382

NRC-5, Germany, Japan (2)

Dro: 225

ORNL/NUREG/TM-18

Progress Report for the Design Criteria for Piping and Nozzles Program for the Two Quarterly Periods July 1 to Sept. 30 and Oct. 1 to Dec. 31, 1975

S. E. Moore
J. W. Bryson

MASTER

OAK RIDGE NATIONAL LABORATORY

OPERATED BY UNION CARBIDE CORPORATION FOR THE ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION

Printed in the United States of America. Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road, Springfield, Virginia 22161
Price: Printed Copy $4.50; Microfiche $2.25

ORNL/NUREG/TM-18
NRC-5

Contract No. W-7405-eng-26

Reactor Division

PROGRESS REPORT FOR THE DESIGN CRITERIA FOR PIPING AND NOZZLES PROGRAM FOR THE TWO QUARTERLY PERIODS JULY 1 TO SEPT. 30 AND OCT. 1 TO DEC. 31, 1975

S. E. Moore
J. W. Bryson

Work funded by the Nuclear Regulatory Commission
under Interagency Agreement 40-495-75

Manuscript Completed - May 11, 1976

JUNE 1976

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37830
operated by
UNION CARBIDE CORPORATION
for the
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION

CONTENTS

FOREWORD

The Design Criteria for Piping and Nozzles Program is an engineering-research activity being conducted at Oak Ridge National Laboratory for the Nuclear Regulatory Commission, Division of Reactor Safety Research (NRC-RSR); E. K. Lynn of the Metallurgy and Materials Branch is the cognizant RSR engineer. S. E. Moore, Reactor Division, ORNL, is the program manager.

Activities under the design-criteria program are coordinated with other safety-related piping-and-pressure-vessel research through the Design Division of the Pressure Vessel Research Committee of the Welding Research Council and through the ASME Boiler and Pressure Vessel Code Committee.

This report, which covers progress of program activities for the period July 1975—December 1975, is the first in a series of formal progress reports to be written under NRC-RSR sponsorship. Subsequent progress reports will be issued quarterly.

PROGRESS REPORT FOR THE DESIGN CRITERIA FOR PIPING AND NOZZLES PROGRAM FOR THE TWO QUARTERLY PERIODS JULY 1 TO SEPT. 30 AND OCT. 1 TO DEC. 31, 1975

S. E. Moore
J. W. Bryson

ABSTRACT

This report reviews the activities and accomplishments of the Design Criteria for Piping and Nozzles Program during the last six months. Précis are given of the reports published by program personnel about stresses at nozzles in spherical shells, about stresses at openings in cylindrical pressure vessels, about stresses at and stress indices for nozzle attachments, recommending stress indices for socket-welding fittings, measuring and analyzing stresses in cylinder-to-cylinder junctions, about stresses in flat plates with attached nozzles, and about stresses in and fatigue failure of forged tees. Summaries are also presented of work in progress dealing with the development of stress indices and flexibility factors for elbows, the development of code rules regarding isolated and closely spaced nozzles in cylindrical pressure vessels, procedures for analyzing the design of lugs attached to straight pipes, the analysis of stresses in flanged joints, load-limit and fatigue studies for elbows, and the elastic-response and fatigue-life behavior of forged tees.

Keywords: Stress analysis, flat plates, nozzles, cylinders, spherical shells, attachments, lugs, elbows, tees, penetrations, ORNL Design Criteria for Piping and Nozzles Program, flexibility factor, stress index, openings, socket-welding fittings, fatigue failure, ASME Boiler and Pressure Vessel Code, pipes, flanged joints.

INTRODUCTION

The Design Criteria for Piping and Nozzles Program is an engineering research activity being conducted at Oak Ridge National Laboratory (ORNL) for the U.S. Nuclear Regulatory Commission, Division of Reactor Safety Research (NRC-RSR). The objectives of the program are to conduct integrated experimental and analytical studies for the development and verification of adequate and safe structural-analysis methods for nuclear piping systems and single and multiple nozzle-to-pressure vessel attachments and to assess and/or develop design rules for use in relevant codes and standards. The program activity is closely coordinated with other federal-government-sponsored research and with industry-sponsored research through the Pressure Vessel Research Committee of the Welding Research Council. Design rules and other related materials developed under this program are submitted periodically to the American Society of Mechanical Engineers (ASME) for use in their codes and standards. Much of the information developed in this manner is now an integral part of the *ASME Boiler and Pressure Vessel Code*.[1]

The present program is a combination of two previous programs concerned with the study of stresses in the region of reinforced openings (nozzles) in pressure vessels and of stresses in piping-system components (or piping products), respectively. These were the ORNL Nozzle Analysis program (189a No. 10161) and the ORNL Design Criteria for Piping, Pumps, and Valves program (189a No. 10178). The studies completed earlier have resulted in the publication of 80 topical reports and open-literature publications. These activities have had a considerable impact on the present state of the art for assessing design rules that assure the safe design of nuclear-power-plant systems.

The current FY 1976-77 piping and nozzles program is organized into a program-administration task and six major activity areas. These are:

1. Code-Rules Development for the assessment of current design rules and/or the development of new or improved rules for use in design codes and standards.
2. Technical Reports for completion and publication of a number of topical reports on studies for which the experimental and analytical work is essentially complete.
3. Spherical-Shell Studies for completion of a series of planned studies on hemispherical shells with attached nozzles.
4. Cylindrical-Shell Studies for investigations of cylindrical pressure-vessel structures containing reinforced openings or nozzles.
5. Elbow and Curved-Pipe Studies for investigations of piping-system bends.

6. ANSI-B16.9-Tee Studies for the study of a class of standard piping tees.

Current plans are to also conduct studies on flat plate structures, which simulate the behavior of large-diameter vessels with relatively small-diameter nozzles; flanged and welded piping-system joints; and other miscellaneous piping products in the future, as budget and programmatic conditions allow.

A summary of the work conducted during the first- and second-quarter periods of FY 1976 (July-December) is in the following sections.

CODE-RULES DEVELOPMENT

Code-related activities this fiscal year include regular participation by staff members on several working groups and subcommittees of the ASME Boiler and Pressure Vessel Code Committee (BPVC) and of the Pressure Vessel Research Committee (PVRC), as well as program efforts to evaluate and/or develop design-analysis rules for specific items. These items include (1) the development and implementation of design-analysis procedures for lug supports on straight pipe, (2) the development of stress indices for standard socket-welding fittings, (3) the development of new or modified stress indices and flexibility factors for elbows to better account for the existence of pipe or other structures welded to the elbow (so-called end effects), and (4) the development of improved design rules for isolated and closely spaced nozzles in pressure vessels.

In brief, the status of each of our development projects is as follows: Item 1, for lugs, was completed and presented to the BPVC through the Working Group on Piping Design (WGPD) in September 1975. It was accepted as a potential code case, and if approved, should be published early in 1976.* Item 2, for socket-welding fittings, was completed and was also presented to the BPVC in September. It was accepted by the Working Group (WGPD) as a potential Code revision but was rejected by one of the higher-level Code committees and returned for further work. We

*This item (Code Case 1745) was approved by the Subcommittee on Nuclear Power on Jan. 8, 1976.

plan to work with the ASME in the development of more-acceptable design rules.

The third item, on elbow end effects, has been inactive during the past six months. However, a substantial amount of information has been collected that can be used in the development of stress indices for Code use. We plan to begin work again on this subject during the third quarter (January-March 1976) and present our recommendations to the ASME in the fall.

The fourth item, on isolated and closely spaced nozzles, is a continuation of work started last year[2,3,4] to provide more definitive rules for the reinforcement and/or separation requirements for nozzles in pressure vessels. During the past six months one of our consultants, Mr. J. L. Mershon (USAEC-retired), has been collecting and analyzing experimental and analytical data from studies conducted in the past and attempting to develop simple correlation formulas. Some progress has been made, but the work cannot be completed nor can better design rules be proposed until results from analytical studies currently being conducted under the cylindrical-shell-studies task are available. We hope to complete this work and to propose improved design rules to the ASME Code Committee by the end of fiscal year 1977.

A summary of the work under items 1 and 2 (i.e., for lugs and socket-welding fittings) is given below.

Design-Analysis Procedure for Lug Supports on Straight Pipe

The *ASME Boiler and Pressure Vessel Code*, Section III, Nuclear Power Plant Components, (hereinafter referred to as the Code) includes a "simplified" method of design analysis for Class-1 piping systems (subarticle NB-3650*) that makes use of stress indices and six rather simple criteria equations. When stress indices are provided for the particular piping component, the Code-analysis method is a relatively simple way to check the design for compliance with Code requirements. However, stress indices are not presently given in the Code for lugs or other support

---

*Reference to articles, subarticles, paragraphs, tables, or figures from the Code are identified by number (e.g., NB-xxxx), as appropriate.

attachments. The designer and the design reviewer must therefore rely on the use of other more-expensive and potentially less-reliable methods in order to show that the design meets the safety criteria of the Code.

In an effort to provide more-appropriate guidance for the design of lug supports on Class-1 pipe, a design-analysis procedure including appropriate stress indices has been developed and presented to the ASME for use in the Code. The analysis method, written specifically for lug attachments that are integral with the pipe or attached with full-penetration welds, includes provisions to evaluate the effects of thermal gradients between the lug and pipe, of internal pressure in the pipe, and of the six shear and moment loads shown in Fig. 1. The analysis method includes a complete set of equations and stress indices that enable calculation of the primary, primary-plus-secondary, and total (peak) stresses at a lug support as well as the stress-limit criteria equations for judging adequacy of the design. Documentation for development of the method is given in two ORNL reports[5,6] and in a *Welding Research Council Bulletin.*[6]

The present status is that the ASME is considering publication of the method as a code case. The complete text of the proposed code case is

ORNL-DWG 73-5839R

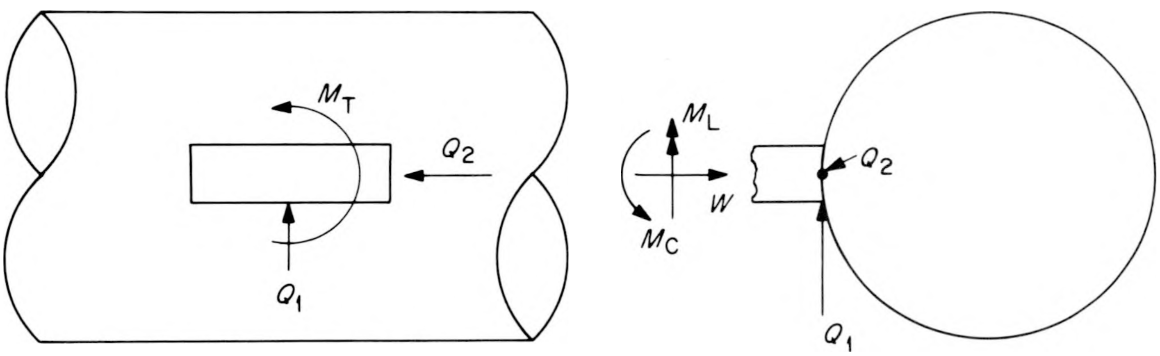

Fig. 1. Shear and moment loads that may act on a lug support: axial thrust W, transverse shear forces Q_1 and Q_2, and moment loads M_T, M_C, and M_L.

given below:

Proposed Code Case on Stress Indices for Integral Structural Attachments

Inquiry: What stress indices and procedures may be used to evaluate stresses in ASME Section III, Class 1 Piping at integral structural attachments?

Reply: It is the opinion of the committee that the stress indices and procedures listed below may be used to evaluate stresses in ASME Section III, Class 1 Piping, at integral structural attachments.

1.0 Limitations to Applicability of Indices

1.1 The attachment is welded to the pipe with a complete-penetration weld.

1.2 The attachment material, weld material, and pipe material have essentially the same moduli of elasticity and coefficients of thermal expansion.

1.3 $\beta_1 \leq 0.5$, $\beta_2 \leq 0.5$, and the product $\beta_1 \times \beta_2 \leq 0.075$, where β_1 and β_2 are defined in (c).

1.4 The attachment is made on straight pipe, with the nearest edge of the attachment weld located at a minimum distance of $\sqrt{rt}$ from any other weld or other discontinuity; r and t are defined in (c).

1.5 $D_o/t \leq 100$.

2.0 Equations To Be Satisfied

For points on a piping system with integral structural attachments, the term $C_3E_{ab}|\alpha_aT_a - \alpha_bT_b|$ shall be deleted from eqs 10, 11, and 13. Equations 9, 10, and 11 shall use the B, C, and K indices given in Table NB-3683.2-1 for "straight pipe, remote from welds or other discontinuities" and shall include the following additional terms [see also Footnote (1)]. Symbols are defined in 3.0.

Equation 9, add to left-hand side:

$$(P_L + P_b)_l = B_l W/A_l + B_L M_L/Z_{lL} + B_c M_c/Z_{lc} + Q_1/2L_1L_a + Q_2/2L_2L_b + M_T$$

Equation 10, add to S_n:

$$S_{nl} = C_T W^*/A_l + C_L M_L^*/Z_{lL} + C_c M_c^*/Z_{lc} + Q_1^*/2L_1L_a + Q_2^*/2L_2L_b + \overline{M}_T^*$$

Equation 11, add to S_p:

$$S_{pl} = [K_1(1.5 + 0.537\beta_1\beta_2\gamma) - 1]\frac{P_oD_o}{2t} + K_l[S_{nl}] + K_l E\alpha|T_l - T_p|$$

The following equation must also be satisfied:

$$S_{nl}^{**} = C_T W^{**}/A_l + C_L M_L^{**}/Z_{lL} + C_c M_c^{**}/Z_{lc} + Q_1^{**}/2L_1L_a + Q_2^{**}/2L_2L_b + \overline{M}_T^{**} \leq 3S_m$$

The value of S_m shall be the lesser of that for the pipe material or of that for the lug material.

