions. The results of the proof tests have been published together with results of the uniaxial and biaxial loadings of the plates containing one and two holes.^{86,87} A summary of this work was included in the previous annual reports.^{62,88}

During this reporting period, the experimental results for the three- and five-hole flat plates were published.^{89,90} The three-hole flat plate is shown in Fig. 51. Electrical-resistance strain gages were installed on one side of the model in the positions indicated in Fig. 52. To facilitate the application of both uniaxial and biaxial loadings to a plate, two sets of hydraulic rams on adjacent sides of the plate were attached to separate manifolds. The procedure followed in the test was essentially the same as that used for the one- and two-hole plate tests.^{86,87} Each plate was tested first with the gaged side up in the loading apparatus

Fig. 51. Three-hole plate.

POSITION	LOCATION	GAGE TYPE
1	1.00R	SINGLE ELEMENT
2	1.07R	DELTA ROSETTE
3	1.5R	DELTA ROSETTE
4	2.0R	DELTA ROSETTE
5	3.0R	DELTA ROSETTE
6	4.0R	DELTA ROSETTE

Fig. 52. Strain-gage layout for the three-hole flat-plate model.

and then with the gaged side down, and the two sets of readings for each gage were averaged to compensate for possible bending or eccentricities in the loading system.

Typical radial and tangential stress distributions for gages along plane 12 in Fig. 52 are shown in Figs. 53 and 54 for uniaxial and for bi-axial loading respectively. The full curves in these figures represent theoretical results obtained at Battelle Memorial Institute,⁹¹ while the points represent the experimental results. The loading directions are indicated by the arrows. The ordinates are in terms of the stress ratio S/C , where S is the stress at a point and C is the undisturbed tensile stress of 1650 psi, and the abscissas represent the distance from the center of the hole divided by the hole radius R_0 . The radial and tangential stresses are denoted by S_r and S_t respectively. In general, the agreement between the analytical and experimental values is good except

Fig. 53. Radial and tangential stresses in uniaxial tension, $\theta = 90^\circ$.

Fig. 54. Radial and tangential stresses in biaxial tension. 1:1 ratio, $\theta = 90^\circ$.

for the tangential stresses S_t for the biaxial loading conditions shown in Fig. 54, where the experimental values are about 20% below the analytical results. Figure 55 shows the distribution of the tangential stresses around the edge of the hole for the biaxial loading condition, where the experimental points again lie beneath the analytical curve. The dashed curve shown in Fig. 55 represents the predicted results obtained by superimposing the results of uniaxial loadings in the two

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Fig. 55. Tangential stresses at edge of hole in biaxial tension, 1:1 ratio.

mutually perpendicular directions. A preliminary evaluation indicates reasonably good agreement among the biaxial test results, the prediction based on superposition of two uniaxial tests, and the analytical results.

The final plate tested in the perforated plate series contained the five-hole cluster shown in Fig. 50 with the central hole being positioned at the center of the plate. This model was instrumented with 60° delta rosettes in the positions shown in Fig. 56 and was tested using the procedure described for the three-hole flat plate. Results for uniaxial and biaxial testing of the five-hole model are shown in Figs. 57 and 58,

POSITION	LOCATION	GAGE TYPE
1	1.00 R	SINGLE ELEMENT
2	1.07 R	DELTA ROSETTE
3	1.5 R	DELTA ROSETTE
4	2.0 R	DELTA ROSETTE
5	3.0 R	DELTA ROSETTE
6	4.0 R	DELTA ROSETTE

Fig. 56. Location of strain gages on five-hole flat plate.

Fig. 57. Radial and tangential stresses for five-hole plate under uniaxial tension, $\theta = 0^\circ$.

respectively, for plane 9, a 0° gage plane. As in the case of the three-hole cluster, the solid lines represent analytical results obtained at BMI⁹¹ and the same normalizing factor C has been employed. The circles and triangles in Fig. 57 indicate data for separate uniaxial loadings applied in the two mutually perpendicular directions. Results for a 1:1 biaxial loading are shown in Fig. 58. The experimental values, as seen for this line of gages, were consistently below the analytical curve for

Fig. 58. Radial and tangential stresses for five-hole plate under biaxial tension. 1:1 loading ratio, $\theta = 0^\circ$.

the tangential stresses (S_t). There is no apparent reason for this discrepancy.