3.0 Definitions (see Fig. 3.0)

r = mean pipe radius (in.)

t = nominal pipe-wall thickness (in.)

γ = r/t

β_1 = L_1/r

β_2 = L_2/r

L_1 and L_2 are defined in Fig. NB-3683.3

L_a = lesser of L_2 and t

L_b = lesser of L_1 and t

L_c = lesser of L_1 and L_2

L_d = greater of L_1 and L_2

C_T = $7.64(\gamma)^{1.64}\beta_1\beta_2\eta^{1.54} \geq 1.0$

C_L = $0.51(\gamma)^{1.74}\beta_1\beta_2^2\eta^{4.74} \geq 1.0$

C_c = $0.76(\gamma)^{1.90}\beta_1^2\beta_2\eta^{3.40} \geq 1.0$

$$\eta = -(X_1 \cos\theta + Y_1 \sin\theta) - \frac{1}{A_o}(X_1 \sin\theta - Y_1 \cos\theta)^2$$

X_1 = $X_o + \log_{10}\beta_1$

Y_1 = $Y_o + \log_{10}\beta_2$

Load	A_o	θ	X_o	Y_o
Thrust	2.2	40°	0	0.05
Longitudinal moment	2.0	50°	−0.45	−0.55
Circumferential moment	1.8	40°	−0.75	−0.60

[1] The value of M_c under equilibrium conditions, is not identical on the two sides of the attachment when M_l or M_c is applied to the lug. Usually the change in M_c will be negligible; if it is significant, however, the mean value is to be used in eqs 9–11.

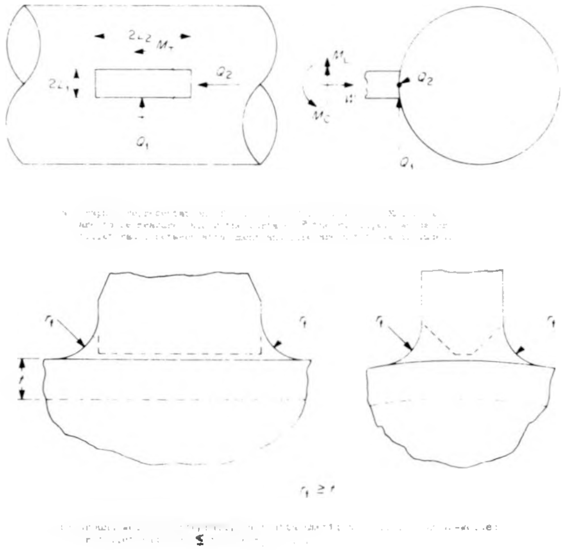

Fig. 3.0– Nomenclature Illustration

B_1 = $(2/3)C_T$
B_L = $(2/3)C_L$
B_c = $(2/3)C_c$
A_l = $4L_1L_2$ (in.^{2})
Z_{lL} = $4L_1L_2^2/3$ (in.^{3})
Z_{lc} = $4L_1^2L_2/3$ (in.^{3})
K_l = 2.0 for "as-welded" fillet welds
K_l = 1.3 for "ground" fillet welds per Fig. 3.0.

W, M_L, M_c, Q_1, Q_2, and M_T are absolute values of the sustained load amplitudes as shown in Fig. 3.0.

These are loads caused by (1) weight, (2) earthquake, considering only one-half the range of the earthquake and excluding the effects of anchor displacement due to earthquake, and (3) all other sustained mechanical loads. Units: W, Q_1, and Q_2: lb; M_L, M_c, and M_T: in.-lb

W^*, M_L^*, M_c^*, Q_1^*, Q_2^*, and M_T^* are ranges of loadings due to (1) thermal expansion, (2) anchor movements from any cause, (3) earthquake effects, and (4) all other mechanical loads which go through a range of magnitude

W^{**}, M_L^{**}, M_c^{**}, Q_1^{**}, Q_2^{**}, *and* M_T^{**} are absolute values of maximum loads from any cause

M_I = greater of $M_T/\{L_c L_a[1 + (L_c/L_a)]t\}$ and $M_T/[0.8 + 0.05(L_a/L_c)]L_c^2L_a$
M_I^* = same as M_I, except for the loading range M_T^*
T_l = average temperature of that portion of the lug within a distance of $2t$ from the surface of the pipe (°F)
T_p = average temperature of the portion of the pipe under the lug and within a distance of $\sqrt{rt}$ from the edge of the lug (°F)
$E\alpha$ = modulus of elasticity (E) times the mean coefficient of thermal expansion (α), both at room temperature (psi/°F)
P_o = range of operating pressure (psi)
D_o = outside diameter of pipe (in.)

Stress Indices for Socket-Welding Fittings

At present, the Code permits the use of ANSI standard B16.11 (Ref. 7) socket-welding fittings 2 in. ips and smaller in Class-1 piping and 4 in. ips and smaller in Class-2 and Class-3 piping. Typical B16.11 fittings are shown in Fig. 2. The Code also provides stress indices (Class-1 piping) and stress-intensification factors (Class-2 and -3 piping) for the girth fillet welds that join the fittings to the pipe, but does not provide comparable quantities for the body of the fittings.

In order to comply with a strict interpretation of the Code rules for Class-1 piping, it is necessary (since stress indices are not provided) to perform either a theoretical or an experimental stress analysis of the fitting and to include the results in the stress report [see paragraph NB-3681(*d*)]. There appears, however, to be rather widespread misunderstanding concerning the extent of coverage provided by the stress indices

PHOTO 79770

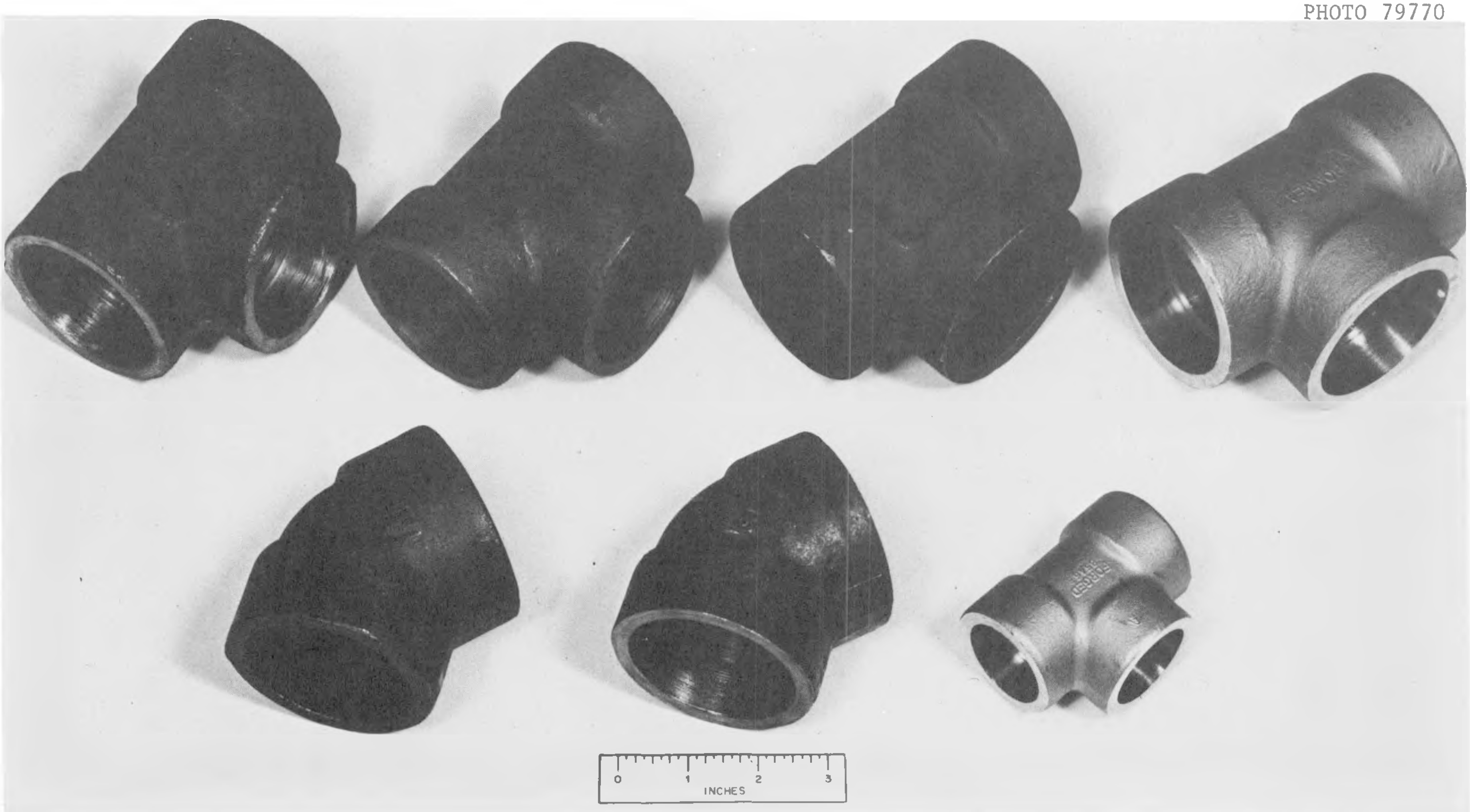

Fig. 2. Typical ANSI B16.11 fittings. Top row: 2-in.-ips, 3000-lb-class tees; bottom row: 2-in.-ips, 3000-lb-class 45° elbows and a 1-in.-ips, 3000-lb-class tee.

given in the Code (Table NB-3683.2-1) for the girth fillet welds. As a result, common practice is not to consider the body of a socket-welding fitting as a distinct item in the design stress analysis. This is not the case for other piping products, such as butt-welding tees or reducers, for which the Code provides stress indices. In these cases, it is clearly understood that qualification of the design requires separate calculations for both the fitting and the adjoining welds.

Part of the reason that stress indices have not previously been included in the Code is because there was no specific information available that could be used in their development. There were some proprietary data in company files, however, that apparently indicated that failures at socket-welding fittings always occurred in the adjoining fillet welds rather than in the body of the fitting. It was anticipated that these data would be made available, and that they would be used in the ORNL piping program for the development of appropriate indices. As it turned out, the only specific information that we were able to obtain was a small amount of unpublished ANSI B16.9-type burst-pressure test data (required by the B16.11 standard) from the Ladish Co. and one indication of a possible cyclic-pressure fatigue failure from nuclear-system field-failure reports on file at the Nuclear Safety Information Center at Oak Ridge.

Since neither adequate experimental nor analytical data were available for proper development of stress indices, the stress indices presented in our report[8] are based largely on engineering judgment and combinations of the following factors: the dimensional and burst-pressure requirements of the ANSI B16.11 standard; the standard pressure-temperature ratings of the fittings; their apparent shapes, as indicated from a small random sampling of off-the-shelf fittings; and analogies with similar butt-welding fittings.

A summary of the stress indices proposed in the report[8] for socket-welding tees, elbows, and couplings 2 in. ips and smaller is given in Table 1 along with the current stress indices from the Code for girth fillet welds for comparison. As this comparison shows, some of the indices are higher for the fittings, and some are higher for the fillet welds. Thus, if the stress indices for the fittings are realistic,

Table 1. Summary of proposed stress indices for ANSI B16.11 socket-welding fittings[a] and stress indices for girth fillet welds for comparison

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
Socket-welding fittings[a]									
Tees[b]	1.0	2.0	4.5	(*c*)	(*c*)	1.0	0.0	0.0	1.0
90 and 45° elbows	1.0	2.0	4.5	(*d*)	(*d*)	1.0	0.0	0.0	1.0
Couplings	0.5	1.0	4.5	1.0	1.0	4.5	0.0	0.0	1.0
Girth fillet welds to socket-welding fittings	0.75	2.0	3.0	1.5	2.1	2.0	1.8	1.0	3.0

[a]Socket-welding fitting made in accordance with ANSI B16.11 in nominal sizes of 2 in. ips and smaller. Applicable only if exterior contour of fitting is forged to shape and if the pressure class of the fitting is rated equal to or greater than the allowable design pressure of the attached pipe.

[b]For socket-welding tees, M_i in Code Eqs. (9) to (13) must be replaced with $M_i = M_r + M_b$, where M_r and M_b are calculated according to the rules in Footnote 5, Table NB-3683.2-1.

[c]$B_2 = 0.75C_2$ and $C_2 = (r/t)^{2/3}$, where r = mean radius and t = nominal wall thickness of equivalent pipe.

[d]$B_2 = 0.75C_2$ and $C_2 = 1.23\ (r/t)^{2/3}$, where r = mean radius and t = nominal wall thickness of equivalent pipe.

there should be loading conditions under which the fittings would be more likely to fail than the weld.

In an effort to identify which fittings (2 in. ips and smaller) would be more likely to fail before the fillet welds, we also conducted a design fatigue-life study for socket-welding fittings under cyclic moment loadings. The results of the study indicated that for most cases the fillet weld is more likely to fail than the fitting, although for several of the fittings the reverse was indicated. We also showed that if a heavier-class fitting than required by the B16.11 standard were used, cyclic-moment-loading fatigue failure should almost always occur in the weld before it would occur in the fitting.

Because of the lack of more-definitive engineering data to support the index values, we proposed that the ASME issue a code case so that the technical community could use and comment on the information without being subject to the mandatory requirements of a Code revision. This proposal was accepted by the ASME Working Group on Piping Design (SG-D) (SC-III) in September 1975, but was rejected by the Subgroup on Design (SC-III). The basis for the rejection was essentially that the new indices would require the designer to do a great deal of additional work (which he presently avoids by ignoring the letter of the Code) with little, if any, gain in assurance for a safe design. We have been asked to help draft a new proposal for ASME consideration.

TECHNICAL REPORTS

The current plan is to publish nine topical reports during the two-year period FY 1976-77 about studies for which the experimental and/or analytical work is essentially complete. These include the four experimental studies, four analytical studies, and one computer-program documentation listed below:

Experimental studies:

1. theoretical and experimental stress analyses of ORNL thin-shell cylinder-to-cylinder model 2 (Ref. 9),
2. experimental stress analysis of a flat plate with two closely spaced nozzles (Ref. 10),

3. experimental stress analyses of machined pipe elbows with specified geometric distortions (Ref. 11),
4. experimental stress analysis of a hemispherical-shell-radial-nozzle model that was tested at The University of Tennessee under subcontract,

Computer program:

5. FLANGE: A computer program for the analysis of flanged joints with ring-type gaskets (Ref. 12),

Analytical studies:

6. stresses in the bolting and flanges of ANSI B16.5 flanged joints — ASME Part A flanges,
7. stresses in flanged joints with contact outside the bolt holes — ASME Part B flanges,
8. dimensional study of ANSI standard pipe fittings, and
9. flexibility factors for small branch connections.

Two of the above reports, items 1 and 2 have been issued. Item 5 will be published next quarter, and the remaining are scheduled for publication before the end of the summer. Brief summaries of the two published studies are given below.

Stress Analysis of Cylinder-to-Cylinder Model No. 2

The report on ORNL thin-shell cylinder-to-cylinder Model 2 gives the results of a series of experimental and finite-element stress analyses for the last of four thin-shell steel models that were studied (1) to obtain quantitative fundamental information on the behavior of the intersection regions between two cylindrical shells of different diameters and (2) to evaluate our ability to correctly analyze the structure for the various different loading conditions. The analytical tool chosen for this part of the study was a flat-plate type finite-element computer program called JØINT, which was originally developed at the University of California, Berkeley, under Prof. R. W. Clough and which was later expanded and updated at ORNL.

The models were very carefully constructed to model as nearly as possible the ideal characteristics embodied in the classical theory of thin plates and shells,[13] which is the basis for most present-day stress-analysis methods. These characteristics are diameter-to-thickness ratios in the approximate range $20 \le D/T \le 100$, abrupt intersections, and no variations in diameter or wall thickness. End restraints are assumed to be sufficiently remote to have no effect on the stresses or strains near the intersection.

A photograph of Model 2 and a table showing the major dimensions of all four models in the test series are shown in Fig. 3. The dimensionless

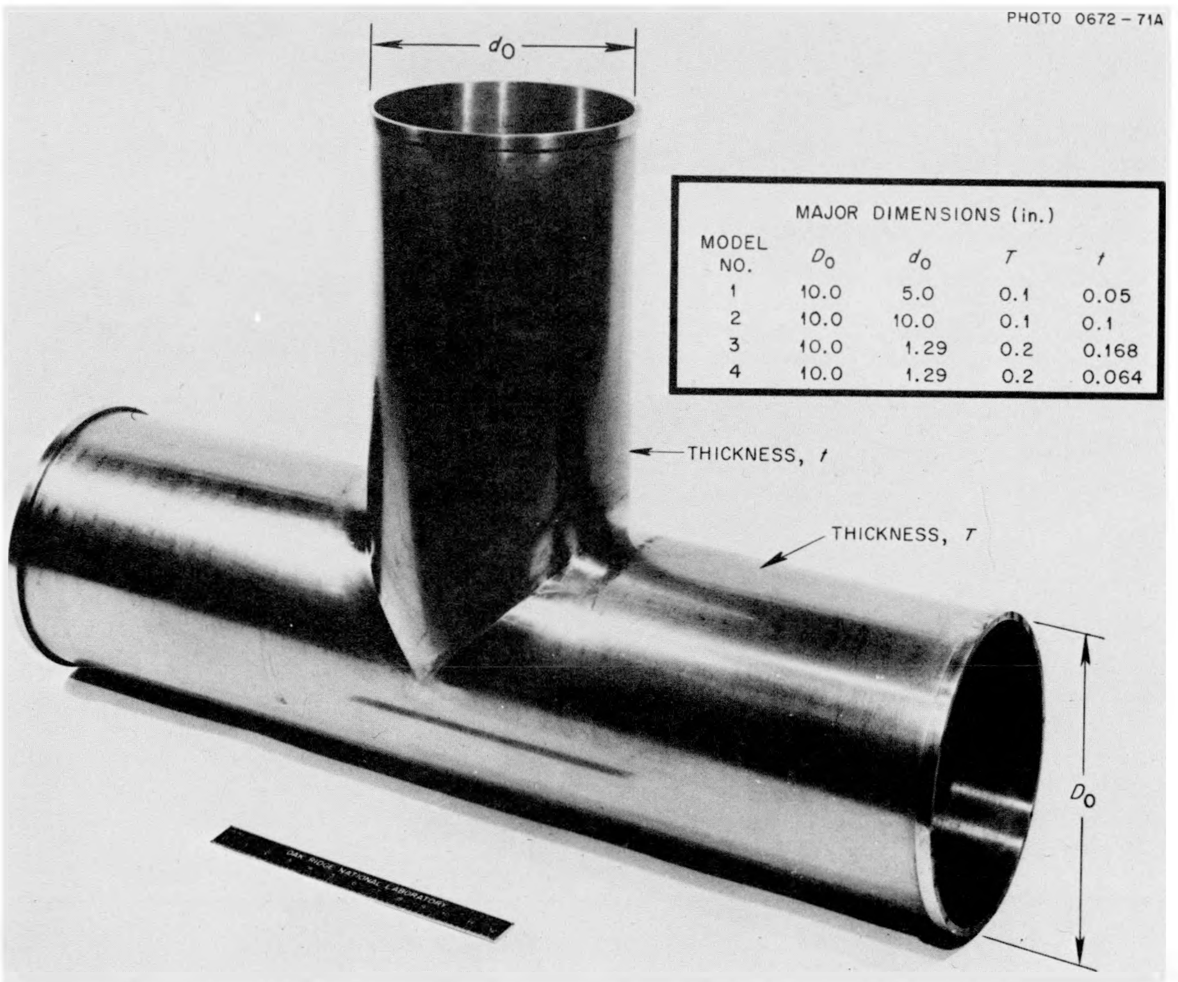

Fig. 3. Thin-shell cylinder-to-cylinder Model 2 and table showing dimensions of all four models in the test series. (For conversion to SI units, 1 in. = 2.54 cm).

parameters for Model 2 are d/D = 1, D/T = 100, and t/T = 1, where D and T are the outside diameter and wall thickness of the main cylinder or run and d and t are similar dimensions for the nozzle or branch. The model is an idealized thin-shell structure in the sense that there are no transitions, fillets, or other reinforcements in the junction region.

Model 2 was instrumented with 152 three-gage strain rosettes located in the crotch of the intersection and along the longitudinal (0°) and transverse (270°) planes of symmetry, as shown in Fig. 4. Half of the

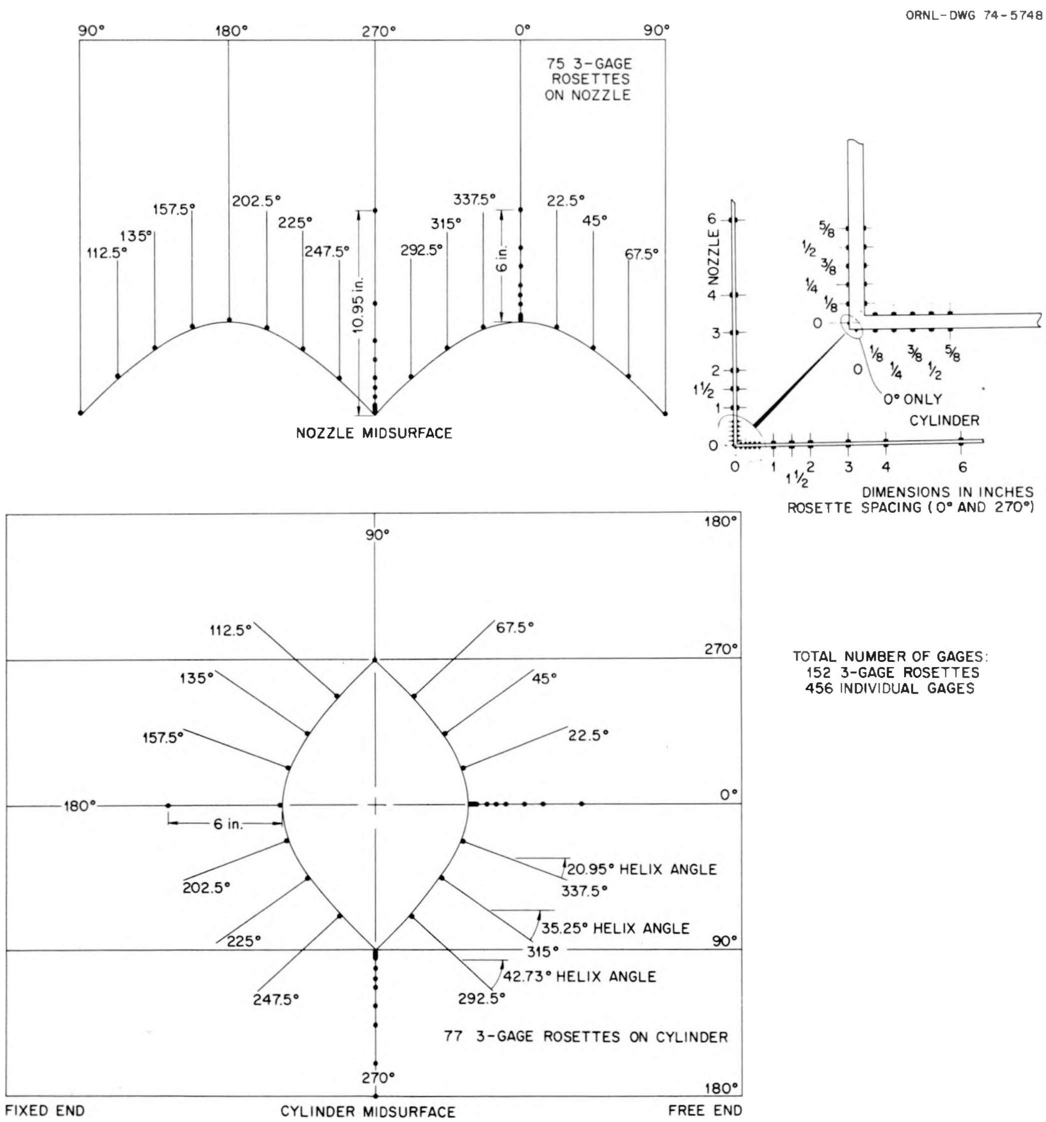

Fig. 4. Strain-gage layout for the cylinder-to-cylinder Model 2. (For conversion to SI units, 1 in. = 2.54 cm).

gages were mounted on the inside surface directly through the wall from corresponding gages on the outside surface, in order to measure bending stresses. The number of gages was sufficient to provide a good description of the stress distributions from the various different loadings.

The model was tested with 16 different loading conditions: 13 with one end of the run rigidly fixed to the loading frame and 3 with both ends of the run fixed. The different loadings and maximum load levels identified in Table 2 are for internal pressure, moment loadings on the nozzle

Table 2. Applied loads and nominal stress levels

Loading case	Load level	Nominal membrane stress [MPa (psi)]
Internal pressure	410 kPa (60 psi)	20.5 (2970)
Out-of-plane moment, M_{XN}	1 100 N·m (10,000 in.-lb)	8.96 (1300)
Torsional moment, M_{YN}	1 800 N·m (16,000 in.-lb)	7.17 (1040)
In-plane moment, M_{ZN}	1 700 N·m (15,000 in.-lb)	13.4 (1950)
In-plane force, F_{XN}	5 300 N (1200 lb)	15.6 (2260)
Axial force, F_{YN}	17 800 N (4000 lb)	8.89 (1290)
Out-of-plane force, F_{ZN}	2 700 N (600 lb)	9.17 (1330)
Torsional moment, M_{XC}	2 300 N·m (20,000 in.-lb)	8.96 (1300)
Out-of-plane moment, M_{YC}	6 800 N·m (60,000 in.-lb)	53.8 (7800)
In-plane moment, M_{ZC}	2 700 N·m (24,000 in.-lb)	21.5 (3120)
Axial force, F_{XC}	36 000 N (8000 lb)	17.7 (2570)
In-plane force, F_{YC}	4 400 N (1000 lb)	17.4 (2530)
Out-of-plane force, F_{ZC}	5 300 N (1200 lb)	21.0 (3040)
Out-of-plane moment with restraints, M_{XN}	1 100 N·m (10,000 in.-lb)	8.96 (1300)
In-plane moment with restraints, M_{ZN}	1 700 N·m (15,000 in.-lb)	13.4 (1950)
Axial force with restraints, F_{YN}	17 800 N (4000 lb)	8.89 (1290)

(M_{XN}, M_{YN}, M_{ZN}), direct-force loadings on the nozzle (F_{XN}, F_{YN}, F_{ZN}), and similar loadings on the run (M_{YC}, ..., F_{XC}, ...).

The 16 loading cases were analyzed theoretically using the finite-element shell program JØINT. The finite-element representation employed 993 nodes and 957 elements. The mesh layout shown in Fig. 5 was developed manually and was arranged so that lines of nodes corresponded to the lines of strain gages in the experimental model.