Experimental Stress Analysis of Spherical Shells and Flat Plates with Clusters of Nozzles - Auburn University (I)

The program for testing clusters of nozzles attached to flat plates and hemispherical shells is continuing at Auburn University. The computerized data reduction and plotting procedures⁶⁶ developed at the University of Tennessee in connection with the single-nozzle model tests have been adapted for use with the Auburn University test results. All data for the two-nozzle hemispherical shell model were assembled in a

report using these procedures. The report, which is a revision of an earlier version⁹² described in last year's annual report, is ready for printing.

Testing of both the one- and two-nozzle flat-plate models shown in Fig. 59 has been completed. Each model has been tested under in-plane biaxial loadings on the plate consisting of a 1:1 loading at two different stress levels, a 1:2 loading, and a 2:1 loading. After the biaxial loadings were completed the plates were restrained along the edges in the test frame and axial thrust and bending moment loadings were applied to the nozzles. The strain-gage rosettes performed satisfactorily with relatively few exceptions, and a ± 5 - μ in./in. agreement has been obtained between successive reloadings. All data for the two models have been examined and loadings with questionable data repeated.

The partially instrumented five-nozzle flat-plate model (see Fig. 59) has been transported from ORNL to the testing site at Auburn. This is the last of the current series of flat plate models to be tested. Completion of the strain-gage installation, which entails the mounting of

Fig. 59. Nozzle design and cluster-pattern details for flat-plate models.

50 additional strain-gage rosettes is under way. The total number of rosettes to be used is about 312.

The loading frame designed for testing the four-nozzle hemispherical shell model has been fabricated at Auburn. This model, previously described,⁶² was produced by attaching two additional nozzles to the two-nozzle hemispherical shell model. Instrumentation of the four-nozzle model with strain gages is also in progress.

Stress Analyses of Perforated Plates and Plates
with Single and Clustered Nozzle Attachments -
Battelle Memorial Institute (II)

A program is under way at Battelle Memorial Institute to develop a satisfactory method of analyses for nozzle clusters attached to flat plates. The computer program form of the analysis is being developed for flat plates having clusters of from one to five closely spaced holes or nozzles. Demonstration analyses have been performed on one- and two-nozzle clusters,⁹¹ and the computer analyses are being extended for application to the five-nozzle model. Included in the analysis capability are pressure, thrust, moment, and shear loadings of the nozzles, and biaxial loading of the plate (to be used for representing vessel internal pressure loading). Later we plan to include spherical shells as well. In all cases the results obtained using this analysis method will be compared with the corresponding experimental results being obtained at Auburn University.

CONCLUSIONS

During the current reporting period we have made substantial progress in the development and verification of analyses for nuclear pressure vessel nozzle attachments. Two of the more extensive experimental investigations, a photoelastic study of single and closely spaced nozzles at Westinghouse Research Laboratories and a strain-gage study of closely

spaced holes in flat plates at the University of Tennessee, have been successfully completed.

At ORNL, analyses of the recently tested nonradial nozzle configuration produce good agreement between the analytical and experimental results. Parameter studies have also been conducted at ORNL in conjunction with code rule development work under way at Battelle. Results of studies of various types of reinforcement for nozzle/spherical shell junctions have been compared with code design rules, and in some cases recommendations have been made for code modifications.

Upper- and lower-bound limit-load studies of nozzle/cylindrical shell attachments have progressed significantly, with portions of the work being completed. These studies, sponsored by the Pressure Vessel Research Committee and the National Research Council of Canada, include both analytical and experimental efforts.

Comparisons of calculated and experimental results for perforated flat plates were very favorable, and analytical work on nozzles attached to flat plates is in advanced stages of development. Good progress has also been made on the Auburn University strain-gage study of closely spaced nozzles attached to flat plates and spherical shells, with testing of the one- and two-nozzle flat-plate models having been completed. Although problems with data-acquisition equipment have caused considerable delay in the University of Tennessee experimental study of nonradial nozzles attached to spherical shells, testing of one nonradial nozzle model was completed and testing of a second model is nearing completion.

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