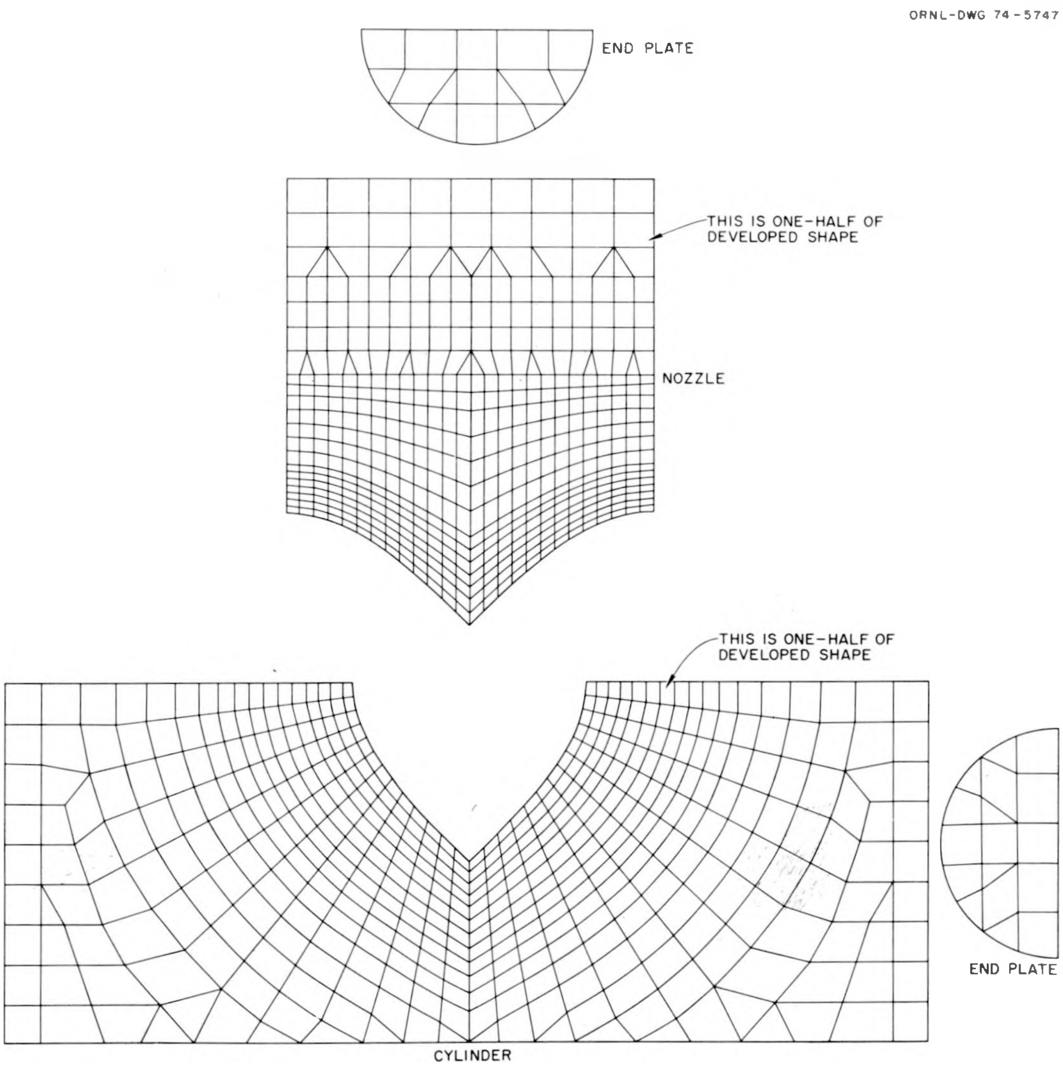

Fig. 5. Finite-element idealization for Model 2.

Table 3 gives a summary of the maximum-stress ratios and maximum-stress locations for the model. Generally good agreement was obtained between the experimental values and those based on finite-element predictions. The report contains complete listings of the stress and strains for each loading condition as well as graphical comparisons between the experimental and analytical results. It is expected that these data will be used extensively in the future as benchmark reference information.

Experimental Stress Analysis of a Flat Plate with Two Closely Spaced Nozzles

The report[10] on the experimental stress analysis of a flat plate with two closely spaced nozzles gives results from the eighth model in a series of flat-plate models that were tested at The University of Tennessee and at Auburn University under subcontract. Dimensional information and model identification are given in Table 4; the model discussed in Ref. 10 is identified as No. 7P.

The flat-plate models in this series were studied to develop a better understanding of the interaction behavior of closely spaced nozzles in pressure vessels under internal pressure or external loadings applied through the nozzles. When the diameter of the opening is small relative to the diameter of the pressure vessel, the curvature of the vessel can be ignored. Consequently, flat plates are convenient models for study.

Flat-plate Model No. 7P, shown schematically in Fig. 6, was instrumented with 160 three-gage strain rosettes along stringer lines located on both surfaces of the plate and on one of the nozzles. Stringer line 5 is located on the top surface of the plate at 0°; stringer 1 is at 90°. Because of model and loading symmetries, it was only necessary to instrument one of the nozzles, identified in Fig. 6 as nozzle 1. Test loadings included three biaxial planar tension loads on the plate to simulate the effects of internal pressure in hemispherical and cylindrical shells, axial thrust loads applied separately to the two nozzles, and bending moment loads applied in the plane and normal to the plane of symmetry containing the two nozzles.

Table 3. Summary of maximum stress ratios and maximum stress locations for ORNL Model No. 2

Loading case	Experimentally determined maximum stress		Theoretical maximum stress		Overall agreement between theory and experiment
	Stress ratio[a]	Location[b]	Stress ratio[a]	Location[b]	
Internal pressure	9.0	Outside nozzle, 180°	7.7	Outside cylinder, 0°	Excellent, excellent
M_{XN}, out-of-plane moment on nozzle	15.8	Inside cylinder, 247°	17.8	Inside nozzle, 249°	Excellent, good
M_{YN}, torsional moment on nozzle	31.3	Inside cylinder, 247°	37.5	Outside cylinder, 256°	Poor, good
M_{ZN}, in-plane moment on nozzle	11.0	Outside nozzle,[c] 270°	15.2	Inside nozzle, 256°	Good, excellent
F_{XN}, in-plane force on nozzle	13.4	Inside nozzle,[c] 270°	17.8	Inside nozzle, 256°	Excellent, excellent
F_{YN}, axial force on nozzle	13.4	Outside nozzle, 180°	17.2	Inside cylinder, 256°	Good, poor
F_{ZN}, out-of-plane force on nozzle	15.9	Inside cylinder, 247°	24.3	Outside cylinder, 256°	Good, excellent
M_{XC}, torsional moment on cylinder	23.9	Inside cylinder, 292°	37.5	Outside cylinder, 284°	Poor, excellent
M_{YC}, out-of-plane moment on cylinder	4.5	Inside nozzle,[c] 270°	5.9	Outside nozzle, 0°	Fair, excellent
M_{ZC}, in-plane moment on cylinder	14.9	Inside nozzle,[c] 270°	10.1	Inside nozzle,[c] 270°	Good, excellent
F_{XC}, axial force on cylinder	14.4	Inside nozzle,[c] 270°	14.7	Inside nozzle,[c] 270°	Good, good
F_{YC}, in-plane force on cylinder	15.7	Inside nozzle,[c] 270°	10.7	Inside nozzle,[c] 270°	Good, excellent
F_{ZC}, out-of-plane force on cylinder	6.9	Inside nozzle,[c] 270°	9.9	Outside cylinder, 256°	Fair, good
M_{XN}, out-of-plane moment on nozzle with restraints	14.9	Inside nozzle,[c] 270°	11.8	Outside cylinder, 284 and 256°	Good, excellent
M_{ZN}, in-plane moment on nozzle with restraints	8.0	Outside nozzle,[c] 270°	12.5	Inside nozzle, 284°	Excellent, excellent
F_{YN}, axial force on nozzle with restraints	31.0	Inside nozzle,[c] 270°	16.0	Inside nozzle,[c] 270°	Good, poor

[a] Ratio of maximum absolute principal stress value to nominal stress value.

[b] Maximums all occurred at the junction, except where noted.

[c] Maximums not at junction; at approximately 2.5 cm (1.0 in.) from junction on transverse plane.

Table 4. Experimental investigations of hole and nozzle clusters in 91.44 × 91.44-cm (36 × 36-in.) flat plates 0.953 cm (0.375 in.) thick

Opening	Model number
Unpierced	1P[a]
Unpierced	2P[a,b]
One hole	3P1[a,c]
Two holes	3P2[a,c]
Three holes	4P[a,c]
Five holes	5P[a,c]
One nozzle	6P[b,c]
Two nozzles	7P[b,c]
Five nozzles	10P[c,d]

[a]Models tested at The University of Tennessee.

[b]Models tested at Auburn University.

[c]Holes 6.668 cm (2.625 in.) in diameter; nozzles are 6.668 cm (2.625 in.) in outer diameter; the wall thickness of the nozzles is 0.635 cm (0.250 in.).

[d]Flat-plate model to be tested.

The report describes in detail the loading procedures and data-acquisition system for the experiment, and contains complete listings of the measured strains and resulting stresses for each of the nine applied loadings. These data are presented in both graphical and tabular formats for each reference. Table 5 gives a summary of the maximum normalized stresses and their locations, as well as comparisons with theoretical results presented in Ref. 14. The values given in Table 5 are normalized with respect to a nominal stress (σ_{nom}). For biaxial planar loads on the plate, σ_{nom} is the biaxial stress for an unperforated plate. For thrust

ORNL-DWG 74-9413R2

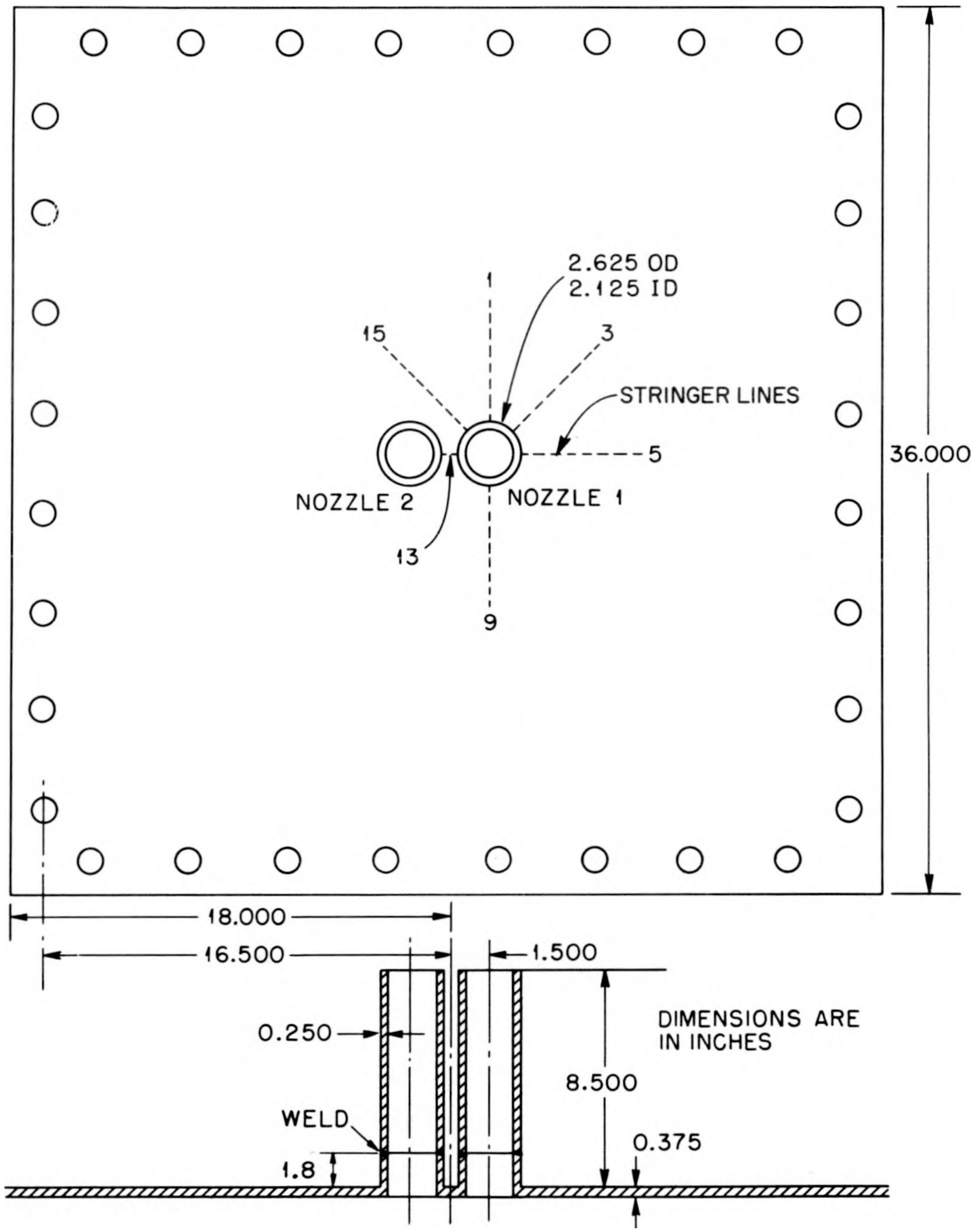

Fig. 6. Schematic of flat-plate Model No. 7P with two nozzles attached. (For conversion to SI units, 1 in. = 2.54 cm).

loading, σ_{nom} = F/A, where F is the load and A is the cross-sectional area of the nozzle. For moment loading, σ_{nom} = M/Z, where M is the moment load and Z is the section modulus of the nozzle. Identification of the various load cases in Table 5 is shown schematically in Fig. 7. The data given in this report will be used in the development of improved design rules for closely spaced nozzles in reactor pressure vessels, as well as in the development of improved stress-analysis methods.

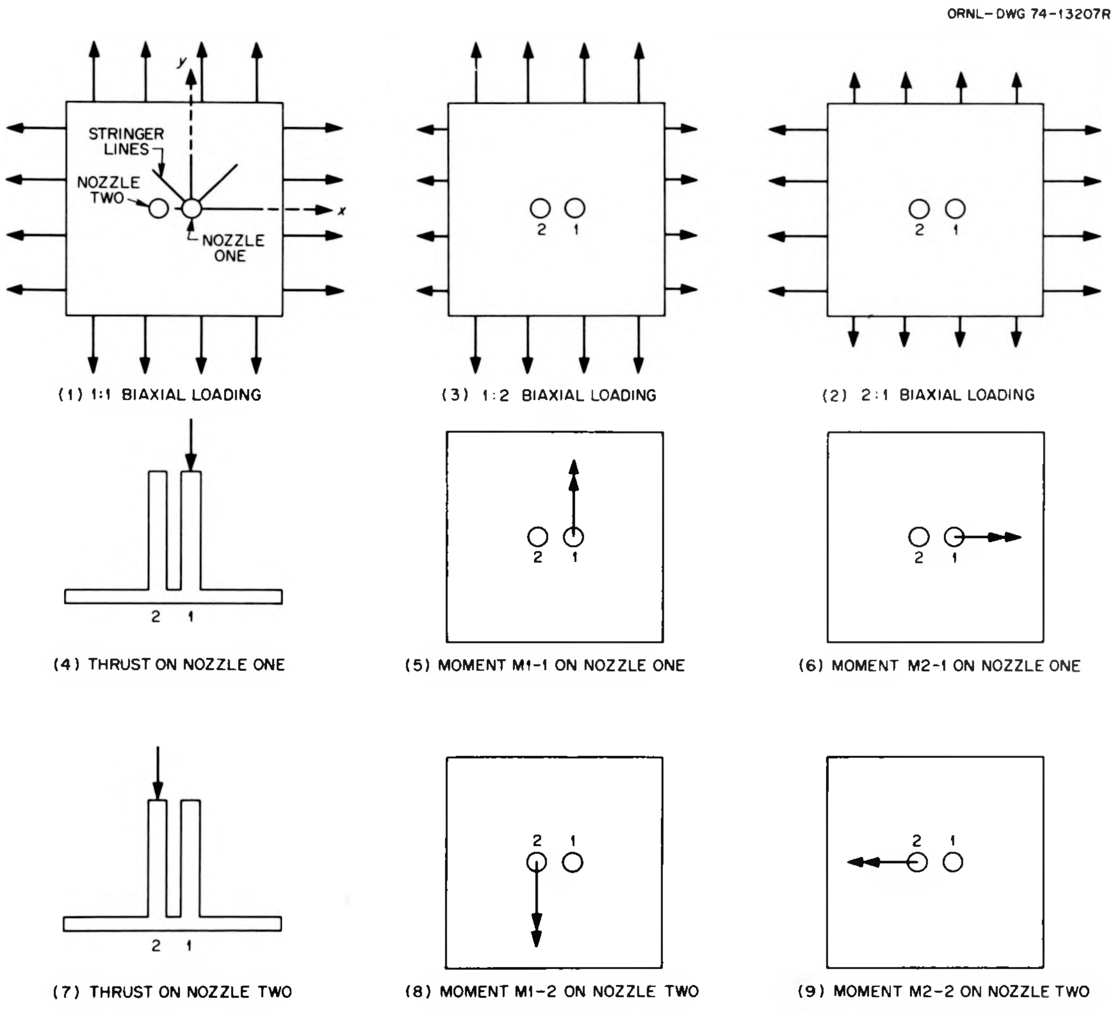

Fig. 7. Loadings applied to the flat-plate model with two nozzles attached.

Table 5. Summary of maximum normalized stresses

Model	Load case per Fig. 7	Stress identification[a]	Experimentally determined maximum normalized stress (absolute value): Nozzle Value	Experimentally determined: Nozzle Location[b]	Experimentally determined: Plate Value	Experimentally determined: Plate Location[b]	Theoretical maximum normalized stress (absolute value): Nozzle Value	Theoretical: Nozzle Location[b]	Theoretical: Plate Value	Theoretical: Plate Location[b]
Two-nozzle Model 7P	1:1	σ_t	3.0[c]	I-180	2.8	I-180	2.6	0-180	2.8	I-180
		σ_m	0.65	I-180	*d*		1.65	0-180	*d*	
	1:1/2	σ_t	2.2[c]	I-90	2.2	I-90	1.75	I-90	2.2	I-90
		σ_m	0.75	I-90	*d*		0.95	0-180	*d*	
	1/2:1	σ_t	3.1[c]	I-180	3.0	I-180	2.6	0-180	2.9	I-180
		σ_m	0.9	I-180	*d*		1.5	0-180	*d*	
	Thrust on nozzle one	σ_t	24.0[c]	0-0	18.0[c]	0-135	9.3	I-180	17.5	I-0
		σ_m	29.0[c]	0-90	19.0[c]	0-45	29.0	0-90	20.0	0-0
	M1-1	σ_t	6.0[c]	0-0	3.5[c]	0-0	2.5	I-180	3.5	I-180
		σ_m	7.5[c]	0-0	4.2[c]	0-0	7.25	0-0	3.5	0-0
	M2-1	σ_t	3.5[c]	0-90	3.4[c]	0-90	2.0	I-90	2.75	I-90
		σ_m	7.5[c]	0-90	4.2[c]	0-90	6.25	0-90	2.95	0-90
	Thrust on nozzle two	σ_t	10.0[c]	I-180	12.8	I-180	7.5	0-90	12.5	I-180
		σ_m	19.0[c]	0-90	13.0	0-180	20.0	I-180	13.0	0-90
	M1-2	σ_t	2.3[c]	0-180	3.0	I-180	1.75	I-180	2.6	I-180
		σ_m	3.8[c]	0-180	3.4	0-180	3.9	I-180	2.4	0-180
	M2-2	σ_t	1.5[c]	0-45	1.5	0-45	1.0	0-135	1.0	0-135
		σ_m	3.0	0-45	2.0	I-135	3.4	0-135	1.3	0-135

[a] σ_t = tangential stress (circumferential to the nozzle); σ_m = meridional stress.

[b] Initial letters indicate surface (I = inside; O = outside); numerals indicate stringer (in degrees, see Fig. 6).

[c] Extrapolated value.

[d] Maximum absolute value is that for the plate remote from opening (= 1.0), which is of limited significance as a "maximum value."

SPHERICAL-SHELL STUDIES

The present plan for studies on spherical-shell geometries during the FY 1976-77 period includes publication of a summary report on existing experimental and analytical data on the stresses in the vicinity of nozzles in spherical pressure-vessel heads and an analytical stress analysis and an experimental-data-comparison study for a series of radial nozzle-to-spherical-shell models. Neither study has been active during the last six months, although we plan to restart work on the summary report in January. A first draft was completed last fiscal year and sent to members of the PVRC Subcommittee on Reinforced Openings and External Loadings for their review and comment. The analytical stress-analysis study is currently projected to be started in FY 1977.

Several experimental and analytical studies are projected for FY 1978, depending on the availability of funds and the needs of the Nuclear Regulatory Commission. These include completing both experimental and analytical studies of isolated and closely spaced reinforced nozzles in spherical shell structures. Data from these studies are needed for the assessment of current Code design rules for nuclear-reactor pressure vessels.

CYLINDRICAL-SHELL STUDIES

Program activities under the general category of cylindrical-shell studies include participation in the PVRC Subcommittee on Reinforced Openings and External Loadings (S/C ROEL), particularly in regard to studies on nonradial nozzles and upper- and lower-bound limit-load studies of radial nozzles in cylindrical pressure vessels, as well as ORNL-based studies for isolated nozzles and for two and three closely spaced nozzles in cylindrical pressure vessels. The ORNL-based studies this year (FY 1976) are concerned with analytical studies for isolated nozzles and for two closely spaced nozzles in vessels under internal-pressure loadings. The status of these two projects as of December 31, 1975, is discussed below.

Isolated Nozzles

We are presently conducting a parameter study on isolated reinforced nozzle-to-cylindrical-shell attachments using the 3-D finite-element computer program CORTES-SA* that was developed for us at the University of California, Berkeley.[15] This program was designed specifically for analyzing tee-joint structures (perpendicular cylinder-to-cylinder connections and ANSI B16.9 tees) under essentially arbitrary mechanical loadings that include internal pressure and force and moment loads, and location- and/or time-dependent temperature distributions. The program contains an automatic mesh-generation package with a number of input-variable options that enable the user to model a wide variety of nozzle-connection designs using a minimum amount of input data. Output from the program may be saved on storage devices and subsequently displayed graphically using several different plotting packages.

During the first part of this year a number of modifications were incorporated into CORTES to improve the utility of the progr and its operating efficiency. The elastic program CORTES-SA was checked out on a number of realistic tee-joint-structure models and is now fully operational on the ORNL IBM computer system. All five programs were also sent to the Argonne Code Center, Argonne National Laboratory, from which they may be obtained upon request. Later this year we plan to complete the check-out phase for CORTES-EP and to send updated versions to Argonne.

The isolated-nozzle parameter study consists of 25 models with dimensional parameters within the range $10 \le D/T \le 100$ and $0.08 \le d/D \le 0.5$, as shown in Table 6. All of the nozzles are designed according to one of the sketches shown in Fig. 8, which was abstracted from paragraph NB-3338.2 of the Code. The six U models listed in Table 6 are essentially unreinforced except for the minimum fillet (r_2) at the junction, as shown in Fig. 8(*d*). The fourteen S1 models (so-called "standard" reinforcing)

---

*CORTES (for California-Oak Ridge TEe Stress Analysis Package) is a package of five finite-element computer porgrams (-SA, EP, THFA, SHFA, TSA) for the elastic-stress analysis, elastic-plastic-stress analysis, transient-heat-flow analysis, and thermal-stress analysis, respectively, of tee joints. Although the codes were initially designed for analyzing tees that conform with the ANSI B16.9 standard, they have sufficient input flexibility for use with other tee-joint geometries.

Table 6. Dimensional parameters for the study of isolated nozzles in cylindrical vessels

D/T[a]	d/D[b]			
	0.08	0.16	0.32	0.50
10	(0.2530)[c] U[d] S1[d] P30[d]	(0.5060) S1	(1.0119) S1 P30	(1.5811) U S1 (*e*)
20	(0.3578) S1	(0.7155) S1	(1.4311) S1 P30	(2.2361) U S1 (*e*)
40	(0.5060) S1	(1.0119) S1	(2.0239) S1 P30	(3.1623) U S1 (*e*)
80	(0.7155)	(1.4311)	(2.8622)	(4.4721) U S1 (*e*)
100	(0.8000)	(1.600)	(3.2000) P30	(5.0000) U S1 (*e*)

[a]D/T is the inside diameter-to-thickness ratio of the cylindrical vessel.

[b]d/D is the ratio of the inside diameters in the nozzle and cylindrical vessel.

[c]Numbers given in parentheses are values for the parameter $(d/D)\sqrt{D/T}$.

[d]U, S1, and P30 refer to nozzle designs shown in Figs. 8(*d*), 8(*a*), and 8(*c*), respectively.

[e]P30 models geometrically impossible for these parameter values.

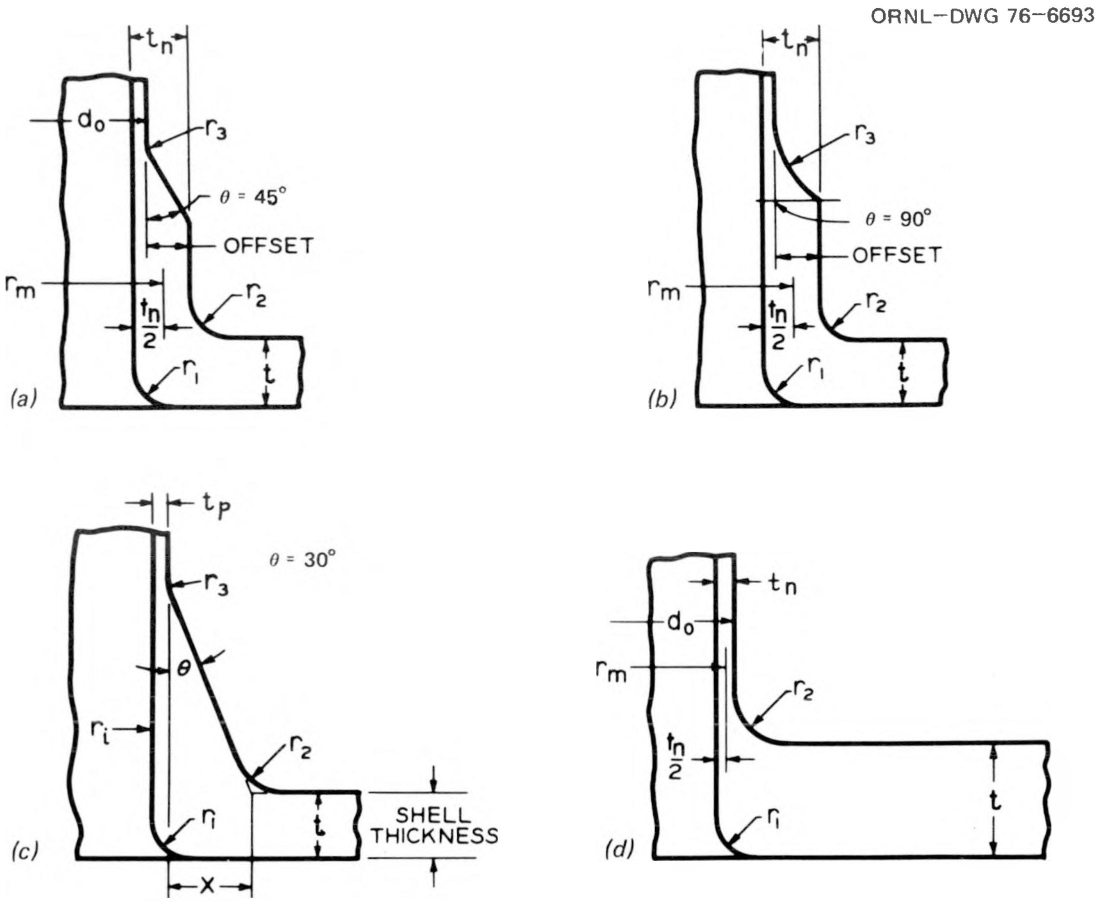

Fig. 8. Nozzle dimensions for various standard reinforcement designs.

depicted in Fig. 8(*a*) and the five P30 models (30° pad reinforcing) depicted in Fig. 8(*c*) are fully reinforced according to the rules of paragraphs NB-3332 through NB-3334. All 25 models will be analyzed for internal pressure loading this year and for external moment loadings applied to the nozzle next year.

Five of the U models have been analyzed to date (with internal-pressure loading), and we expect to complete the numerical portion of the study by the first of May. Our current plan is to then correlate the data and publish a report by the end of September.

Two Closely Spaced Nozzles

Studies this fiscal year on closely spaced nozzles in cylindrical pressure vessels are directed toward the development of a finite-element computer program that will enable us to conduct parameter studies on the spacing between multiple reinforced or unreinforced nozzles. The current Code requirements, given in Subarticle NB-3300 for Class-1 vessels, are somewhat ambiguous and may be in need of considerable revision. However, at present there are no theoretical data and only limited experimental data upon which to base more-realistic design rules. The purpose of our study is to provide sufficient data, in parametric form, to support the development of improved design rules.

The plan is to first develop a computer program, including an automatic mesh generator, for the stress analysis of two identical nozzle configurations radially attached to a cylindrical vessel under internal-pressure loading. The program will then be used to conduct parameter studies and at the same time will be extended to include external loadings on the nozzles and to include up to three nozzles rather than two.

Initially, the program will include provisions for automatically generating a finite-element model with two identical nozzle configurations of the type identified in Figs. 8(*a*) and 8(*d*). The nozzles may be spaced arbitrarily close together (within a specified parameter range) and positioned either on a longitudinal plane (i.e., along the length) or on a transverse plane (i.e., around the circumference) of a cylindrical vessel. The program will use variable 8- to 20-node isoparametric solid elements and an advanced matrix-solving routine and will compute stresses, strains, and displacements for the internal-pressure-load case. Output, in both global and local-element coordinate systems, will be saved along with the finite-element-model description on a suitable storage device to facilitate interpretation and presentation of the results.

A subcontract has been signed with Mechanics Research, Inc., of Los Angeles to develop the computer program and to verify and demonstrate its performance by analyzing three models and comparing the results with experimental results in the literature. We plan to complete this phase of the study by the end of June and then to install the program at ORNL and to prepare the report by the end of the summer (Sept. 30, 1976). The project will be completed in FY-1977.

ELBOW AND CURVED-PIPE STUDIES

Three items are scheduled for completion this fiscal year under the category of elbow and curved-pipe studies. These are (1) a finite-element parameter study on the stresses and flexibility of elbows as functions of bend angle and restraint against deformation provided by components, such as pipe or flanges welded to the elbow; (2) a summary and evaluation of experimental limit-load data for elbows under external force and moment loadings; and (3) a cyclic internal-pressure fatigue test of a 10-in. ips machined model elbow.* Of these three, only the fatigue test has been active so far this year. Plans are to begin work on the other two shortly after the first of the year.

Elbow Fatigue Test

Current Code design rules do not specify dimensional tolerances for elbows other than those required by the manufacturing standards ANSI B16.28, ANSI B16.9, and MSS-SP48 on the assumption that either the standards provide sufficient control or that significant variations will be covered by the Code design-analysis rules. In the case of welding-end elbows, the standards permit significant variations in wall thickness and out of roundness, but the Code analysis procedure considers only out of roundness and that only in the fatigue-life evaluation for cyclic pressure loading (Footnote 1, Table NB-3683.2-1). These rules, however, are based on analytical elastic studies for straight pipe.[16] Thus, in order to properly assess the current design rules, experimental data on the effects of out of roundness and variations in wall thickness are needed from tests of carefully constructed elbow models.

---

*This elbow is one of the models — ME-2 — for which the elastic-response experimental-stress-analysis tests are discussed in Ref. 11.

The four 10-in.-ips machined-elbow models listed in Table 7 have been experimentally stress analyzed for internal-pressure, external-moment, and combined pressure and moment loadings;[12] currently, model ME-2, the out of round elbow, is being fatigue tested with a cyclic internal pressure. Each test model consisted of an elbow welded to two rather short [450-mm (18-in.)] 10-in. sched-40 A106 Grade-B carbon-steel pipe extensions capped and flanged at the ends, as shown in Fig. 9. To ensure proper fit-up for welding, the pipe stubs for the ovalled and thinned

Table 7. Machined-elbow test models

Model number	Identification	Model parameters
ME-1	Ideal torus	90°, long radius, 27.219-cm (10.716-in.) mean outer diameter, 0.991-cm (0.390-in.) mean wall thickness
ME-2[a]	Ovalled torus	90°, long radius, 27.252-cm (10.729-in.) mean outer diameter, 0.975-cm (0.384-in.) mean wall thickness, flattened 5.7% of the diameter[b]
ME-3	Thinned elbow	90°, long radius, 27.315-cm (10.754-in.) mean outer diameter, 0.709-cm (0.279-in.) minimum wall thickness, 1.05-cm (0.414-in.) maximum wall thickness[c]
ME-4	Ovalled and thinned	90°, long radius, 27.264-cm (10.734-in.) mean outer diameter, 0.711-cm (0.280-in.) minimum wall thickness, 1.10-cm (0.433-in.) maximum wall thickness, flattened to 5.1% of the diameter[b]

[a]This model is being fatigue tested with a cyclic internal pressure loading.

[b]Approximately elliptical cross sections were formed with mean major-to-minor-diameter ratios of 1.059 for ME-2 and 1.053 for ME-4 with each major axis lying in the plane of the bend. Percentage of flattening is calculated according to the formula $[(D_{max} - D_{min})/D_{mean}]100$.

[c]The eccentricity of the bore is away from the center of the bend so that the minimum wall thickness is along the back of the elbow (the extrados).

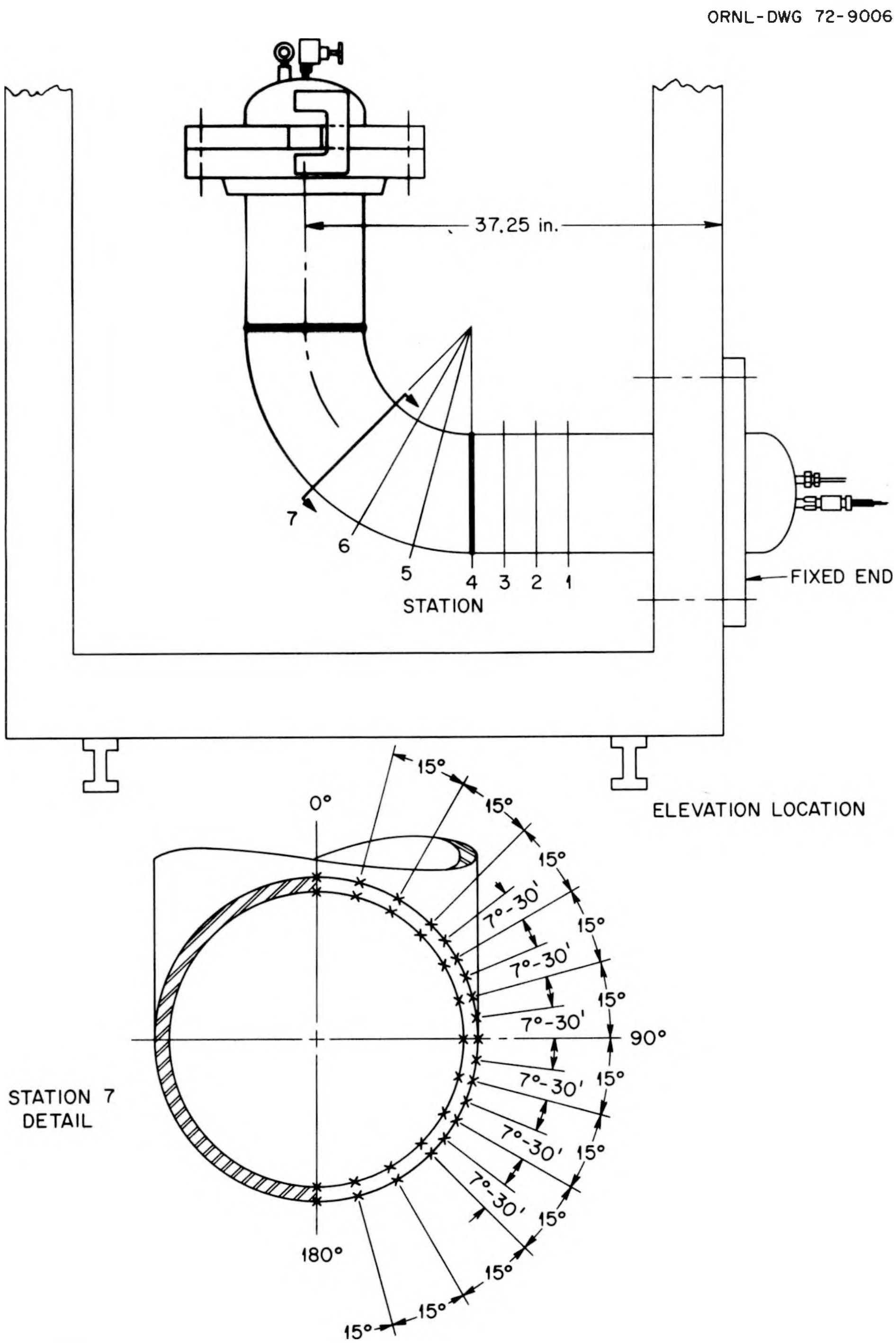

Fig. 9. Strain-gage layout for machined-elbow models.

models were flattened and/or tapered for about one pipe-diameter length. Each model was instrumented with about 90 three-gage strain rosettes, most of which were located on the lines marked "station 1" through "station 7." Station 7, located at 45°, was instrumented with gages on both the inside and outside surfaces.

Model 2 was selected for fatigue testing in order to provide direct experimental evidence for use in assessing the Code design-analysis procedure for cyclic pressure loading. The test loading range of from 0 to 15 200 kPa (2200 psi) was fixed to be as high as possible without exceeding about 90% of the yield strength of the attached pipe stubs. At 15 200 kPa (2200 psi), the nominal circumferential stress in the pipe is 215 800 kPa (31,300 psi), whereas the minimum yield strength of the pipe material is 241 000 kPa (35,000 psi). The corresponding peak stress intensity in the elbow (based on nominal dimensions, 5.7% measured out of roundness, and the stress indices given by the Code) is 652 000 kPa (94,500 psi). This is essentially the same value determined from the results of the strain-gage elastic-response tests. Under these conditions, the Code-allowable design life is about 5000 cycles.

An allowable design life of 5000 cycles implies an actual fatigue life of somewhat greater than 100,000 cycles, which is beyond our present fatigue-testing capabilities. However, since the Code analysis procedure is based on elastic methods of analysis, whereas the real behavior of the structure would be elastic-plastic, useful information on the expected fatigue life can be obtained by observing the shakedown behavior during cyclic loading, and a much shorter test can be justified. For this purpose, ten of the strain-gage rosettes, five on the inside and five on the outside surface, located at station 7 were monitored during the first 17 loading cycles of the fatigue test. Strain readings were taken at 1380-kPa (200-psi) increments during both loading and unloading.

Strain readings taken during the first few loading cycles showed that, as expected, the elbow deformed considerably upon initial loading and became less out of round. After the second loading cycle, the strain gages indicated a maximum elastic-stress-intensity range of about 580,000 kPa (84,000 psi), which is about 90% of the value predicted by the Code formulas. If this number were used with the Code fatigue curves, the allowable design life would be about 8000 cycles, and the expected life would be greater than 200,000 cycles. Thus, these test results tend to confirm the Code design-analysis predictions. Since both the Code analysis and the cyclic-strain data indicate that the allowable design life of 5000 cycles is conservative, we plan to run the test for 50,000 cycles in order to be on the safe side. Although 50,000 cycles gives a safety factor of only 10, if the elbow survives with no indication of fatigue-crack initiation, we may be able to conclude that the Code procedure is adequate. To date, the test assembly has been subjected to about 30,000 loading—unloading cycles.

ANSI-B16.9-TEE STUDIES

ANSI B16.9 tees are a class of piping tee joints that are manufactured according to the requirements of the ANSI Standard B16.9 "Wrought Steel Butt Welding Fittings." These fittings are characterized by a smooth transition region, which is normally rather heavily reinforced during the forging process. They are different from other types of tee joints in that the overall dimensions (height, length, and diameters at the welding ends) are controlled and in that the fittings are required to be able to pass a specified burst-pressure test. Such standard tees are sold commercially by several manufacturers.

At the beginning of our program, relatively little structural-behavior information for these tees was available in the literature upon which stress indices could be based. However, since tees are important components in most piping systems and since B16.9 tees in particular had been used successfully for many years, it was almost mandatory that the Code include specific design guidance for this class of fittings, as well as for other fittings for which more information was available. Stress indices, based largely on engineering judgment, were therefore put into the Code,[17] and the ORNL program included plans for rather extensive experimental and analytical studies on the 17 tees identified in Table 8 to provide basic information for use in qualification of the Code design-analysis procedures. Many, but not all, of the studies identified in Table 8 have been completed. The photoelastic studies were completed and reported,[18] computer codes were developed[15,19,20,21] and are in use, and experimental tests have been completed on 10 of the 17 tees.[22-32]

To date, we have accumulated an extensive amount of experimental and analytical data on the elastic-response and fatigue-life behavior of ANSI B16.9 tees, but these data have not been properly correlated or used to critically examine the Code design rules. Our plan for the two-year period FY 1976—1977 is (1) to summarize and evaluate these data, (2) to conduct empirical studies for single and combined loadings for the purpose of evaluating the current design rules, (3) to conduct analytical finite-element studies and to compare the results with the experimental data to qualify the computer codes and to extend the range of the baseline data, and (4) to conduct tests on the two 12-in.-ips tees T-5 and T-9. During the past six months, work has been under way on the first item. We plan to complete item (1) and begin work on item (2) this year.

Summary and Evaluation of Elastic-Response and Fatigue-Test Data

Two series of elastic-response and fatigue tests on ANSI B16.9 tees have been conducted under this program. The first series, consisting of the five 12-in.-ips tees identified in Table 9, was tested under subcontract at Southwest Research Institute (SwRI). The second series, consisting of the five 24-in.-ips tees identified in Table 10, was tested under subcontract in the Nuclear Components Test Laboratory, Combustion Engineering, Inc., (CE) at Chattanooga. The final draft of a summary and evaluation report on the SwRI test series has been completed[33] with the assistance of Mechanics Research, Inc., (MRI), and we are in the process of preparing the report for publication as an ORNL document; MRI is currently preparing a similar report for the CE test series.

Each of the five tees tested at SwRI was instrumented with approximately 225 three-gage strain rosettes located in two quadrants of the tee

Table 8. Experimental and analytical studies for ANSI B16.9 tees

Tee No.	Identification[a]					Experimental tests				Analytical studies		
	Run (in.)	Branch (in.)	Wall (sched)	Steel[b]	Manfacturer	(*c*)	(*d*)	(*e*)	(*f*)	(*g*)	(*h*)	(*i*)
T-1	6	6	40	S	I	X	T1-P				X	
T-2	6	6	160	C	I	X	T2-P				X	ΔT
T-3	6	6	40	S	I	X	T3-MOP				X	
T-4	12	12	80	C	I	X	T4-M				X	
T-5	12	12	80	C	II	X	T5-P				X	
T-6	12	12	80	C	III	X	T6-M	Ph6-M	TS-6	FE-6	X	
T-7	12	12	160	S	II	X	T7-M	Ph7-P		FE-7	X	
T-8	12	6	40	S	II	X	T8-M	Ph8-M	TS-8	FE-8	X	ΔT
T-9	12	6	160	S	II	X	T9-P	Ph9-P			X	
T-10	24	24	40	C	III	X	T10-M				X	
T-11	24	24	160	C	III	X	T11-P				X	ΔT
T-12	24	10	40	C	III	X	T12-P				X	
T-13	24	10	160	C	III	X	T13-P				X	ΔT
T-14	12	6	40	S	III	X					X	
T-15	12	6	40	S	I	X	T15-M				X	
T-16	24	24	10	S	III	X	T16-M				X	
T-17	24	10	10	S	III	X	T17-M				X	

[a]Dimensions are nominal iron pipe sizes (ips).

[b]S, type-304L stainless steel; C, type-A-106 grade-B carbon steel

[c]Elastic-response tests on all tees.

[d]Low-cycle fatigue-to-failure tests. Loadings: M, cyclic moment plus constant design pressure; P, cyclic internal pressure; MOP, cyclic moment plus zero (3 to 5 psi) internal pressure.

[e]Photoelastic stress analyses. Loadings: P, internal pressure; M, out-of-plane bending moment applied to the branch.

[f]Thermal-stress tests.

[g]Development of finite-element computer code and stress-analyses verification.

[h]Finite-element elastic analyses for all tees and comparison with experimental data.

[i]Analyses of the response of mixing tees to step and monotonic changes in fluid temperatures in branch and run. The ΔT indicates analysis for thermal stress caused by temperature differences.

Table 9. ANSI B16.9 tees tested at Southwest Research Institute

Tee No.	Nominal size (iron pipe size)	Material[a]	Manufacturer[b]
T-4	12 × 12 × 12-in. sched. 80	A106B	I
T-6	12 × 12 × 12-in. sched. 80	A106B	III
T-7	12 × 12 × 12-in. sched. 160	TP-304L	II
T-8	12 × 12 × 6-in. sched. 40	TP-304L	II
T-15	12 × 12 × 6-in. sched. 40	TP-304L	I

[a]A-106 grade-B carbon steel and type-304L stainless steel.
[b]Tees from three different manufacturers were used.

Table 10. ANSI B16.9 tees tested at Combustion Engineering, Inc.

Tee No.	Nominal size (iron pipe size)	Material[a]	Manufacturer[b]
T-10	24 × 24 × 24-in. sched. 40	A106B	III
T-11	24 × 24 × 24-in. sched. 160	A106B	III
T-12	24 × 24 × 10-in. sched. 40	A106B	III
T-13	24 × 24 × 10-in. sched. 160	A106B	III
T-16	24 × 24 × 24-in. sched. 10	TP-304L	III

[a]A-106 grade-B carbon steel and type-304L stainless steel.
[b]All the 24-in.-ips tees were made by the same firm.

on both the inside and outside surfaces, as indicated for the 0-90° quadrant of T-8 shown in Fig. 10. Five-foot-long pipe extensions were welded to each of the three tee outlets, and the assemblies were tested in one of two specially built loading frames, as shown in Fig. 11. One end

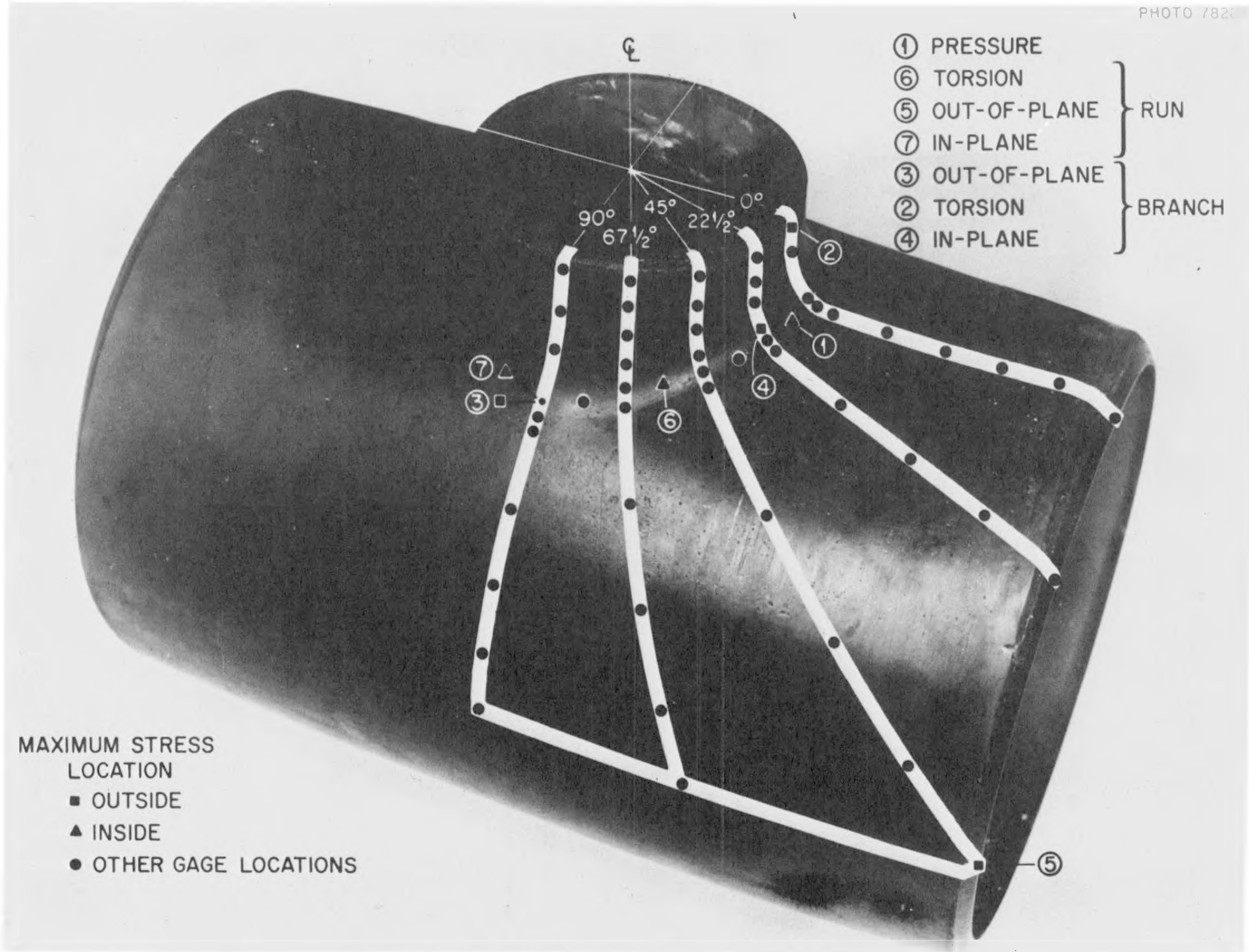

Fig. 10. Strain-gage locations in the 0° to 90° quadrant and maximum stress-intensity locations for internal-pressure and bending moment loads on T-8.

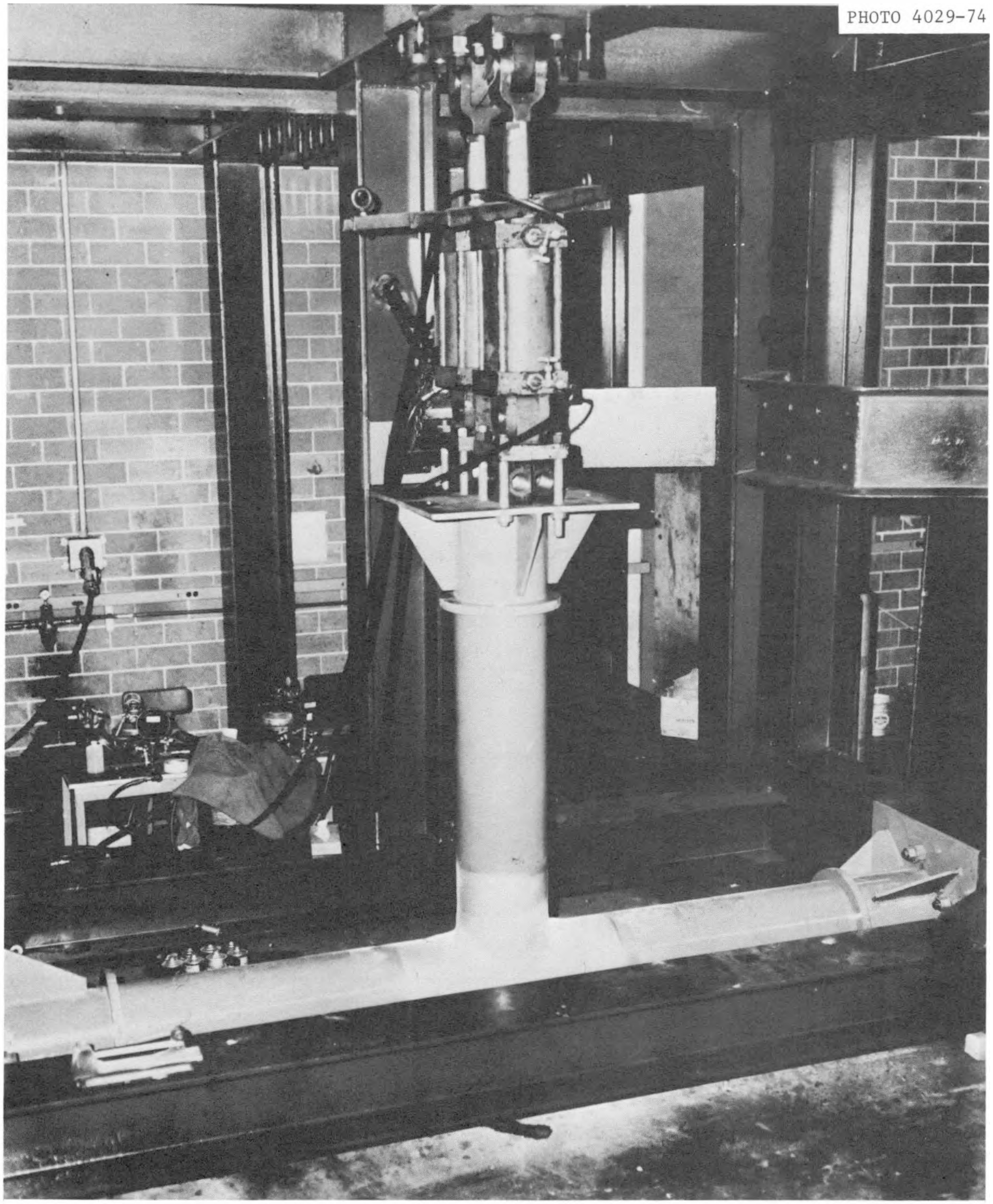

Fig. 11. Tee No. T-4 prior to the brittle-coating analysis.

of the run pipe was fixed to the loading frame; loading jacks were mounted on the other two ends. In some cases, the tees also were tested with brittle lacquer prior to mounting the outside strain gages in order to identify the high-stress regions.

Each tee in the series was tested under elastic-response conditions with 13 different loadings: internal pressure; six moments (in-plane, out-of plane, and torsion on both the branch and run pipe extensions); and 6 direct forces (axial thrust, and in-plane and out-of-plane shear). Normalized values for the maximum stresses are summarized in Table 11 for each of the tees and for each loading condition. The tabulated values are normalized to a nominal stress of 6 900 kPa (1000 psi) in the attached pipe. For pressure, the nominal stress $\sigma_{nom} = PD_o/2t$; for moment loadings,

Table 11. Values of the maximum normalized stress intensities for the 12-in.-ips tees tested at Southwest Research Institute

	Internal-pressure load (P)	Loadings on the branch					
		In-plane moment (M3Z)	In-plane shear (F3Z)	Out-of-plane moment (M3X)	Out-of-plane shear (F3Z)	Torsion (M3Y)	Axial force (F3Y)
T-4	4.417	2.250	2.191	2.732	2.762	2.738	7.513
T-6	3.310	2.268	2.095	2.709	2.640	2.662	6.276
T-7	4.425	2.189	2.444	2.002	1.882	1.730	5.063
T-8	2.700	1.206	1.158	2.241	1.961	1.165	5.917
T-15	3.654	1.610	1.547	2.307	2.178	1.110	7.607
		Loadings on the run					
		In-plane moment (M2Z)	In-plane shear (F2Y)	Out-of-plane moment (M2Y)	Out-of-plane shear (F2Z)	Torsion (M2X)	Axial force (F2X)
T-4		2.013	2.146	1.231	1.288	2.322	4.031
T-6		2.250	2.424	1.216	1.087	2.244	3.841
T-7		1.425	1.920	1.450	1.486	1.737	2.569
T-8		2.006	1.991	1.369	1.377	1.824	2.377
T-15		1.893	1.927	1.367	1.485	1.979	2.265

σ_{nom} = M/Z; for axial thrust, σ_{nom} = F/A; and for in-plane and out-of-plane shear, σ_{nom} = FL/Z. The symbols P, M, and F identify the loadings pressure, moment, and force respectively; D_o, t, Z, and A are for the nominal outside diameter, wall thickness, section modulus, and cross-sectional area of the corresponding pipe; and L is the length of the moment arm for transverse shear loading.

After the elastic-response tests were completed, each tee was fatigue tested to failure with a constant internal pressure equal to the Code-allowable design pressure and with a displacement-controlled alternating bending load on the branch pipe extension. For all of the tees, the total equivalent alternating elastic stress range was 1.15 MPa (167,000 psi). Fatigue-test results and comparisons with the calculated design lives are summarized in Table 12.

Table 12. Experimental fatigue life compared with calculated values for the 12-in.-ips tees tested at Southwest Research Institute

Tee	Constant internal pressure [kPa (psi)]	N_t[a]	N_c[b]	N_t/N_c	N_e[c]	N_t/N_e
T-4	13 270 (1925)	2,062	18	115	25	82
T-6	13 270 (1925)	1,309	18	73	25	52
T-7	22 340 (3240)	11,475	170	68	95	121
T-8	6 550 (950)	8,970	24	374	95	94
T-15	6 550 (950)	10,200	33	309	95	107

[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 indices of Table NB-3683.2-1.

[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 a maximum-stress range of 1.15 MPa (167,000 psi), as calculated by using the normalized stress intensities of Table 10 instead of the Code values for the stress indices.

As mentioned earlier, the draft of the summary report for the SwRI 12-in.-tee-test series has been completed and is currently being prepared for publication, and MRI is currently writing a similar report for the CE 24-in.-tee-test series. The draft for the latter should be completed by the end of June.

REFERENCES

1. *ASME Boiler and Pressure Vessel Code*, Sect. III, Div. 1, "Nuclear Power Plant Components," American Society of Mechanical Engineers, New York, 1974.

2. E. C. Rodabaugh and R. C. Gwaltney, *Elastic Stresses at Reinforced Nozzles in Spherical Shells with Pressure and Moment Loading*, ORNL-Sub-3131-9 (Apr. 28, 1975).

3. J. L. Mershon and E. C. Rodabaugh, *Preliminary Evaluation of Closely Spaced Reinforced Openings in a Cylindrical Pressure Vessel Under Internal Pressure Loading*, ORNL-Sub-3131-10 (June 30, 1975).

4. E. C. Rodabaugh, *Exploratory Study of the Feasibility of Developing Stress Indices for Thermal Gradients in Pressure Vessel Nozzles*, ORNL-Sub-3131-11 (May 12, 1975).

5. W. G. Dodge, *Secondary Stress Indices for Integral Structural Attachments to Straight Pipe*, ORNL-TM-3476 (June 1973); also in *Welding Research Council Bulletin 198*, Sept. 1974.

6. E. C. Rodabaugh, W. G. Dodge, and S. E. Moore, *Stress Indices at Lug Supports on Piping Systems*, ORNL-TM-4211 (June 1973); also in *Welding Research Council Bulletin 198*, Sept. 1974.

7. *ANSI B16.11-1973, Forged Steel Fittings, Socket-Welding and Threaded*, ASME, New York, 1966.

8. E. C. Rodabaugh and S. E. Moore, *Stress Indices for ANSI Standard B16.11 Socket-Welding Fittings*, ORNL-TM-4929 (Aug. 1975).

9. R. C. Gwaltney, J. W. Bryson, and S. E. Bolt, *Theoretical and Experimental Stress Analyses of ORNL Thin-Shell Cylinder-to-Cylinder Model 2*, ORNL-5021 (Oct. 1975).

10. R. L. Battiste et al., *Stress Analyses of Flat Plates with Attached Nozzles, Vol. 2. Experimental Stress Analyses of a Flat Plate with One Nozzle Attached*, ORNL-5044, Vol. 2 (July 1975). J. W. Bryson and W. F. Swinson, *Stress Analyses of Flat Plates with Attached Nozzles, Vol. 3. Experimental Stress Analyses of a Flat Plate with Two Closely Spaced Nozzles of Equal Diameter Attached*, ORNL-5044, Vol. 3 (Dec. 1975).

11. W. G. Dodge, S. E. Bolt, and F. M. O'Hara, Jr., *Experimental Stress Analysis of Machined 10-in.-diam Pipe Elbows with Specified Geometric Distortions*, ORNL-TM-4834 (to be published).

12. E. C. Rodabaugh, F. M. O'Hara, Jr., and S. E. Moore, *FLANGE: A Computer Program for the Analysis of Flanged Joints with Ring-Type Gaskets*, ORNL-5035 (in preparation).

13. S. P. Timoshenko and S. Wainowsky-Krieger, *Theory of Plates and Shells*, 2nd Ed., McGraw-Hill, New York, 1959.

14. J. W. Bryson, J. P. Callahan, and R. C. Gwaltney, *Stress Analyses of Flat Plates with Attached Nozzles, Vol. 1. Comparison of Stresses in a One-Nozzle-to-Flat-Plate Configuration and in a Two-Nozzle Configuration with Theoretical Predictions*, ORNL-5044, Vol. 1 (July 1975).

15. A. N. Gantayat and G. H. Powell, *Stress Analysis of Tee Joints by the Finite Element Method*, University of California Report UC-SESM-73-6 (ORNL-Sub-3193-1) (Feb. 1973).

16. E. C. Rodabaugh, *Phase Report No. 115-8, Stresses in Out-of-Round Pipe Due to Internal Pressure*, ORNL-TM-3244 (Jan. 1971).

17. *USA Standard Code for Pressure Piping, Nuclear Power Piping, USAS B31.7*, American Society of Mechanical Engineers, New York, 1969.

18. R. L. Johnson, *Photoelastic Determination of Stresses in ASA B16.9 Tees*, Westinghouse Research Report 71-9E7-PHOTO-R2 (Nov. 1971).

19. G. H. Powell, *Finite Element Analysis of Elasto-Plastic Tee Joints*, ORNL-Sub-3193-2 (Sept. 1974).

20. R. W. Clough, G. H. Powell, and A. N. Gantayat, "Stress Analysis of B16.9 Tees by the Finite Element Methods," Paper F4/7, *First International Conference on Structural Mechanics in Reactor Technology*, Berlin, Germany, Sept. 20-24, 1971.

21. R. E. Textor, *User's Guide for SHFA: Steady-State Heat Flow Analysis of Tee Joints by the Finite Element Method*, UCCND/CSD/INF-60 (in press).

22. S. C. Grigory and G. W. Deel, *Experimental Stress Analysis and Fatigue Test of ASA Standard B16.9 Tees, Summary Report, A106 Grade B Tee Number T-4*, Southwest Research Institute, San Antonio, Tex. (Jan. 15, 1971).

23. S. C. Grigory and G. W. Deel, *Experimental Stress Analysis and Fatigue Test of ASA Standard B16.9 Tees, Summary Report, 304L Stainless Steel Tee Number T-7*, Southwest Research Institute, San Antonio, Tex. (May 24, 1971).

24. S. C. Grigory and G. W. Deel, *Experimental Stress Analysis and Fatigue Test of ASA Standard B16.9 Tees, Summary Report, A106 Grade B Tee Number T-6*, Southwest Research Institute, San Antonio, Tex. (Dec. 2, 1971).

25. S. C. Grigory and G. W. Deel, *Experimental Stress Analysis and Fatigue Test of ASA Standard B16.9 Tees, Summary Report, 304L Stainless Steel Tee Number T-8*, Southwest Research Institute, San Antonio, Tex. (Dec. 8, 1971).

26. S. C. Grigory and G. W. Deel, *Experimental Stress Analysis and Fatigue Test of ASA Standard B16.9 Tees, Summary Report, 304L Stainless Steel Tee Number T-15*, Southwest Research Institute, San Antonio, Tex. (Dec. 8, 1971).

27. J. K. Hayes and B. Roberts, "Experimental Stress Analysis of 24-In. Tees," *J. Eng. Ind.* 93, 919-28 (1971).

28. J. K. Hayes, *Test Report on Experimental Stress Analysis of a 24-Inch Diameter Tee (T-10)*, Combustion Engineering Report CENC-1169 (ORNL-Sub-3310-1) (Nov. 1971).

29. J. E. Wilson, *Test Report on Experimental Stress Analysis of a 24-Inch Diameter Tee (T-11)*, Combustion Engineering Report CENC-1186 (ORNL-Sub-3310-2) (Sept. 1972).

30. D. R. Henley, *Test Report on Experimental Stress Analysis of a 24-Inch Diameter Tee (ORNL T-13)*, Combustion Engineering Report CENC-1189 (ORNL-Sub-3310-3) (Mar. 1975).

31. D. R. Henley, *Test Report on Experimental Stress Analysis of a 24-Inch Diameter Tee (ORNL T-12)*, Combustion Engineering Report CENC-1237 (ORNL-Sub-3310-4) (Apr. 1975).

32. D. R. Henley, *Test Report on Experimental Stress Analysis of a 24-Inch Diameter Tee (ORNL T-16)*, Combustion Engineering Report CENC-1239 (ORNL-Sub-3310-5) (June 1975).

33. R. L. Weed and N. E. Johnson, *Experimental Stress Analysis and Fatigue Test of ANSI B16.9 Tees*, ORNL-Sub-4356, Report No. 1 (to be published).

ORNL/NUREG/TM-18
NRC-5

INTERNAL DISTRIBUTION

1. R. L. Battiste
2. M. Bender
3. J. J. Blass
4. S. E. Bolt
5. R. H. Bryan
6-11. J. W. Bryson
12. J. H. Butler
13. J. P. Callahan
14. C. J. Claffey
15. J. A. Clinard
16. C. W. Collins
17. W. E. Cooper
18. J. M. Corum
19. W. B. Cottrell
20. J. S. Crowell
21. F. L. Culler
22. W. G. Dodge
23. G. G. Fee
24. M. H. Fontana
25. W. R. Gall
26. M. J. Goglia
27-28. W. L. Greenstreet
29. R. C. Gwaltney
30. W. O. Harms
31. G. Hofmann
32. H. W. Hoffman
33. P. R. Kasten
34. K. C. Liu
35. R. E. MacPherson
36. W. J. McAfee
37. W. J. McCarthy, Jr.
38. H. C. McCurdy
39. J. G. Merkle
40-54. S. E. Moore
55. F. M. O'Hara (consultant)
56. T. W. Pickel
57. H. Postma
58. C. E. Pugh
59. M. Richardson
60. D. N. Robinson
61. R. A. Schmidt
62. J. E. Smith
63. I. Spiewak
64. W. C. T. Stoddart
65. J. J. Taylor
66. R. E. Textor
67. D. B. Trauger
68. G. D. Whitman
69. W. J. Wilcox
70. G. T. Yahr
71. ORNL Patent Office
72-74. Central Research Library
75. Document Reference Section
76-81. Laboratory Records Department
82. Laboratory Records, ORNL RC

Subcontractors

83. E. C. Rodabaugh, Battelle-Columbus Laboratories, Columbus, OH 43201
84-85. N. E. Johnson, Mechanics Research, Inc., Oak Ridge, TN 37830

EXTERNAL DISTRIBUTION

86-93. Director, Office of Nuclear Regulatory Research, NRC, Washington, DC 20555
94. Director, Reactor Division, ERDA, ORO
95. Research and Technical Support Division, ERDA-ORO
96-179. Design Criteria for Piping and Nozzles Program Distribution
180-483. Given distribution as shown under categories NRC-1 and NRC-5 (25 copies — NTIS)

☆ U.S. GOVERNMENT PRINTING OFFICE: 1976-748-189/375